Heating Ventilating Air Conditioning Guide 1945

HEATING VENTILATING AIR CONDITIONING GUIDE

1945

An Instrument of Service prepared for the Profession—Containing a

Technical Data Section

of reference material on the design and specification of heating, ventilating and air conditioning systems based on—the Transactions—the Investigations of the Research Laboratory and Co-operating Institutions—and the Practice of the Members and Friends of the Society

together with a

Manufacturers' Catalog Data Section

Containing Essential and Reliable Information Concerning Modern Equipment

also

The Roll of Membership of the Society

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PREFACE TO THE 23rd EDITION

THE HEATING, VENTILATING, AIR CONDITIONING GUIDE 1945 is the 23rd edition of this widely known handbook and is the fourth to be released under wartime conditions.

Several steps were taken to comply with wartime limitations in the use of paper, including a change in format to increase the amount of text per page and the omission of the geographical list of members. The change in format afforded an opportunity to examine a larger part of THE GUIDE than has been customary and consequently, wherever practicable, much new valuable data which became available during the year have been incorporated.

Further progress has been made in the current edition in the adoption of *A.S.A.* Standard Symbols and Abbreviations.

Chapter 4 on Heat Transmission of Building Materials has been carefully reviewed by the A.S.H.V.E. Technical Advisory Committee on Insulation. Some new materials have been listed and separate columns have been provided in the tables for the values of k and C to facilitate their use. Chapter 6 has been rearranged, and a column showing annual minimum temperatures has been added to Table 1. Engineers may compare these data with the locally accepted design temperature before making load calculations.

The tabular data in Chapter 8, Fuels and Combustion, have been checked and revised. A new chart for sizing chimneys for gas appliances and additional data on chimney construction have been added to Chapter 9. New codes for rating stokers have recently been adopted and references to such codes have been added to Chapter 10, Automatic Fuel Burning Equipment. The discussion in Chapter 11, Estimating Fuel Consumption, on the use of fuel consumption records and degree-day data has been considerably amplified.

Chapter 12, Heating Boilers, has been revised to include present accepted practice in rating boilers and to clarify the meaning of various terms referring to heating loads.

The subjects, Steam Heating Systems and Piping, have been combined in Chapter 14 and new data on high pressure systems and pipe size selection have been included. The chapter has been rearranged to facilitate reference to the different types of systems.

In Chapter 17, Pipe Insulation, heat transmission data have been revised to include generally accepted values for representative materials as now manufactured. Chapter 21, Unit Heaters, Unit Ventilators, Unit Humidifiers, has been expanded in the light of present accepted practice. Numerous improvements in nomenclature have been made in revising Chapter 23, Dehumidification.

Chapter 26, Spray Equipment, has been revised particularly with reference to cooling tower practice. Laws of fan performance have been stated in detail in Chapter 29, Fans, to which laws of homologous fans have also been added. Recently adopted names and definitions of types of fans have also been included. In Chapter 31, Air Duct Design, attention of the designer is called to the importance of including a factor of safety when applying friction loss values.

Important changes which increase the value of Chapter 35, Motors and Motor Controls, include revised performance curves and the addition of: typical motor specifications, recommended control combinations, a glossary of motor terms, and a section on gear motors. A section on airplane air conditioning has been added to Chapter 37, Transportation

Air Conditioning. The chapter has also been revised to include many new items relating to present railroad car heating practice.

New material and references have been added to Chapter 39, Industrial Air Conditioning. Chapter 40, Industrial Exhaust Systems, has been extensively changed with the addition of a number of tables and data which will be useful for practical design purpose. A section on Electronic Heating has been added to Chapter 44, Electric Heating.

Chapter 45, Panel Heating and Radiant Heating, has been rewritten so that the first part of the chapter deals with the influence of radiant heat on the occupant and the objectives of radiant heating. The latter part of the chapter outlines a new method for practical calculation of panel heating and radiant heating. The method is illustrated by a problem in which simplifying assumptions are made for the purpose of reducing the work required to design a satisfactory system.

A chart showing friction loss in fairly rough pipe has been added to Chapter 46, Hot Water Supply Systems. Several new and revised definitions have been added to Chapter 47, Terminology. In Chapter 48, Abbreviations, Symbols, Standards, separate sections have been used for Abbreviations and for Symbols. Conversion equations and factors listed have been corrected by the National Bureau of Standards.

Many Society members and other engineers, although busily engaged in activities related to war production, have contributed generously of their knowledge and time in preparing the 1945 edition. In addition to those mentioned in previous editions, who are responsible for past contributions, grateful acknowledgment is made also to those who have made this issue a valuable reference work for the profession and industry by reviewing chapters or preparing major revisions:

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A carefully prepared Catalog Data Section, listing an enlarged number of products from 212 leading manufacturers, constitutes a valuable reference list for the designer seeking equipment for any type of heating, ventilating and air conditioning installation. For ready reference the section is fully indexed with cross-references.

The 1945 edition of The Guide maintains the established tradition of constant improvement from year to year resulting from close contact of members of our Society with the latest practice in the field of Heating, Ventilating and Air Conditioning.

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CODE *of* ETHICS *for* ENGINEERS

ENGINEERING work has become an increasingly important factor in the progress of civilization and in the welfare of the community. The engineering profession is held responsible for the planning, construction and operation of such work and is entitled to the position and authority which will enable it to discharge this responsibility and to render effective service to humanity.

That the dignity of their chosen profession may be maintained, it is the duty of all engineers to conduct themselves according to the principles of the following Code of Ethics:

1—The engineer will carry on his professional work in a spirit of fairness to employees and contractors, fidelity to clients and employers, loyalty to his country and devotion to high ideals of courtesy and personal honor.

2—He will refrain from associating himself with or allowing the use of his name by an enterprise of questionable character.

3—He will advertise only in a dignified manner, being careful to avoid misleading statements.

4—He will regard as confidential any information obtained by him as to the business affairs and technical methods or processes of a client or employer.

5—He will inform a client or employer of any business connections, interests or affiliations which might influence his judgment or impair the disinterested quality of his services.

6—He will refrain from using any improper or questionable methods of soliciting professional work and will decline to pay or to accept commissions for securing such work.

7—He will accept compensation, financial or otherwise, for a particular service, from one source only, except with the full knowledge and consent of all interested parties.

8—He will not use unfair means to win professional advancement or to injure the chances of another engineer to secure and hold employment.

9—He will cooperate in upbuilding the engineering profession by exchanging general information and experience with his fellow engineers and students of engineering and also by contributing to work of engineering societies, schools of applied science and the technical press

10—He will interest himself in the public welfare in behalf of which he will be ready to apply his special knowledge, skill and training for the use and benefit of mankind.

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CHAPTER 1

Thermodynamics of Air and Water Mixtures

Dry Air, Specific Enthalpy, Water Vapor, Moist Air, Dalton's Law, Humidity Ratio, Relative Humidity, Dew-Point, Enthalpy, Thermodynamic Wet-bulb Temperature, Mollier Diagram, Typical Air Conditioning Processes, Adiabatic Saturation, Psychrometric Chart, Steady Flow Energy Equation, U. S. Standard Atmosphere

THE working substance of the air conditioning engineer may be regarded, for the purpose of analysis, as a mixture of only two constitutents, dry air and water. The mixture may consist of two, and possibly three distinct phases, solid, liquid and vapor. The vapor phase is conveniently referred to as *moist air* and is regarded as a mixture of *dry air* and *water vapor*.

DRY AIR

Composition. Dry air is itself a mixture of several gases, but its composition is subject to such slight variation that it may be regarded as fixed. According to *International Critical Tables*, the mol-fraction composition of dry air is given by the first column of figures in Table 1. Molecular weights are given in the second column; the last figure in the third column is the *apparent* molecular weight of the mixture; the fourth column of figures gives the ordinary weight-fraction composition.

It is well known that dry air contains other gases besides those listed in Table 1; but these are present in such minute amounts that they can be grouped together as argon. Values in the lower section of Table 1 give the approximate mol-fraction composition of what is called argon in the upper portion of the table.

In physical and chemical thermodynamics, there is a distinct advantage in using a different unit of weight, the mol, for each different substance involved. A mol of oxygen weighs 32.000 lb as a matter of definition; a mol of any other substance is a weight, in pounds, equal to its molecular weight.

Specific Volume and Density

The ratio of total volume to total weight is called *specific volume*, *v*. In the English system, volume is expressed in cubic feet and weight in pounds; hence, specific volume is expressed in cubic feet per pound. The reciprocal of specific volume, that is, weight per unit volume is called *weight density*, *d*. The unit of density is the pound per cubic foot.

The earliest investigation into the relation between pressure, specific volume, and temperature for gases was made by Boyle (1661) who was able to confirm the hypothesis that the volume of a given weight of gas should vary inversely as the absolute pressure if temperature is maintained constant. Thus, within the limits of his experimental error Boyle found that at constant temperature the product pv, pressure times specific volume, has a constant value over a considerable range of pressures. These results are best visualized by plotting values of the product pv as ordinate against values of pressure p itself as abscissa. According to Boyle's experimental findings lines of constant temperature (isotherms) of a gas are *straight* and *horizontal* on this pv, p-plane.

The first rough experiments of Charles (1787) and the subsequent more refined experiments of Gay-Lussac (1802) suggested the possibility

TABLE 1. COMPOSITION OF DRY AIR

GAS	MOL PER MOL DRY AIR		LB PER MOL		LB PER MOL DRY AIR	LB PER LB DRY AIR
Nitrogen	0.7803	×	28.016	=	21.861	0.7547
Oxygen	0.2099	×	32.000	=	6.717	0.2319
Carbon Dioxide	0.0003	×	44.003	=	0.013	0.0004
Hydrogen	0.0001	×	2.016	=	0.000	0.0000
Argon	0.0094	×	39.944	=	0.376	0.0130
	1.0000				28.967	1.0000

COMPOSITION OF ARGON	MOL PER MOL DRY AIR
Argon	0.00933
Neon	0.000018
Helium	0.000005
Krypton	0.000001
Xenon	....
	0.00935

of establishing a universal temperature scale such that the product pv *for any gas* is simply proportional to temperature measured on this scale in accordance with Equation 1,

$$pv = BT \tag{1}$$

where B is a constant characteristic of the given gas. Referring to the graphical representation previously described in which the product pv is plotted as ordinate against pressure p as abscissa, the vertical spacing of the isotherms should be such that the ordinates to any two isotherms are in the ratio of corresponding absolute temperatures and therefore in the same ratio for any gas.

Precise measurements by modern methods have shown that the experimental findings of Boyle, Charles and Gay-Lussac are only approximately correct. In the range of sufficiently low pressures the isotherms of gases are indeed *straight* on the pv, p-plane; but they are *not horizontal* in accordance with Boyle's Law, being inclined downward to the right at relatively low temperatures, upward to the right at higher temperatures. Extrapolation of each isotherm to zero pressure has revealed the remarkable fact that the limiting value of the product pv thus obtained is strictly proportional to absolute temperature as suggested by Equation 1, this strict proportionality providing an accurate basis for the establishment of the absolute temperature scale.

The experimental facts of the preceding paragraph are expressed mathematically by Equation 2,

$$pv = BT - A(T)\,p \tag{2}$$

where

p = absolute pressure, pounds per square foot.
v = specific volume, cubic feet per pound.
B = a constant depending on the molecular weight of the gas.
T = absolute temperature, degrees Fahrenheit.
$A(T)$ = a temperature function called *second virial coefficient*, cubic feet per pound [2].* The name undoubtedly originated from consideration of Clausius' Virial Theorem according to which the mean kinetic energy of a molecular

*Bracketed numbers refer to references at end of chapter.

aggregate is equal to the mean value of a quantity, which Clausius called the *virial* of the system, depending solely on the forces acting upon the molecules and not upon the motion of the molecules. This name is used extensively. For some gases, the magnitude of the second virial coefficient can be predicted from theory; but, at present, direct experimental measurements are more reliable.

It will appear in what follows that the error committed in computing values of specific volume from Equation 1 instead of Equation 2 is extremely small. Thermodynamically, however, the former would deny the effect of pressure on the thermal properties of a gas which experiment shows to be appreciable. Therefore Equation 1 cannot be made the basis of an accurate analysis.

The numerical value of the constant B in Equation 2 is different for every different gas, but can be calculated if the molecular weight m, pounds per mol, is known; for the product mB is a universal gas constant R, namely,

$$R = 1545.4$$

Example 1. Find the value of B for dry air and water vapor.

Solution. $B_a = 1545.4 \div 28.967 = 53.351$
$B_w = 1545.4 \div 18.0154 = 85.782$

The temperature function $A(T)$, the so-called *second virial coefficient*, expresses the effect of intermolecular forces. It is positive at low temperatures where these forces are predominantly attractive, negative at higher temperatures where they are predominantly repulsive. It is known with satisfactory accuracy for both dry air and water vapor. Values of specific volume are listed in Table 2 for dry air at standard atmospheric pressure (29.921 in. Hg) as computed from Equation 2.

TABLE 2. SPECIFIC VOLUME OF DRY AIR[a] AT 29.921 IN. HG

TEMP F t	CU FT PER LB v_a	TEMP F t	CU FT PER LB v_a	TEMP F t	CU FT PER LB v_a
−96	9.1488	32	12.3888	160	15.6229
−64	9.9597	64	13.1977	192	16.4310
−32	10.7699	96	14.0063	224	17 2389
0	11.5796	128	14.8147	256	19.0467

[a]Prepared by John A. Goff.

The fact that $A(T)$ is multiplied by pressure in Equation 2 means that intermolecular forces vanish at zero pressure and infinite volume where infinite distances separate the molecules. The finite value of the product pv at zero pressure is due entirely to the translational kinetic energy of the molecules. In ordinary calculations not requiring too great accuracy, the effect of intermolecular forces may be ignored and Equation 2 simplified to

$$pv = BT \tag{1}$$

Example 2. Calculate an approximate value for the specific volume of dry air at 64 F, 29.921 in. Hg.

Solution. $v = \dfrac{53.351 \times 523.70}{29.921 \times 0.49115 \times 144} = 13.200$ cu ft per pound.

Note: This answer may be compared with the value in Table 2. The difference is due to intermolecular forces. It should not be concluded, however, that because the effect of intermolecular forces on the volume is so small these forces can be ignored entirely.

The relationship of Equation 1 expresses certain familiar laws approximately true for gases at not too high pressures. Thus, *with temperature constant, the volume of a given weight of gas is inversely proportional to its absolute pressure* is a statement of Boyle's Law. If v_1 denotes the specific volume at absolute pressure p_1, then *at the same temperature*, the specific volume v_2 at absolute pressure p_2 is approximately,

$$v_2 = v_1 \left(\frac{p_1}{p_2}\right)$$

Also, *with pressure constant the volume of a given weight of gas is directly proportional to its absolute temperature* is a statement of Charles' Law. If v_1 denotes the specific volume at absolute temperature T_1, then *at the same pressure*, the specific volume v_2 at absolute temperature T_2 is approximately,

$$v_2 = v_1 \left(\frac{T_2}{T_1}\right)$$

Absolute Temperature

For the range 0 to 660 C, the standard temperature scale is the International Centigrade Scale, namely, the readings of a platinum resistance thermometer standardized at the ice-point (0 C), the steam-point (100 C) and the sulphur-point (444.60 C). The corresponding Fahrenheit scale t used in scientific work is derived from the International Centigrade Scale by means of the relation,

$$t = 1.8 \text{ (Int. Cent. Temp.)} + 32 \qquad (3)$$

Temperatures on the absolute Fahrenheit scale are then obtained by adding 459.70 according to the equation

$$T = t + 459.70 \qquad (4)$$

Absolute temperatures computed from Equations 3 and 4 are *practically* identical with the fundamental thermodynamic temperatures to which the zero-pressure values of the product pv for gases are proportional in accordance with Equation 2.

Specific Enthalpy

Most air conditioning processes are of the steady-flow type. In steady flow the energy convected with the fluid crossing a given section is the sum of (a) kinetic energy due to velocity, (b) gravitational energy due to elevation, (c) enthalpy due to the condition of temperature, pressure and composition at a given section. It is clear, therefore, that in order to apply the Law of Conservation of Energy to steady-flow processes, information regarding the enthalpy is needed.

Recent developments in quantum mechanics have made it possible to calculate the zero-pressure specific enthalpy of a gas from spectroscopic measurements, and with a degree of accuracy exceeding that with which this property can be inferred from direct calorimetric measurements. Available data for each gas listed in Table 1 have been assembled and critically examined; and from them have been calculated best values for the specific enthalpy of dry air at zero pressure. These are listed in Table 3. The unit of energy is the Btu which is related to the foot-pound as follows:

$$1 \text{ Btu} = 778.18 \text{ ft-lb}^1 \qquad (5)$$

[1]This conversion factor is not exact by definition, but involves an experimental determination of the relation between the absolute and the standard electrical units of energy. The value 1 int. joule = 1 00019 abs. joule, recommended by Osborne, Stimson and Ginnings [8] was used.

TABLE 3. SPECIFIC ENTHALPY OF DRY AIR AT ZERO PRESSURE[a]

TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_a^o	MEAN SPECIFIC HEAT $\left[c_p^o\right]_o^t$	TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_a^o	MEAN SPECIFIC HEAT $\left[c_p^o\right]_o^t$	TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_a^o	MEAN SPECIFIC HEAT $\left[c_p^o\right]_o^t$
−96	−22.839	0.2393	32	7.796	0.2395	160	38.529	0.2400
−64	−15.186	0.2393	64	15.466	0.2396	192	46.238	0.2401
−32	−7.529	0.2394	96	23.145	0.2397	224	53.962	0.2403
0	+0.131	0.2394	128	30.831	0.2398	256	61.702	0.2405

[a]Prepared by John A. Goff from published data computed from spectroscopic measurements.

In Table 3 are also listed values of *mean zero-pressure specific heat* for the range 0 to t F. This is simply the increase of specific enthalpy from 0 to t F, divided by the increase of temperature or, with 0 F as the reference point, by the temperature itself. The numerical values indicate that a rounded figure of 0.24 Btu per pound can be used in ordinary calculations.

Applying well known identical relations of thermodynamics to Equation 2, the following expression for specific enthalpy, valid at not too high pressures, is obtained

$$h = h^0 + \left[T^2 \frac{d\,(A/T)}{dT}\right] p \tag{6}$$

where h^0 denotes specific enthalpy at zero pressure. This equation emphasizes that the effect of pressure on specific enthalpy is not so much due to the second virial coefficient A itself as to its variation with temperature. In other words, the pressure effect may be much more important than the corresponding effect on specific volume. Values of the specific enthalpy of dry air at standard atmospheric pressure (29.921 in. Hg) as computed from Equation 6 are given in Table 4.

Reference Point. It is desired to give some prominence to the choice of reference point. As energy, and therefore enthalpy, is purely relative, any convenient state can be selected at which to assign the value *zero* to specific enthalpy. The state chosen is 0 F, 29.921 in. Hg. Perhaps the only really valid argument for this particular choice is that, for ordinary calculations at, or near, atmospheric pressure, a very simple equation can be used, namely,

$$h_a = 0.24t \tag{7}$$

WATER VAPOR

Saturation Pressure. It is common knowledge that a substance like water can exist in at least three distinct phases, solid (ordinary ice), liquid and vapor; and that under certain conditions two or more phases can *co-exist* in stable equilibrium. For example, steam having a quality of

TABLE 4. SPECIFIC ENTHALPY OF DRY AIR[a] AT 29.921 IN. HG

TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_a	SPECIFIC HEAT $\left[c_p\right]_o^t$	TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_a	SPECIFIC HEAT $\left[c_p\right]_o^t$	TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_a	SPECIFIC HEAT $\left[c_p\right]_o^t$
−96	−23.035	0.2399	32	7.680	0.2400	160	38.454	0.2403
−64	−15.356	0.2399	64	15.363	0.2400	192	46.172	0.2405
−32	−7.678	0.2399	96	23.053	0.2401	224	53.903	0.2406
0	0.000	0.2400	128	30.749	0.2402	256	61.649	0.2408

[a]Prepared by John A Goff.

98 per cent is a mixture of two co-existing phases, vapor and liquid, 98 per cent by weight being vapor and 2 per cent by weight, liquid. When two phases can co-exist in stable equilibrium, each is said to be *saturated* with respect to the other.

One of the important problems of thermodynamics is to formulate the conditions for *saturation* in mathematical terms. The answer to the problem can be stated quite generally as equality, between the several co-existing phases, of (a) pressure, (b) temperature, (c) each component *chemical potential.*

In the case of a pure substance like water, containing a single component, there is only one component chemical potential; and this becomes identical with a thermodynamic property called *specific free enthalpy* denoted by the letter g (Btu per pound) and defined by the equation:

$$g = h - Ts$$

where

h = specific enthalpy, Btu per pound.
T = absolute temperature, degrees Fahrenheit.
s = specific entropy, Btu per pound per degree Fahrenheit.

To illustrate, *liquid* water at 212 F, 14.696 lb per square inch has a specific free enthalpy of $180.07 - 671.70 \times 0.3120 = -25.90$ Btu per pound. At the same temperature and pressure, water *vapor* has a specific free enthalpy of $1150.4 - 671.70 \times 1.7566 = -25.90$ Btu per pound. The numerical data used in these calculations are to be found in the steam tables[2]. Since the two specific free enthalpies are equal *at the same temperature and the same pressure,* the two phases can co-exist in stable equilibrium to form a saturated mixture and are therefore saturated with respect to each other.

But suppose that a different pressure had been assumed, the temperature being 212 F as before; for example, assume a pressure of 14 lb per square inch. The specific free enthalpy of the liquid phase will be practically the same as before, but that of the vapor phase will change from -25.90 to -32.84 Btu per pound, most of this change being due to change of entropy which, in the case of a vapor, depends markedly upon the pressure. Since the specific free enthalpies of the two phases are no longer equal, they cannot co-exist in stable equilibrium and neither is saturated. As a matter of fact the vapor is superheated while the liquid is supersaturated.

From this analysis it will be seen that *to a given temperature T there corresponds a definite saturation pressure p_s.* This is also called the *vapor pressure* of the liquid or solid as the case may be. It will also be seen that a working definition of saturation can only be arrived at by application of the fundamental laws of thermodynamics.

Referring specifically to the vapor phase, if the actual pressure is *less* than the saturation pressure corresponding to the actual temperature, the vapor is said to be *superheated*; if it is *greater*, as it may well be under proper circumstances, the vapor is said to be *supersaturated.* Values of the saturation pressure of pure water are given in Table 6[3].

Specific Volume

Accurate values of the specific volume of water vapor at pressures equal or near the saturation pressure (for the given temperature) can be

[2]Thermodynamic Properties of Steam, by J. H. Keenan and F. G. Keyes, published by John Wiley & Sons, Inc., 1936, of which Table 8 is an abridgment.

[3]Strictly speaking the values listed in Table 6 are not values of p_s as labeled, but of p'_s (Equation 13b) with the Dalton Factor (*DF*) taken to be unity.

computed from Equation 1 since the second virial coefficient $A(T)$ is known with satisfactory accuracy. Usually, however, the desired information can be read directly from the steam tables. Values for the specific volume of the *saturated* vapor, v_g, are also listed in Table 8.

Specific Enthalpy

The zero-pressure specific enthalpy, as calculated by A. R. Gordon from spectroscopic measurements, has recently been corrected for distortion of the water molecules due to centrifugal forces. Best values at present available are listed in Table 5.

From the numerical values of mean specific heat, it is clear that for ordinary calculations the following simple relation may be used:

$$h_w^\circ = 0.444t + 1061 \tag{8}$$

Reference Point. The reference point for water has been chosen as *saturated liquid at 32 F* in conformity with usual steam table practice. In order to refer the zero-pressure values of specific enthalpy to this

TABLE 5. SPECIFIC ENTHALPY OF WATER VAPOR AT ZERO PRESSURE[a]

TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_w°	MEAN SPECIFIC HEAT $[c_p^\circ]_0^t$	TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_w°	MEAN SPECIFIC HEAT $[c_p^\circ]_0^t$	TEMP F t	SPECIFIC ENTHALPY BTU PER LB h_w°	MEAN SPECIFIC HEAT $[c_p^\circ]_0^t$
−96	1018.61	0.4425	32	1075.28	0.4435	160	1132.38	0.4455
−64	1032.76	0.4427	64	1089.51	0.4440	192	1146.76	0.4462
−32	1046.92	0.4429	96	1103.76	0.4444	224	1161.20	0.4469
0	1061 09	0.4431	128	1118.05	0 4450	256	1175.70	0.4477

[a]Prepared by John A Goff from published data comi 'ited from spectroscopic measurements.

datum, best available information regarding latent heat, saturation pressure and second virial coefficient at 32 F has been used. The values in Table 5 do not agree exactly with those in the steam tables, but do agree with later information from the National Bureau of Standards [8].

MOIST AIR

Dalton's Law. Having accurate information regarding the thermodynamic properties of dry air and water vapor *separately*, it is desired to predict the properties of moist air which is regarded as a mixture of these two constituents. Statistical mechanics furnishes a starting point in the form of a prediction that, at not too high pressures,

$$Pv = RT - \left[A_{aa}x^2 + 2A_{aw}\,x\,(1 - x) + A_{ww}\,(1 - x)^2\right]P \tag{9}$$

where

P = observed pressure, pounds per square foot.

v = specific volume, cubic feet per *mol.*

A_{aa} = second virial coefficient for the dry air expressing the effect of forces between air—air molecules, cubic feet per *mol.*

A_{ww} = second virial coefficient for the water vapor, expressing the effect of forces between water—water molecules, cubic feet per *mol.*

A_{aw} = *interaction constant* expressing the effect of forces between air—water molecules, cubic feet per *mol.*

x = mol-fraction of dry air in the mixture, mols dry air per mol mixture.

Equation 9 will be recognized as a generalization of Equation 2. Both A_{aa} and A_{ww} are known; but until recently no reliable information on the

interaction constant A_{aw} has been available. Preliminary results of a cooperative investigation between the A.S.H.V.E. and the Towne Scientific School, University of Pennsylvania, have indicated that the ratio $2A_{aw}/(A_{aa} + A_{ww})$ has an approximately constant value $\lambda = 0.075$ [10]. However, before attempting to make use of this information it is advisable, in the interest of simplicity, to first ignore the complications arising from intermolecular forces.

Now, in the absence of intermolecular forces, each constituent gas in a mixture such as moist air would behave exactly as if it alone occupied the volume V at the temperature T of the mixture and: (1) the observed pressure P would be the sum of individual *partial* pressures p; (2) the total enthalpy H would be the sum of the individual enthalpies. This is the essence of Dalton's Law of Partial Pressures.

Referring to dry air by the subscript a and, to water vapor by the subscript w, Dalton's Law would predict

$$V = \frac{n_a RT}{p_a} = \frac{n_w RT}{p_w} = \frac{(n_a + n_w)\, RT}{P} \tag{10a}$$

where

$$P = p_a + p_w \tag{10b}$$

From these equations are easily obtained,

$$\frac{n_w}{n_a} = \frac{p_w}{P - p_w} \quad \text{or} \quad \frac{p_w}{P} = \frac{n_w/n_a}{1 + n_w/n_a} \tag{10c}$$

in which,

p_a = partial pressure of the *dry air*.
p_w = partial pressure of the *water vapor*.
P = observed pressure of the mixture.
n_a = weight of *dry air* (mols).
n_w = weight of *water vapor* (mols).

Humidity Ratio

In Equation 10c the ratio by weight of water vapor to dry air, n_w/n_a, is expressed in mols per mol. Most engineers prefer to express it in pounds per pound which can easily be done, since the molecular weights of both water vapor (18.0154 lb per mol) and of dry air (28.967 lb per mol) are known. Thus Equation 10c becomes

$$W = 0.62193 \frac{p_w}{P - p_w} \quad \text{or} \quad \frac{p_w}{P} = \frac{W}{0.62193 + W} \tag{11}$$

There is little doubt but that the weight ratio W is the most convenient parameter in terms of which to express the composition of moist air; but to choose a suitable name and one that would have general acceptance has always been a perplexing problem. In previous issues of the GUIDE, *specific humidity* was adopted even though it was recognized that the adjective *specific* should properly refer to weight of water vapor per pound of *mixture*, and not per pound of dry air. Various other names have been proposed from time to time including: mixing ratio, proportionate humidity, density ratio, absolute humidity. It is believed that the name *humidity ratio* is most suggestive of the meaning which it is desired to express, that it violates no well established usage as does the name specific humidity and that its adoption will avoid much confusion.

To repeat: in the case of moist air, the ratio by weight (pounds) of water vapor to dry air is called *humidity ratio* and denoted by the letter W.

Saturation

It is often stated that moist air is *saturated* when the water vapor in it is itself in the *dry saturated condition* at the given temperature. This statement would imply that the humidity ratio of saturated moist air is, in accordance with Equation 11,

$$W_s = 0.62193 \frac{p_s}{P - p_s} \tag{12}$$

where p_s is the saturation pressure of pure water vapor.

This statement lacks thermodynamic soundness due to actual departures from Dalton's Law, but has real practical merit as an approximation.

Example 3. Calculate the humidity ratio of saturated moist air at 68 F, 30 in. Hg

Solution. The saturation pressure of pure water at 68 F from Table 6 is 0.68980 in. Hg; hence,

$$W_s = \frac{0.62193 \times 0.68980}{29.3102} = 0.01464 \text{ (pound per pound of dry air).}$$

It is also frequently stated that moist air is saturated when the space (volume) occupied by it contains the *maximum weight of water vapor* at the given temperature. This means that any additional water would have to be in the liquid or solid phase. But under proper circumstances the water vapor can be *supersaturated*, in which case the space occupied by the mixture can contain more than the *maximum possible* water vapor. The statement is therefore meaningless as a definition of saturation.

A precise definition must necessarily refer to the co-existence of at least two distinct phases, say, liquid and vapor. These can only co-exist in stable equilibrium if evaporation of the liquid or condensation of the vapor under conditions of constant total volume and constant total internal energy would have to involve a decrease of total entropy. This would be the situation if, and only if, the pressure, the temperature, and each component chemical potential has the same value in each phase.

In the case of moist air, the general conditions for saturation previously stated can be deduced from Equation 9 together with available data on the solubility of air in the liquid. They can be reduced to the form,

$$W_s = 0.62193 \frac{p_s'}{P - p_s'} \tag{13a}$$

where

$$p_s' = \frac{(PF)\ (DF)}{(RF)} p_s \tag{13b}$$

The liquid (or solid) phase will contain a small amount of dissolved air and the Raoult factor (RF) expresses the effect of this dissolved air in lowering the vapor pressure in accordance with Raoult's Law. The Poynting factor (PF) accounts for the fact that the very presence of dry air requires the liquid (or solid) to support a higher pressure at saturation than it would if no dry air were present. The Dalton factor (DF) expresses the effect of intermolecular forces in the vapor phase. All three factors depend more or less on pressure as well as on temperature.

The Raoult and Poynting factors are calculable. The order of magnitude of the Dalton factor can now be determined by computing its value at one temperature and pressure using the information previously referred to, namely, $2A_{aw} = 0.075\ (A_{aa} + A_{ww})$. At 68 F, 29.921 in. Hg, for example,

$$p_s' = \frac{1.00073 \times 1.0052}{1.00002} p_s$$

Table 6. Thermodynamic Properties of Moist Air[a], 29.921 In. Hg

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds × 10^5	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure $p_s \times 10^5$: In. of Hg	Lb per Sq In.	Temp Deg F
−60	2 108	0 14756	10 07	0 00	10 07	−14 48	0 02	−14 46	−185 4	101 4	49.808	−60
−59	2 262	.15834	10 09	00	10 09	−14 23	.02	−14.21	−185 0	108.8	53.443	−59
−58	2 418	.16926	10 12	00	10 12	−13 99	.02	−13 97	−184 6	116.3	57.127	−58
−57	2 595	.18165	10 14	.00	10 14	−13 75	.03	−13 72	−184 2	124 8	61 302	−57
−56	2 773	.19411	10 17	00	10 17	−13 50	.03	−13.47	−183 8	133.4	65 526	−56
−55	2 973	0.20811	10 19	0 00	10 19	−13 26	0 03	−13 23	−183 4	143 0	70 242	−55
−54	3 181	.22267	10 22	00	10 22	−13 02	.03	−12 99	−182 9	153 0	75.154	−54
−53	3 399	.23793	10 24	.00	10 24	−12 78	.04	−12 74	−182 5	163.5	80.311	−53
−52	3.636	.25452	10.27	00	10 27	−12 53	.04	−12 49	−182 1	174.9	85.911	−52
−51	3.888	.27216	10.29	.00	10 29	−12 29	04	−12 25	−181 7	187.0	91 854	−51
−50	4.156	0 29092	10 32	0.00	10 32	−12 05	0 04	−12 01	−181.3	199 9	98 191	−50
−49	4.428	.30996	10 34	.00	10.34	−11 81	05	−11 76	−180.9	213.0	104.63	−49
−48	4 738	.33166	10 37	.00	10 37	−11 57	.05	−11 52	−180 4	227 9	111.94	−48
−47	5 054	.35378	10 40	.00	10 40	−11 32	.05	−11 27	−180 0	243.1	119 41	−47
−46	5 395	.37765	10 42	.00	10.42	−11 08	.06	−11 02	−179 6	259 5	127.47	−46
−45	5 753	0 40271	10 45	0 00	10 45	−10 84	0 06	−10.78	−179 2	276 7	135 92	−45
−44	6 133	.42931	10 47	.00	10 47	−10 60	.06	−10 54	−178 7	295 0	144 90	−44
−43	6 543	.45801	10 50	.00	10 50	−10 35	.07	−10 28	−178 3	314.7	154 58	−43
−42	6 971	.48797	10 52	00	10 52	−10 11	.07	−10 04	−177 9	335 3	164 70	−42
−41	7 435	.52045	10 55	.00	10 55	−9 872	.077	−9 795	−177 4	357 6	175.65	−41
−40	7 907	0 55349	10 57	0 00	10 57	−9 629	0 082	−9 547	−177 0	380 3	186 80	−40
−39	8.431	.59017	10 60	.00	10 60	−9 388	.088	−9 300	−176 6	405 5	199.18	−39
−38	8 965	.62755	10 62	00	10 62	−9 146	093	−9 053	−176 1	431 2	211.81	−38
−37	9 548	.66836	10 65	00	10 65	−8 905	.100	−8 805	−175 7	459 2	225 56	−37
−36	10.16	.71120	10 67	.00	10 67	−8 663	.106	−8 557	−174 8	488 4	239.90	−36
−35	10 80	0.75600	10 69	0 00	10 69	−8 422	0 113	−8 309	−174 4	519 5	255 18	−35
−34	11 49	.80430	10.72	00	10 72	−8 180	.120	−8 060	−174 0	552 4	271.34	−34
−33	12 20	.85400	10 75	.00	10 75	−7 939	.127	−7 812	−173 5	586 5	288 09	−33
−32	12 97	.90790	10.77	.00	10 77	−7 698	136	−7 562	−173 1	623 7	306.36	−32
−31	13.76	96320	10 80	.00	10 80	−7.457	.144	−7 313	−172 6	661 8	325 08	−31
−30	14.58	1.0206	10 82	0.00	10.82	−7 216	0 152	−7 064	−172 2	701 0	344.33	−30
−29	15 43	1.0801	10 85	.00	10 85	−6 975	.161	−6 814	−171 7	742 2	364.57	−29
−28	16.45	1.1515	10.87	.00	10 87	−6.734	.172	−6 562	−171 3	791 2	388.64	−28
−27	17.49	1.2243	10 90	.00	10.90	−6 493	.183	−6 310	−170.9	841.0	413.10	−27
−26	18.55	1.2985	10.92	.00	10.92	−6.251	.194	−6.057	−170.4	892 1	438.20	−26

[a]Compiled by W. M Sawdon and extended by John A Goff

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (CONTINUED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air		Volume Cu Ft per Lb Dry Air			Enthalpy Btu per Lb Dry Air			Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure $p_s \times 10^5$		Temp Deg F
	Pounds $\times 10^5$	Grains	Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s		In of Hg	Lb per Sq In.	
−25	19 68	1 3776	10 95	0 00	10 95	−6 011	0 206	−5 805	−170.4	946 4	464.87	−25
−24	20 86	1.4602	10 97	00	10 97	−5 770	219	−5 551	−170 0	1003.	492.67	−24
−23	22 13	1 5491	11 00	00	11 00	−5 529	.232	−5 297	−169 5	1064.	522 64	−23
−22	23.42	1 6394	11 02	.00	11 02	−5 288	.246	−5 042	−169 0	1126.	553.09	−22
−21	24 79	1 7353	11 05	00	11 05	−5 047	260	−4 787	−168 6	1192.	585.51	−21
−20	26.25	1 8375	11 07	0 00	11.07	−4 807	0 276	−4 531	−168 2	1262.0	619.89	−20
−19	27.81	1.9467	11 10	00	11.10	−4.566	.292	−4 274	−167 7	1337.	656 73	−19
−18	29 45	2 0615	11.13	.00	11 13	−4.325	.310	−4 015	−167 2	1416.	695 54	−18
−17	31.12	2.1784	11.15	00	11 15	−4 085	.327	−3 758	−166 8	1496.	734 84	−17
−16	32.95	2 3065	11 18	00	11 18	−3.844	.347	−3 497	−166 3	1584.	778 06	−16
−15	34.84	2 4388	11 20	0 01	11 21	−3 604	0.367	−3 237	−165 9	1675 0	822.76	−15
−14	36 86	2 5802	11 23	01	11 24	−3 363	.388	−2.975	−165 4	1772.	870 41	−14
−13	38 98	2 7286	11.25	01	11 26	−3 123	411	−2 712	−165 0	1874.	920.51	−13
−12	41 19	2 8833	11 28	01	11 29	−2.883	434	−2 449	−164 5	1980.	972 58	−12
−11	43 54	3 0478	11 30	01	11 31	−2 642	.459	−2 183	−164 0	2093.	1028 1	−11
−10	45.98	3 2186	11 33	0 01	11 34	−2 402	0 485	−1 917	−163 6	2210 0	1085 6	−10
−9	48.58	3 4006	11 35	01	11 36	−2.162	.513	−1 649	−163 1	2335.	1147.0	−9
−8	51 25	3 5875	11 38	01	11 39	−1 921	.541	−1.380	−162 7	2463.	1209 8	−8
−7	52.06	3 6442	11.40	01	11 41	−1 681	.570	−1 111	−162 2	2502.	1229 0	−7
−6	57.12	3 9984	11 43	.01	11 44	−1 441	603	−0 838	−161.7	2745.	1348.3	−6
−5	60 30	4 2210	11.45	0 01	11 46	−1.201	0 637	−0 564	−161 3	2898 0	1423.5	−5
−4	63.57	4 4499	11 48	.01	11 49	−0 960	672	−0.288	−160 8	3055	1500 6	−4
−3	67 05	4 6935	11 50	.01	11 51	−0 720	.709	−0 011	−160 3	3222.	1582 6	−3
−2	70 69	4 9483	11 53	01	11 54	−0 480	.748	+0 268	−159 9	3397.	1668 6	−2
−1	74.50	5.2150	11 55	01	11 57	−0 240	789	+0 549	−159 4	3580	1758 5	−1
0	78 52	5 5000	11 58	0 01	11 59	0	0 832	+0 832	−158 9	3773 0	1853 3	0

[a] Compiled by W. M. Sawdon and extended by John A Goff

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (CONTINUED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure p_s: In of Hg	Lb per Sq In	Temp Deg F
0	0.0007852	5.50	11.58	0.01	11.59	0.000	0.832	0.832	−158.9	0.03773	0.01853	**0**
1	.0008275	5.79	11.60	.02	11.62	.240	.877	1.117	−158.5	.03975	.01963	1
2	.0008714	6.10	11.63	.01	11.64	.480	.924	1.404	−158.0	.04186	.02056	2
3	.0009179	6.43	11.65	.02	11.67	.720	.974	1.694	−157.5	.04409	.02166	3
4	.0009671	6.77	11.68	.02	11.70	.960	1.026	1.986	−157.0	.04645	.02282	4
5	0.001017	7.12	11.70	0.02	11.72	1.200	1.080	2.280	−156.6	0.04886	0.02400	**5**
6	.001071	7.50	11.73	.02	11.75	1.440	1.137	2.577	−156.1	.05144	.02527	6
7	.001127	7.89	11.75	.02	11.77	1.680	1.197	2.877	−155.6	.05412	.02658	7
8	.001186	8.30	11.78	.02	11.80	1.920	1.260	3.180	−155.1	.05692	.02796	8
9	.001247	8.73	11.80	.03	11.83	2.160	1.336	3.486	−154.7	.05988	.02941	9
10	0.001311	9.18	11.83	0.02	11.85	2.400	1.395	3.795	−154.2	0.06295	0.03092	**10**
11	.001379	9.65	11.86	.02	11.88	2.640	1.468	4.108	−153.7	.06618	.03251	11
12	.001450	10.15	11.88	.03	11.91	2.880	1.544	4.424	−153.2	.06958	.03418	12
13	.001523	10.66	11.91	.02	11.93	3.120	1.622	4.742	−152.7	.07309	.03590	13
14	.001600	11.20	11.93	.03	11.96	3.359	1.705	5.064	−152.2	.07677	.03771	14
15	0.001682	11.77	11.96	0.03	11.99	3.599	1.793	5.392	−151.8	0.08067	0.03963	**15**
16	.001766	12.36	11.98	.03	12.01	3.839	1.883	5.722	−151.3	.08469	.04160	16
17	.001855	12.99	12.00	.04	12.04	4.079	1.979	6.058	−150.8	.08895	.04369	17
18	.001947	13.63	12.03	.04	12.07	4.319	2.078	6.397	−150.3	.09337	.04586	18
19	.002043	14.30	12.06	.03	12.09	4.559	2.182	6.741	−149.8	.09797	.04812	19
20	0.002144	15.01	12.08	0.04	12.12	4.798	2.290	7.088	−149.3	0.1028	0.05050	**20**
21	.002250	15.75	12.11	.04	12.15	5.038	2.405	7.443	−148.8	.1078	.05295	21
22	.002361	16.53	12.13	.05	12.18	5.278	2.524	7.802	−148.3	.1132	.05560	22
23	.002476	17.33	12.16	.04	12.20	5.518	2.648	8.166	−147.8	.1186	.05826	23
24	.002596	18.17	12.18	.05	12.23	5.758	2.778	8.536	−147.3	.1244	.06111	24
25	0.002722	19.05	12.21	0.05	12.26	5.998	2.914	8.912	−146.8	0.1304	0.06405	**25**
26	.002853	19.97	12.23	.06	12.29	6.237	3.055	9.292	−146.4	.1366	.06710	26
27	.002991	20.94	12.26	.06	12.32	6.477	3.205	9.682	−145.9	.1432	.07034	27
28	.003133	21.93	12.28	.06	12.34	6.717	3.358	10.075	−145.4	.1500	.07368	28
29	.003283	22.99	12.31	.06	12.37	6.957	3.520	10.477	−144.9	.1571	.07717	29
30	0.003439	24.07	12.33	0.07	12.40	7.197	3.689	10.886	−144.4	0.1645	0.08080	**30**
31	.003601	25.21	12.36	.07	12.43	7.437	3.865	11.302	−143.9	.1722	.08458	31
32	.003771	26.40	12.38	.08	12.46	7.677	4.049	11.726	−143.4	.1803	.08856	32
33	.003931	27.52	12.41	.08	12.49	7.917	4.222	12.139	+ 1.0	.1879	.09230	33
34	.004094	28.66	12.43	.08	12.51	8.157	4.399	12.556	+ 2.0	.1957	.09610	34

[a]Compiled by W M Sawdon and extended by John A Goff.

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29 921 IN HG (CONTINUED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Liquid Water Btu per Lb h'_w	Saturation Pressure p_s: In of Hg	Lb per Sq In	Temp Deg F
35	0 004262	29 83	12 46	0 08	12 54	8 397	4 582	12 979	3 0	0 20360	0 1000	35
36	.004438	31 07	12 48	09	12 57	8 636	4 773	13 409	4 0	21195	.1041	36
37	.004618	32 33	12 51	09	12 60	8 876	4 969	13 845	5 0	.22050	1083	37
38	004803	33 62	12 53	.10	12 63	9 116	5 169	14 285	6 0	22925	.1126	38
39	.004996	34.97	12 56	10	12 66	9 356	5 380	14 736	7 0	.23842	.1171	39
40	0 005194	36 36	12 59	0 10	12 69	9 596	5 595	15 191	8 0	0 24778	0 1217	40
41	.005401	37 80	12 61	11	12 72	9 836	5 821	15 657	9 1	.25755	.1265	41
42	005616	39 31	12 64	11	12 75	10 08	6 05	16 13	10 1	.26773	.1315	42
43	005840	40 88	12 66	.12	12 78	10 32	6 30	16 62	11 1	.27832	1367	43
44	006069	42 48	12 69	12	12 81	10 56	6 55	17 11	12 1	.28911	.1420	44
45	0 006306	44.14	12 71	0 13	12 84	10 80	6 81	17 61	13 1	0 30031	0 1475	45
46	006553	45 87	12 74	13	12 87	11 04	7 08	18 12	14 1	31191	.1532	46
47	006808	47 66	12 76	14	12 90	11 28	7 36	18 64	15 1	32393	.1591	47
48	007072	49 50	12 79	14	12 93	11 52	7 64	19 16	16 1	.33635	.1652	48
49	.007345	51 42	12 81	15	12 96	11 76	7 94	19 70	17 1	.34917	1715	49
50	0 007626	53 38	12 84	0 15	12 99	12 00	8 25	20 25	18 1	0 36241	0 1780	50
51	007921	55 45	12 86	16	13 02	12 23	8 57	20 80	19 1	37625	.1848	51
52	008226	57 58	12 89	17	13 06	12 47	8 91	21 38	20 1	.39051	1918	52
53	008534	59 74	12 91	18	13 09	12 71	9 24	21 95	21 1	.40496	1989	53
54	008856	61 99	12 94	18	13 12	12 95	9 60	22 55	22 1	.42003	.2063	54
55	0 009192	64 34	12 96	0 19	13 15	13 19	9 96	23 15	23 1	0 43570	0 2140	55
56	.009536	66 75	12 99	20	13 19	13 43	10 34	23 77	24 1	45179	2219	56
57	.009890	69 23	13 01	21	13 22	13 67	10 73	24 40	25 1	.46828	.2300	57
58	.01026	71 82	13 04	21	13 25	13 91	11 14	25.05	26 1	48538	.2384	58
59	01064	74 48	13 06	23	13 29	14 15	11 55	25 70	27 1	.50310	.2471	59
60	0 01103	77 21	13 09	0 23	13 32	14 39	11 98	26 37	28 1	0 52142	0 2561	60
61	.01144	80 08	13 11	.24	13 35	14.63	12 43	27 06	29 1	54035	.2654	61
62	01186	83 02	13 14	.25	13 39	14 87	12 89	27 76	30 1	55970	.2749	62
63	01229	86 03	13 16	26	13 42	15 11	13 37	28 48	31 1	.57985	.2848	63
64	.01274	89 18	13 19	.27	13 46	15 35	13 86	29 21	32 1	60042	.2949	64
65	0 01320	92 40	13 21	0 28	13 49	15 59	14 37	29 96	33 1	0 62179	0 3054	65
66	.01368	95 76	13 24	29	13 53	15 83	14 90	30 73	34 1	.64378	.3162	66
67	.01417	99 19	13 26	31	13 57	16 07	15 44	31 51	35 1	66638	.3273	67
68	.01468	102 8	13 29	31	13 60	16 31	16 00	32 31	36 0	.68980	.3388	68
69	.01520	106 4	13 31	33	13.64	16 55	16 57	33 12	37 0	.71382	3506	69

[a]Compiled by W. M Sawdon and extended by John A. Goff.

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (CONTINUED)

TEMP DEG F	SATURATION HUMIDITY RATIO W_s WEIGHT OF WATER PER LB OF DRY AIR		VOLUME CU FT PER LB DRY AIR			ENTHALPY BTU PER LB DRY AIR			SPECIFIC ENTHALPY OF LIQUID WATER BTU PER LB h'_w	SATURATION PRESSURE p_s		TEMP DEG F
	Pounds	Grains	Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s		In. of Hg	Lb per Sq In.	
70	0 01574	110 2	13.34	0 34	13 68	16 79	17.17	33.96	38.0	0.73866	0 3628	**70**
71	.01631	114.2	13.37	.34	13 71	17.03	17.80	34.83	39 0	.76431	.3754	71
72	.01688	118.2	13 40	35	13 75	17 27	18.43	35.70	40.0	.79058	.3883	72
73	.01748	122 4	13.42	37	13 79	17.51	19 09	36.60	41 0	.81766	.4016	73
74	.01809	126 6	13.44	39	13 83	17.75	19 76	37.51	42.0	.84555	.4153	74
75	0.01873	131.1	13 47	0 40	13 87	17.99	20 47	38.46	43 0	0 87448	0 4295	**75**
76	.01938	135.7	13 49	.42	13 91	18 23	21.19	39 42	44 0	.90398	.4440	76
77	.02005	140 4	13 52	.43	13 95	18 47	21 93	40 40	45 0	.93452	.4590	77
78	.02075	145 3	13 54	45	13 99	18 71	22 71	41 42	46 0	.96588	.4744	78
79	.02147	150 3	13.57	46	14 03	18 95	23 51	42 46	47 0	99825	.4903	79
80	0 02221	155.5	13 59	0 49	14 08	19 19	24 32	43 51	48 0	1 0316	0 5067	**80**
81	.02298	160.9	13 62	.50	14 12	19 43	25 18	44 61	49 0	1 0661	.5236	81
82	.02377	166 4	13 64	52	14 16	19 67	26 05	45 72	50 0	1 1013	5409	82
83	.02459	172 1	13 67	54	14 21	19 91	26 97	46 88	51 0	1 1377	.5588	83
84	.02543	178 0	13 69	57	14 26	20 15	27 90	48 05	52 0	1.1752	.5772	84
85	0 02629	184 0	13 72	0 58	14 30	20 39	28 85	49 24	53 0	1.2135	0 5960	**85**
86	.02718	190 3	13 74	60	14 34	20 63	29.84	50 47	54 0	1.2527	.6153	86
87	.02810	196 7	13 77	62	14 39	20 87	30 87	51 74	55 0	1 2933	.6352	87
88	.02904	203 3	13 79	65	14 44	21 11	31 91	53 02	56 0	1 3346	.6555	88
89	.03002	210.1	13 82	66	14 48	21 35	33 00	54 35	57 0	1.3774	.6765	89
90	0 03102	217 1	13.84	0 69	14 53	21 59	34 11	55 70	58 0	1 4211	0 6980	**90**
91	.03205	224 4	13 87	.71	14 58	21 83	35 26	57 09	59 0	1 4661	.7201	91
92	.03312	231 8	13 89	74	14 63	22 07	36 45	58 52	60 0	1 5125	.7429	92
93	.03421	239.5	13 92	77	14 69	22 32	37 67	59 99	61 0	1 5600	.7662	93
94	03535	247 5	13 94	.79	14.73	22 56	38 94	61 50	62 0	1.6088	.7902	94
95	0 03652	255 6	13.97	0 82	14 79	22 80	40 25	63 05	63 0	1 6591	0 8149	**95**
96	.03772	264 0	13 99	85	14 84	23 04	41 58	64 62	64 0	1 7108	.8403	96
97	03896	272 7	14 02	88	14 90	23 28	42 97	66 25	65 0	1.7638	.8663	97
98	04024	281 7	14 04	91	14 95	23 52	44 40	67 92	66 0	1 8181	.8930	98
99	.04156	290 9	14 07	94	15 01	23 76	45 87	69 63	67 0	1 8741	.9205	99
100	0 04293	300 5	14.10	0 97	15 07	24 00	47 40	71 40	68 0	1 9316	0 9487	**100**
101	.04433	310 3	14 12	1 00	15 12	24 24	48 97	73 21	69 0	1.9904	9776	101
102	.04577	320.4	14 15	1 03	15 18	24 48	50 58	75 06	70 0	2 0507	1 0072	102
103	.04726	330 8	14.17	1 08	15 25	24 72	52 25	76 97	71 0	2 1128	1 0377	103
104	.04879	341 5	14 20	1 11	15 31	24 96	53 96	78 92	72 0	2.1763	1 0689	104

[a]Compiled by W. M Sawdon and extended by John A. Goff.

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (CONTINUED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Liquid Water Btu per Lb h'_w	Saturation Pressure p_s: In of Hg	Lb per Sq In.	Temp Deg F
105	0 05037	352.6	14 22	1.15	15 37	25 20	55 73	80 93	73 0	2.2414	1.1009	**105**
106	.05200	364.0	14 25	1 19	15 44	25 44	57 56	83 00	74 0	2.3084	1.1338	106
107	.05368	375 8	14 27	1.23	15.50	25 68	59 45	85 13	75 0	2.3770	1.1675	107
108	.05541	387.9	14 30	1.27	15 57	25 92	61 38	87 30	76 0	2 4473	1.2020	108
109	.05719	400.3	14 32	1.32	15.64	26 16	63 38	89 54	76.9	2 5196	1.2375	109
110	0 05904	413 3	14 35	1.36	15 71	26 40	65 46	91 86	77.9	2 5939	1 274	**110**
111	06092	426 4	14 37	1 41	15 78	26 64	67 57	94 21	78 9	2 6692	1 311	111
112	.06292	440 4	14 39	1.46	15 85	26 88	69 82	96 70	79 9	2 7486	1 350	112
113	.06493	454 5	14 42	1 51	15 93	27 12	72 08	99 20	80 9	2 8280	1 389	113
114	.06700	469 0	14 45	1.55	16 00	27 36	74 40	101 76	81.9	2 9094	1.429	114
115	0 06913	483 9	14 47	1 61	16 08	27 60	76 80	104 40	82 9	2.9929	1.470	**115**
116	07134	499 4	14 50	1 66	16.16	27 84	79 29	107 13	83 9	3 0784	1 512	116
117	07361	515 3	14 52	1 72	16 24	28 08	81 84	109 92	84 9	3 1660	1.555	117
118	.07600	532 0	14 55	1 77	16 32	28 32	84 53	112 85	85 9	3 2576	1 600	118
119	07840	548 8	14 57	1 84	16 41	28 56	87 24	115 80	86 9	3 3492	1 645	119
120	0 08093	566 5	14 60	1 90	16 50	28 80	90 09	118 89	87.9	3 4449	1.692	**120**
121	.08348	584 4	14 62	1 96	16 58	29 04	92 97	122 01	88 9	3 5406	1.739	121
122	.08616	603 1	14 65	2 03	16 68	29 28	95 99	125 27	89 9	3 6404	1 788	122
123	.08892	622 4	14 67	2 10	16 77	29 52	99 11	128 63	90 9	3 7422	1 838	123
124	.09175	642 3	14 70	2 17	16 87	29 76	102 30	132 06	91.9	3 8460	1 889	124
125	0 09466	662 6	14 72	2 24	16 96	30 00	105 59	135 59	92 9	3 9519	1 941	**125**
126	.09770	683 9	14 75	2 31	17 06	30 24	109 02	139.26	93.9	4 0618	1 995	126
127	.1008	705 6	14 77	2 40	17 17	30 48	112 53	143 01	94.9	4 1718	2 049	127
128	.1040	728 0	14 80	2 47	17 27	30 72	116 15	146 87	95.9	4 2858	2.105	128
129	.1074	751 8	14 83	2 55	17 38	30 96	120 00	150 96	96.9	4.4039	2.163	129
130	0.1107	774 9	14 85	2 64	17.49	31 20	123 73	154.93	97 9	4 5220	2 221	**130**
131	.1143	800 1	14 88	2 73	17 61	31 45	128 81	159 26	98 9	4 6441	2.281	131
132	.1180	826.0	14 90	2 83	17.73	31 69	131 99	163 68	99.9	4.7703	2.343	132
133	.1218	852 6	14 93	2 92	17 85	31.93	136 31	168 24	100 9	4 8986	2.406	133
134	.1257	879.9	14 95	3.02	17.97	32 17	140 72	172.89	101.9	5.0289	2.470	134
135	0.1297	907.9	14 98	3 12	18 10	32 41	145 26	177 67	102.9	5 1633	2.536	**135**
136	.1339	937.3	15 00	3 23	18 23	32 65	150 02	182 67	103 9	5 2997	2.603	136
137	.1382	967 4	15 03	3 33	18 36	32 89	154.91	187.80	104.9	5.4402	2 672	137
138	.1427	998 9	15 05	3 45	18 50	33 13	160.01	193 14	105 9	5.5827	2.742	138
139	.1473	1031.1	15 08	3 57	18 65	33 37	165 24	198 61	106.9	5.7293	2.814	139

[a]Compiled by W. M. Sawdon and extended by John A. Goff.

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (CONTINUED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air		Volume Cu Ft per Lb Dry Air			Enthalpy Btu per Lb Dry Air			Specific Enthalpy of Liquid Water Btu per Lb h'_w	Saturation Pressure p_s		Temp Deg F
	Pounds	Grains	Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s		In of Hg	Lb per Sq In	
140	0 1521	1064.7	15 10	3 69	18 79	33 61	170 69	204 30	107 9	5 8779	2 887	**140**
141	1570	1099 0	15 13	3 81	18 94	33 85	176 26	210 11	108 9	6.0306	2.962	141
142	1622	1135 4	15 15	3 95	19 10	34 09	182 17	216.26	109 9	6 1874	3 039	142
143	.1675	1172 5	15 18	4 08	19 26	34 33	188 20	222 53	110 9	6 3482	3 118	143
144	.1730	1211 0	15 20	4 23	19 43	34 57	194 45	229 02	111 9	6 5111	3 198	144
145	0 1787	1250 9	15 23	4 37	19 60	34 81	200 95	235 76	112 9	6 6781	3 280	**145**
146	1846	1292 2	15 25	4 53	19 78	35 05	207 66	242 71	113 9	6 8471	3 363	146
147	.1908	1335 6	15 28	4 68	19 96	35 29	214 73	250 02	114 9	7.0222	3 449	147
148	1971	1379 7	15 30	4 85	20 15	35 53	221 90	257 43	115 9	7 1993	3 536	148
149	2037	1425 9	15 33	5 02	20 35	35 77	229 43	265 20	116 9	7 3805	3 625	149
150	0 2105	1473 5	15 35	5 20	20 55	36 02	237 17	273 19	117 9	7 5658	3 716	**150**
151	2176	1523 2	15 38	5 38	20 76	36 26	245 28	281 54	118 9	7.7551	3 809	151
152	.2250	1575 0	15 40	5 57	20 97	36 50	253 71	290 21	119 9	7 9485	3 904	152
153	2327	1628 9	15 43	5 77	21 20	36 74	262 51	299 25	120 9	8 1460	4 001	153
154	2407	1684 9	15 45	5 98	21 43	36 98	271 63	308 61	121 9	8 3476	4 100	154
155	0 2490	1743 0	15 48	6 19	21 67	37 22	281 12	318 34	122 9	8 5532	4 201	**155**
156	2577	1803 9	15 50	6 43	21.93	37.46	291 05	328 51	123 9	8 7650	4 305	156
157	.2667	1866 9	15 53	6 66	22 19	37 70	301 24	339 04	124 9	8 9788	4 410	157
158	.2761	1932 7	15 56	6 90	22.46	37 94	312 08	350 02	125 9	9 1986	4 518	158
159	.2858	2000 6	15 58	7 16	22 74	38 18	323 18	361 36	126 9	9 4206	4 627	159
160	0.2961	2072 7	15 61	7 42	23 03	38 43	334 95	373 38	127 9	9 6486	4 739	**160**
161	3067	2146 9	15 63	7 70	23 33	38 67	347 09	385 76	128 9	9 8807	4 853	161
162	3179	2225 3	15 66	7 99	23 65	38 91	359 89	398 80	129 9	10 119	4 970	162
163	3295	2306 5	15 68	8 30	23 98	39 15	373 19	412 34	130 9	10 361	5 089	163
164	3416	2391 2	15 71	8 62	24 33	39 39	387 03	426 42	131 9	10 608	5 210	164
165	0 3544	2480 8	15 73	8 96	24 69	39 63	401 71	441 34	132 9	10 860	5 334	**165**
166	3677	2573 9	15 76	9 31	25 07	39 87	416 94	456 81	133 9	11 117	5 460	166
167	3817	2671 9	15 78	9 68	25 46	40 11	433 00	473 11	134 9	11 379	5 589	167
168	3964	2774 8	15 81	10 07	25 88	40 35	449 83	490 18	135 9	11 646	5 720	168
169	4118	2882 6	15 83	10 48	26 31	40 59	467 52	508 11	136 9	11 919	5 854	169
170	0 4280	2996 0	15 86	10 91	26 77	40 83	486 08	526 91	137 9	12 196	5 990	**170**
171	4451	3115 7	15 88	11 36	27 24	41 07	505 72	546 79	138 9	12 480	6 130	171
172	4631	3241 7	15 91	11 83	27 74	41 32	526 36	567 68	139 9	12 770	6 272	172
173	4821	3374 7	15 93	12 35	28 28	41 56	548 20	589 76	140 9	13 065	6 417	173
174	5022	3515 4	15 96	12 88	28 84	41 80	571 25	613 05	141 9	13 366	6 565	174

[a]Compiled by W. M Sawdon and extended by John A. Goff

TABLE 6. THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29 921 IN. HG (CONCLUDED)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air		Volume Cu Ft per Lb Dry Air			Enthalpy Btu per Lb Dry Air			Specific Enthalpy of Liquid Water Btu per Lb h'_w	Saturation Pressure p_s		Temp Deg F
	Pounds	Grains	Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s		In of Hg	Lb per Sq In.	
175	0 5235	3664 5	15 98	13 45	29 43	42 04	595 74	637 78	142 9	13 674	6.716	**175**
176	5459	3821 3	16 01	14 04	30 05	42 28	621 45	663 73	143 9	13 985	6 869	**176**
177	5697	3987 9	16 03	14 68	30 71	42 52	648 83	691 35	144 9	14 303	7 025	**177**
178	5949	4164 3	16 06	15 35	31 41	42 76	677 77	720 53	145 9	14 627	7.184	**178**
179	6215	4350 5	16 08	16 07	32 15	43 00	708 39	751 39	146 9	14 954	7 345	**179**
180	0 6501	4550 7	16 11	16 83	32 94	43 24	741 24	784 48	147 9	15 290	7.510	**180**
181	6805	4763 5	16 13	17 65	33 78	43 49	776 25	819 74	148 9	15 632	7.678	**181**
182	7131	4991 7	16 16	18 52	34 68	43 73	813 72	857 45	149 9	15 981	7.849	**182**
183	7481	5236 7	16 18	19 47	35 65	43 97	854 03	898 00	150 9	16 337	8 024	**183**
184	7854	5497 8	16 21	20 46	36 67	44 21	896 93	941 14	151 9	16 697	8 201	**184**
185	0 8258	5780 6	16 23	21 55	37 78	44 45	943 48	987 93	152 9	17 066	8 382	**185**
186	8693	6085 1	16 26	22 72	38 98	44 69	993 52	1038 21	153 9	17 440	8 566	**186**
187	.9162	6413 4	16 28	23 99	40 27	44 93	1047 58	1092 51	154 9	17 821	8.753	**187**
188	9673	6771 1	16 31	25 36	41 67	45 18	1106 40	1151 58	155 9	18 210	8.944	**188**
189	1 0227	7158 9	16 34	26 70	43 04	45 42	1170 62	1216 04	157 0	18 605	9.138	**189**
190	1 083	7581 0	16 36	28 49	44 85	45 66	1239 71	1285 37	158 0	19 008	9.336	**190**
191	1 150	8050 0	16 39	30 29	46 68	45 90	1316 98	1362 88	159 0	19 419	9.538	**191**
192	1 224	8568 0	16 41	32 29	48 70	46 14	1402 21	1448 35	160 0	19 839	9.744	**192**
193	1 306	9142 0	16 44	34 49	50 93	46 38	1496 81	1543 19	161 0	20 266	9.954	**193**
194	1 397	9779 0	16 46	36 96	53 42	46 62	1601 66	1648 28	162 0	20 702	10.168	**194**
195	1.499	10493 0	16 49	39 71	56 20	46 86	1719 35	1766 21	163 0	21.144	10.385	**195**
196	1.613	11291 0	16 51	42 80	59 31	47 10	1850 76	1897 86	164 0	21.592	10.605	**196**
197	1 742	12194 0	16 54	46 31	62 85	47 34	1999 64	2046 98	165 0	22 048	10.829	**197**
198	1 890	13230 0	16 56	50 32	66 88	47 59	2170 29	2217.88	166 0	22 512	11.057	**198**
199	2 061	14427 0	16 59	54 95	71 54	47 83	2367.68	2415 51	167 0	22 984	11.289	**199**
200	2 261	15827 0	16 61	60 38	76 99	48 07	2598 34	2646.41	168 0	23.465	11.525	**200**

[a]Compiled by W M Sawdon and extended by John A Goff.

This indicates departures from Dalton's Law of the order of 0.5 per cent. The data in Table 6 which are based on an assumed value of unity for the Dalton factor have not been revised pending final results on the measurement of the interaction constant [10].

Relative Humidity

The ratio of actual humidity ratio W to the saturation humidity ratio W_s corresponding to the actual temperature and the observed pressure is denoted by the symbol μ and may be called alternatively *degree of saturation* or *per cent saturation;* thus,

$$W = \mu W_s \tag{14}$$

Example 4. Air is to be maintained at 70 F, 40 per cent saturation when outside air is at 0 F, 70 per cent. The observed pressure may be taken to be 29.921 in. Hg. Find the weight of water to be added to each pound of dry air using Table 6.

Solution. The desired humidity ratio is $0.40 \times 0.01574 = 0.006296$ while that of outside air is $0.70 \times 0.0007852 = 0.000550$. Hence the weight of water to be added is $0.006296 - 0.000550 = 0.005746$ lb per pound dry air.

Under Dalton's Law the water vapor exerts a partial pressure p_w which may be calculated from the given humidity ratio W and the observed pressure P by means of Equation 11. The ratio of this partial pressure p_w to the saturation pressure of pure water p_s corresponding to the actual temperature is called *relative humidity* and may be denoted by the symbol Φ; thus,

$$\Phi = \frac{p_w}{p_s} \tag{15}$$

The relation between μ and Φ is obtained directly from Equations 11 and 12 and is

$$\mu = \left(\frac{P - p_s}{P - p_w}\right) \Phi \tag{15a}$$

whence it is clear that for ordinary temperatures where p_s and therefore p_w are small compared with P, the two are approximately equal.

As an aid in quickly translating degree of saturation μ into relative humidity Φ, the following empirical equation may be substituted for Equation 15a:

$$\Phi = \mu + C\mu\,(1 - \mu) \tag{15b}$$

where C depends upon temperature *for standard atmospheric pressure,* as shown by the values in Table 7.

Within the limits of accuracy of (15b) this may also be written

$$\mu = \Phi - C\,\Phi\,(1 - \Phi) \tag{15c}$$

and used to translate relative humidity Φ into degree of saturation μ.

For example, corresponding to 40 per cent saturation at 100 F, the relative humidity is $0.40 + 0.0667 \times 0.40 \times 0.60 = 0.416$ or 41.6 per

TABLE 7. VALUES OF THE CONSTANT C IN EQUATIONS 15b AND 15c

TEMP F t	PER CENT C	TEMP F t	PER CENT C	TEMP F t	PER CENT C	TEMP F t	PER CENT C
5	0 16	30	0.55	55	1.47	80	3.51
10	0.21	35	0.68	60	1.76	85	4.14
15	0.27	40	0.83	65	2.10	90	4.86
20	0.34	45	1.01	70	2.50	95	5.70
25	0.44	50	1.22	75	2.97	100	6 67

cent (15b). Conversely, corresponding to a relative humidity of 41.6 per cent, the degree of saturation is $0.416 - 0.0667 \times 0.416 \times 0.584 = 0.400$ or 40 per cent (15c).

Dew-Point

If moist air is cooled at constant humidity ratio W and constant observed pressure P, a temperature will be reached at which the air just becomes saturated and formation of a liquid (or solid) phase just commences. This temperature is called the *dew-point* corresponding to the given humidity ratio and observed pressure.

Example 5. Find the dew-point of the humidified air of Example 4.

Solution. The given humidity ratio is 0.006296 which is the saturation value at 44.96 F (Table 6, assuming the total pressure to be 29 921 in. Hg). This is therefore the dew-point of the humidified air.

Example 6. Find the degree of saturation of air having a temperature of 90 F, a dew-point of 60 F.

Solution. Assuming the total pressure to be 29.921 in. Hg, the humidity ratio is given in Table 6 as 0 01103 lb per pound dry air. The saturation humidity ratio at 90 F is 0.03102 lb per pound dry air; hence the degree of saturation is $0.01103 \div 0.03102 = 0.355$ or 35 5 per cent.

Volume

The volume of moist air *per pound of dry air* contained in it is a very useful quantity. It should not be called specific volume; for the adjective *specific* should properly refer to volume per pound of *mixture.* Using Equations 10a and 14 an expression for the volume per pound of dry air is obtained, namely,

$$v = \frac{B_a T}{P} + \mu \left(\frac{W_s B_w T}{P}\right) \tag{16}$$

Example 7. Find the volume (per pound of dry air) of the humidified air of Example 4.

Solution. $v = \left(\frac{53.35 \times 529\ 7}{29.92 \times 0.49115 \times 144}\right) + 0.40 \left(\frac{0.01574 \times 85.78 \times 529.7}{29.92 \times 0.49115 \times 144}\right)$
$= 13.354 + 0.40 \times 0.338 = 13.489$ cu ft per pound dry air.

Equation 16 is linear in degree of saturation μ and of the form

$$v = v_a + \mu\, v_{as} \tag{17}$$

where v_a denotes specific volume of dry air at temperature T and pressure P; and v_{as} denotes the difference between this and the volume of the saturated mixture per pound of dry air v_s. Strict linearity is, of course, a result of the use of Dalton's Law; but it is expected that it can be retained as a very close approximation even when the abandonment of Dalton's Law becomes possible.

Example 8. Work Example 7 using Table 6.

Solution. $v = 13.34 + (0.40 \times 0.34) = 13.48$ cu ft per pound dry air.

By putting $\mu = 1$ (100 per cent saturation) in Equation 16 an expression for v_s, the volume of saturated air per pound of dry air, is obtained. Values for standard atmospheric pressure (29.921 in. Hg) are listed in Table 6. Often it is preferred to express this information in terms of *density,* that is, weight of saturated air per unit volume. This can easily be done by dividing v_s (volume of saturated air per pound of dry air) *into* $(1 + w_s)$ (*weight* of saturated air per pound of dry air). Thus, at 100 F, 29.921 in. Hg, the density of saturated air is, from Table 6, $1.04293 \div 15.07 = 0.06921$ lb per cubic foot.

Values in Table 9 are intended to aid in determining the density of saturated air at different pressures. Values for temperatures and pressures other than those listed can be obtained by linear interpolation which is aided by the next to last column of figures. Thus, at 100 F, 29.921 in. Hg, the density of saturated air is, from Table 9, 0.06818 + (4.21 × 0.00024) = 0.06919 lb per cubic foot, in approximate agreement with Table 6.

A column of figures is included in Table 9 giving the approximate average increase in density per degree wet-bulb depression. This makes it easy to calculate a value for the density of moist air taking into account its moisture content as well as its temperature and pressure.

Volume Chart

A volume chart drawn for a total pressure of 29.921 in. Hg will be found in the envelope attached to the inside back cover of this book. On this chart values of volume per pound of dry air v are plotted as abscissa against values of humidity ratio W as ordinate. The chart is self-explanatory.

Enthalpy

Thermodynamically, Equation 10a implies that the specific enthalpies of dry air and water vapor are independent of pressure and that the enthalpy of moist air (per pound of dry air) is the sum of separate contributions from the dry air and water vapor according to the simple equation

$$h = h_a + \mu\,(W_s h_w) \qquad (18)$$

Equation 18 is also linear in degree of saturation μ and of the form

$$h = h_a + \mu\, h_{as} \qquad (19)$$

where h_a denotes the specific enthalpy of dry air at the given temperature and total pressure; and h_{as} denotes the difference between this and the enthalpy of the saturated mixture per pound of dry air h_s. Provisional values are listed in Table 6.

Example 9. Find the enthalpy (per pound of dry air) of air at 96 F, 60 per cent saturation and 29.921 in. Hg.

Solution. Using Table 6, h = 23.04 + (0 60 × 41.58) = 47.99 Btu per pound dry air.

Thermodynamic Wet-bulb Temperature

If liquid (or solid) water be injected into an air stream it will evaporate and thus increase the humidity ratio of the air. Enough water may be injected to saturate the air. If the process is one of steady flow with *observed pressure constant*; if it is *adiabatic*; and if the temperature at which the air reaches saturation *coincides* with the temperature of the liquid (or solid) as added; then the common temperature is called *thermodynamic wet-bulb temperature*. This lengthy definition is easily visualized by referring to Fig. 1 in which h'_w denotes the specific enthalpy of the liquid (or solid) as injected.

The process being adiabatic, weight and energy accountings give

$$h_1 + (W_s - W_1)\, h'_w = h_s \qquad (20)$$

If the temperature of the saturated air at the leaving section *coincides* with that of the injected liquid (or solid), then W_s, h'_w and h_s are functions of a single temperature t' which can therefore be determined by solving

(20). This is the thermodynamic wet-bulb temperature corresponding to conditions at the entering section.

Example 10. Find the thermodynamic wet-bulb temperature of dry air at 80 F and 29.921 in. Hg.

Solution. Using Table 6, the equation to be solved is $19\ 19 + (W_s - 0)\ h'_w = h_s$.

A trial value is obtained by ignoring the small quantity $(W_s - 0)\ h'_w$; it is 48 F corresponding to $h_s = 19.19$ Btu per pound dry air. A final value of 48.26 F is then obtained from $h_s = 19.19 + (0.007072 \times 16\ 1) = 19.30$ Btu per pound dry air.

Example 11 Find the degree of saturation of moist air at 90 F dry-bulb, 70 F wet-bulb and 29 921 in. Hg

Solution. Using Table 6, the equation to be solved is $(21.59 + 34.11\ \mu) + (0.01574 - 0.03102\mu) \times 38\ 0 = 33.96$ from which

$$\mu = 11.77 \div 32.93 = 0\ 357 \text{ or } 35.7 \text{ per cent.}$$

It is important to note in connection with Equation 20 that the enthalpy per pound dry air is not constant along a line of constant thermodynamic wet-bulb temperature on account of the term $(W_s - W_1)\ h'_w$. In rough calculations, however, it is usually legitimate to ignore this term.

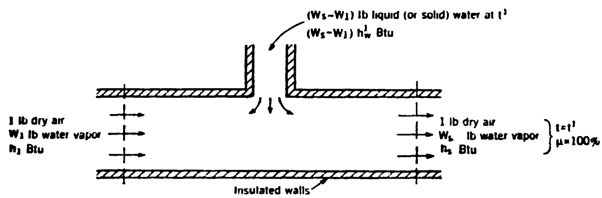

FIG 1. DIAGRAM ILLUSTRATING THERMODYNAMIC WET-BULB TEMPERATURE

Thermodynamic wet-bulb is an important property of moist air because it is approximately the temperature indicated by the wet-bulb psychrometer. This instrument consists of a thermometer with its bulb covered with gauze moistened with clean liquid water. It is whirled through the air until the thermometer reads a steady temperature. At this point, the temperature of the liquid evaporating from the wetted surface has adjusted itself so that the air immediately in contact with the liquid is brought to saturation at the same temperature. Unfortunately, the mixing taking place beyond the liquid surface is not adiabatic; for one reason because the wet-bulb *sees* objects at dry-bulb temperature and considerable heat is transferred by radiation. Also there are other reasons why the readings of the psychrometer depend upon the design of the instrument, the velocity of the air stream in which it is placed, and other factors. Therefore wet-bulb temperature as indicated by the psychrometer cannot be regarded as a thermodynamic property; in fact, the approximate agreement with thermodynamic wet-bulb temperature in the case of moist air has been shown to be largely fortuitous [4].

Mollier Diagram

A thermodynamic analysis of any air conditioning process consists in writing: (1) a weight balance for the dry air; (2) a weight balance for the water; (3) an energy balance. The first is reduced to its simplest form by basing all quantities on one pound of dry air. The second is the most simply expressed in terms of humidity ratio, or weight of water per

TABLE 8. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE[a]

ABS. PRESS. IN. HG p	TEMP F t	SPECIFIC VOLUME		ENTHALPY			ENTROPY			ABS. PRESS. IN. HG p
		Sat. Liquid v_f	Sat. Vapor v_g	Sat Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid S_f	Evap. S_{fg}	Sat. Vapor S_g	
0.25	40 23	0 01602	2423 7	8.28	1071.1	1079.4	0 0166	2.1423	2.1589	**0.25**
0.50	58.80	0.01604	1256.4	26.86	1060.6	1087.5	0.0532	2.0453	2.0985	**0.50**
0.75	70.43	0.01606	856.1	38.47	1054.0	1092.5	0.0754	1.9881	2.0635	**0.75**
1.00	79.03	0.01608	652.3	47.05	1049.2	1096.3	0.0914	1.9473	2 0387	**1.00**
1.5	91.72	0.01611	444.9	59.71	1042.0	1101.7	0.1147	1.8894	2.0041	**1.5**
2	101.14	0.01614	339.2	69.10	1036.6	1105.7	0.1316	1 8481	1.9797	**2**
4	125.43	0 01622	176.7	93.34	1022 7	1116.0	0.1738	1.7476	1.9214	**4**
6	140 78	0 01630	120.72	108.67	1013.6	1122 3	0 1996	1.6881	1.8877	**6**
8	152.24	0.01635	92.16	120.13	1006.9	1127.0	0 2186	1.6454	1.8640	**8**
10	161.49	0 01640	74.76	129.38	1001 4	1130 8	0.2335	1.6121	1.8456	**10**
12	169.28	0.01644	63.03	137.18	996.7	1133 9	0.2460	1.5847	1.8307	**12**
14	176.05	0.01648	54 55	143.96	992 6	1136.6	0.2568	1.5613	1.8181	**14**
16	182.05	0 01652	48.14	149.98	988 9	1138.9	0 2662	1.5410	1 8072	**16**
18	187 45	0.01655	43.11	155.39	985.7	1141.1	0 2746	1.5231	1.7977	**18**
20	192.37	0.01658	39.07	160.33	982.7	1143.0	0.2822	1.5069	1.7891	**20**
22	196 90	0 01661	35.73	164.87	979 8	1144 7	0.2891	1.4923	1 7814	**22**
24	201.09	0 01664	32.94	169.09	977 2	1146.3	0 2955	1 4789	1.7744	**24**
26	205.00	0.01667	30.56	173.02	974.8	1147.8	0 3014	1.4665	1 7679	**26**
28	208.67	0.01669	28.52	176.72	972.5	1149.2	0.3069	1.4550	1.7619	**28**
30	212.13	0 01672	26.74	180.19	970.3	1150.5	0 3122	1.4442	1 7564	**30**
LB/SQ IN										**LB/SQ IN**
14 696	212 00	0 01672	26 80	180 07	970 3	1150 4	0 3120	1 4446	1 7566	**14.696**
16	216 32	0 01674	24 75	184 42	967.6	1152 0	0 3184	1.4313	1.7497	**16**
18	222 41	0.01679	22.17	190.56	963 6	1154 2	0 3275	1 4128	1 7403	**18**
20	227 96	0.01683	20 089	196 16	960 1	1156.3	0 3356	1.3962	1.7319	**20**
22	233 07	0 01687	18.375	201 33	956 8	1158 1	0 3431	1 3811	1 7242	**22**
24	237 82	0 01691	16.938	206 14	953 7	1159.8	0 3500	1 3672	1 7172	**24**
26	242 25	0 01694	15 715	210.62	950 7	1161.3	0 3564	1 3544	1 7108	**26**
28	246.41	0.01698	14 663	214.83	947 9	1162 7	0 3623	1 3425	1 7048	**28**
30	250.33	0 01701	13.746	218 82	945 3	1164.1	0.3680	1.3313	1 6993	**30**
32	254.05	0 01704	12 940	222.59	942 8	1165 4	0 3733	1 3209	1.6941	**32**
34	257.58	0 01707	12 226	226.18	940 3	1166.5	0 3783	1 3110	1 6893	**34**
36	260 95	0 01709	11.588	229.60	938.0	1167.6	0 3831	1.3017	1 6818	**36**
38	264 16	0 01712	11.015	232 89	935.8	1168 7	0 3876	1 2929	1.6805	**38**
40	267.25	0.01715	10 498	236.03	933.7	1169 7	0 3919	1.2844	1 6763	**40**
42	270.21	0.01717	10 029	239 04	931.6	1170.7	0 3960	1.2764	1.6724	**42**
44	273.05	0 01720	9 601	241.95	929.6	1171.6	0.4000	1.2687	1 6687	**44**
46	275.80	0 01722	9.209	244.75	927.7	1172 4	0 4038	1 2613	1 6652	**46**
48	278 45	0 01725	8 848	247.47	925.8	1173.3	0 4075	1 2542	1 6617	**48**
50	281 01	0.01727	8.515	250.09	924.0	1174 1	0 4110	1 2474	1 6585	**50**
52	283 49	0 01729	8 208	252.63	922.2	1174.8	0.4144	1.2409	1.6553	**52**
54	285.90	0 01731	7 922	255 09	920.5	1175 6	0 4177	1 2346	1 6523	**54**
56	288.23	0.01733	7.656	257.50	918.8	1176 3	0.4209	1 2285	1 6494	**56**
58	290.50	0 01736	7 407	259.82	917.1	1176 9	0 4240	1.2226	1.6466	**58**
60	292.71	0.01738	7 175	262.09	915.5	1177.6	0 4270	1.2168	1 6438	**60**
62	294 85	0.01740	6.957	264.30	913.9	1178 2	0 4300	1 2112	1 6412	**62**
64	296.94	0.01742	6.752	266 45	912.3	1178.8	0.4328	1.2059	1.6387	**64**
66	298.99	0 01744	6 560	268.55	910 8	1179 4	0 4356	1 2006	1 6362	**66**
68	300 98	0 01746	6.378	270 60	909.4	1180.0	0 4383	1 1955	1.6338	**68**
70	302 92	0.01748	6.206	272.61	907 9	1180.6	0 4409	1.1906	1 6315	**70**
72	304.83	0 01750	6.044	274.57	906.5	1181 1	0 4435	1.1857	1 6292	**72**
74	306 68	0.01752	5 890	276.49	905.1	1181.6	0.4460	1.1810	1 6270	**74**
76	308 50	0.01754	5 743	278.37	903.7	1182.1	0.4484	1.1764	1.6248	**76**
78	310.29	0.01755	5.604	280.21	902.4	1182.6	0.4508	1.1720	1.6228	**78**
80	312 03	0 01757	5 472	282.02	901 1	1183 1	0 4531	1.1676	1 6207	**80**
82	313.74	0 01759	5 346	283 79	899 7	1183 5	0 4554	1.1633	1 6187	**82**
84	315.42	0 01761	5 226	285 53	898 5	1184 0	0.4576	1 1592	1 6168	**84**
86	317.07	0 01762	5 111	287 24	897 2	1184 4	0.4598	1.1551	1.6149	**86**
88	318.68	0.01764	5.001	288.91	895 9	1184.8	0 4620	1 1510	1.6130	**88**
90	320.27	0 01766	4 896	290 56	894.7	1185.3	0 4641	1.1471	1 6112	**90**
92	321 83	0 01768	4.796	292 18	893 5	1185 7	0.4661	1 1433	1 6094	**92**
94	323 36	0 01769	4 699	293 78	892 3	1186 1	0 4682	1 1394	1 6076	**94**
96	324 87	0.01771	4 606	295 34	891 1	1186 4	0 4702	1 1358	1 6060	**96**
98	326 35	0 01772	4 517	296.89	889 9	1186 8	0 4721	1 1322	1.6043	**98**
100	327.81	0 01774	4.432	298 40	888 8	1187 2	0 4740	1.1286	1.6026	**100**
150	358.42	0 01809	3.015	330.51	863.6	1194 1	0.5138	0.0556	1 5694	**150**
200	381.79	0 01839	2 288	355.36	843 0	1198 4	0 5435	1.0018	1 5453	**200**
300	417.33	0.01890	1.5433	393.84	809 0	1202 8	0 5879	0.9225	1.5104	**300**
400	444.59	0 0193	1 1613	424.0	780.5	1204 5	0 6214	0 8630	1.4844	**400**
500	467.01	0 0197	0 9278	449 4	755 0	1204 4	0.6487	0 8147	1.4634	**500**

[a]Reprinted by permission from *Thermodynamic Properties of Steam*, by J H Keenan and F G. Keyes, published by John Wiley and Sons, Inc., 1936 edition.

TABLE 8. PROPERTIES OF SATURATED STEAM: TEMPERATURE TABLE[a]

Temp F t	Abs. Pressure Lb per Sq In. p	Abs. Pressure In. Hg p	Specific Volume Sat. Liquid v_f	Specific Volume Evap. v_{fg}	Specific Volume Sat. Vapor v_g	Enthalpy Sat. Liquid h_f	Enthalpy Evap. h_{fg}	Enthalpy Sat. Vapor h_g	Entropy Sat. Liquid S_f	Entropy Evap. S_{fg}	Entropy Sat. Vapor S_g	Temp F t
32	0.08854	0.1803	0.01602	3306	3306	0.00	1075.8	1075.8	0.0000	2.1877	2.1877	32
33	0.09223	0.1878	0 01602	3180	3180	1.01	1075.2	1076.2	0.0020	2.1821	2.1841	33
34	0.09603	0.1955	0 01602	3061	3061	2 02	1074 7	1076.7	0.0041	2.1764	2.1805	34
35	0.09995	0.2035	0.01602	2947	2947	3.02	1074.1	1077.1	0.0061	2.1709	2.1770	35
36	0.10401	0.2118	0.01602	2837	2837	4.03	1073.6	1077.6	0.0081	2 1654	2.1735	36
37	0.10821	0.2203	0.01602	2732	2732	5 04	1073.0	1078 0	0.0102	2.1598	2.1700	37
38	0.11256	0.2292	0 01602	2632	2632	6 04	1072.4	1078.4	0.0122	2.1544	2.1666	38
39	0.11705	0.2383	0 01602	2536	2536	7.04	1071.9	1078.9	0.0142	2 1489	2.1631	39
40	0 12170	0.2478	0 01602	2444	2444	8.05	1071 3	1079 3	0.0162	2.1435	2.1597	40
41	0.12652	0.2576	0 01602	2356	2356	9.05	1070.7	1079.7	0 0182	2.1381	2.1563	41
42	0 13150	0 2677	0.01602	2271	2271	10.05	1070.1	1080.2	0 0202	2.1327	2.1529	42
43	0 13665	0.2782	0 01602	2190	2190	11 06	1069.5	1080 6	0 0222	2 1274	2.1496	43
44	0.14199	0 2891	0 01602	2112	2112	12 06	1068.9	1081 0	0.0242	2.1220	2.1462	44
45	0 14752	0 3004	0 01602	2036 4	2036 4	13.06	1068.4	1081.5	0.0262	2.1167	2.1429	45
46	0 15323	0 3120	0 01602	1964.3	1964 3	14.06	1067.8	1081 9	0 0282	2.1113	2 1395	46
47	0 15914	0 3240	0.01603	1895 1	1895.1	15.07	1067.3	1082 4	0.0302	2.1060	2.1362	47
48	0 16525	0.3364	0 01603	1828 6	1828 6	16 07	1066 7	1082 8	0.0321	2.1008	2 1329	48
49	0 17157	0.3493	0 01603	1764 7	1764.7	17.07	1066 1	1083 2	0.0341	2 0956	2 1297	49
50	0 17811	0 3626	0 01603	1703 2	1703 2	18 07	1065 6	1083 7	0 0361	2 0903	2.1264	50
51	0 18486	0 3764	0 01603	1644 2	1644.2	19 07	1065 0	1084 1	0 0380	2.0852	2.1232	51
52	0 19182	0 3906	0 01603	1587 6	1587.6	20 07	1064 4	1084.5	0 0400	2 0799	2.1199	52
53	0 19900	0 4052	0 01603	1533 3	1533 3	21.07	1063 9	1085 0	0 0420	2 0747	2 1167	53
54	0 20642	0.4203	0 01603	1481 0	1481.0	22 07	1063 3	1085 4	0 0439	2 0697	2 1136	54
55	0 2141	0 4359	0 01603	1430 7	1430 7	23 07	1062 7	1085 8	0 0459	2.0645	2 1104	55
56	0 2220	0.4520	0 01603	1382 4	1382 4	24 06	1062 2	1086 3	0 0478	2.0594	2.1072	56
57	0.2302	0.4686	0 01603	1335 9	1335.9	25 06	1061 6	1086 7	0 0497	2 0544	2.1041	57
58	0.2386	0 4858	0 01604	1291 1	1291 1	26 06	1061 0	1087 1	0 0517	2 0493	2.1010	58
59	0 2473	0 5035	0 01604	1248 1	1248 1	27 06	1060 5	1087 6	0 0536	2 0443	2 0979	59
60	0 2563	0.5218	0 01604	1206 6	1206 7	28 06	1059 9	1088 0	0.0555	2.0393	2 0948	60
61	0 2655	0.5407	0 01604	1166 8	1166 8	29 06	1059.3	1088 4	0.0574	2.0343	2.0917	61
62	0 2751	0.5601	0.01604	1128 4	1128.4	30.05	1058.8	1088 9	0.0593	2.0293	2.0886	62
63	0 2850	0.5802	0 01604	1091.4	1091 4	31.05	1058 2	1089 3	0.0613	2.0243	2 0856	63
64	0 2951	0.6009	0.01605	1055 7	1055 7	32 05	1057.6	1089 7	0 0632	2 0194	2.0826	64
65	0.3056	0.6222	0 01605	1021 4	1021 4	33 05	1057 1	1090 2	0 0651	2 0145	2 0796	65
66	0 3164	0 6442	0 01605	988 4	988 4	34 05	1056 5	1090 6	0 0670	2.0096	2 0766	66
67	0 3276	0 6669	0 01605	956 6	956 6	35.05	1056 0	1091 0	0 0689	2 0047	2.0736	67
68	0 3390	0 6903	0 01605	925.9	925.9	36 04	1055 5	1091.5	0 0708	1 9998	2 0706	68
69	0 3509	0.7144	0 01605	896.3	896.3	37.04	1054 9	1091 9	0.0726	1.9950	2 0676	69
70	0 3631	0.7392	0 01606	867.8	867.9	38.04	1054.3	1092 3	0 0745	1.9902	2.0647	70
71	0 3756	0 7648	0 01606	840.4	840 4	39.04	1053.8	1092.8	0 0764	1.9854	2 0618	71
72	0.3886	0.7912	0 01606	813 9	813.9	40 04	1053 2	1093 2	0 0783	1.9805	2 0588	72
73	0 4019	0.8183	0 01606	788 3	788.4	41.03	1052.6	1093.6	0 0802	1.9757	2 0559	73
74	0 4156	0.8462	0 01606	763 7	763.8	42 03	1052 1	1094 1	0.0820	1.9710	2.0530	74
75	0.4298	0.8750	0 01607	740 0	740 0	43 03	1051 5	1094 5	0 0839	1 9663	2.0502	75
76	0 4443	0 9046	0.01607	717 1	717.1	44 03	1050 9	1094.9	0 0858	1.9615	2 0473	76
77	0 4593	0.9352	0 01607	694 9	694.9	45 02	1050.4	1095.4	0 0876	1.9569	2.0445	77
78	0 4747	0.9666	0 01607	673 6	673 6	46 02	1049 8	1095.8	0 0895	1.9521	2.0416	78
79	0 4906	0.9989	0 01608	653 0	653 0	47.02	1049.2	1096 2	0 0913	1.9475	2 0388	79
80	0 5069	1 0321	0 01608	633 1	633 1	48 02	1048 6	1096 6	0.0932	1.9428	2 0360	80
81	0.5237	1 0664	0 01608	613 9	613 9	49 02	1048.1	1097 1	0.0950	1.9382	2.0332	81
82	0 5410	1 1016	0 01608	595 3	595 3	50 01	1047 5	1097.5	0 0969	1.9335	2.0304	82
83	0 5588	1.1378	0 01609	577.4	577.4	51.01	1046 9	1097.9	0.0987	1 9290	2.0277	83
84	0 5771	1.1750	0 01609	560.1	560.2	52 01	1046 4	1098 4	0.1005	1.9244	2 0249	84
85	0.5959	1.2133	0.01609	543 4	543.5	53 00	1045 8	1098 8	0 1024	1.9198	2.0222	85
86	0 6152	1.2527	0 01609	527.3	527.3	54 00	1045 2	1099 2	0 1042	1.9153	2 0195	86
87	0 6351	1.2931	0 01610	511 7	511 7	55 00	1044 7	1099.7	0.1060	1.9108	2 0168	87
88	0 6556	1 3347	0.01610	496 6	496 7	56 00	1044.1	1100 1	0 1079	1.9062	2 0141	88
89	0 6766	1.3775	0 01610	482.1	482 1	56 99	1043 5	1100 5	0 1097	1 9017	2.0114	89
90	0 6982	1.4215	0 01610	468 0	468 0	57.99	1042 9	1100 9	0.1115	1 8972	2 0087	90
91	0 7204	1.4667	0 01611	454 4	454 4	58.99	1042.4	1101 4	0.1133	1 8927	2 0060	91
92	0.7432	1.5131	0.01611	441 2	441 3	59.99	1041.8	1101 8	0 1151	1.8883	2.0034	92
93	0 7666	1.5608	0 01611	428 5	428 5	60 98	1041 2	1102.2	0 1169	1 8838	2 0007	93
94	0 7906	1 6097	0.01612	416 2	416.2	61.98	1040.7	1102·6	0 1187	1.8794	1.9981	94
95	0.8153	1 6600	0 01612	404 3	404.3	62 98	1040.1	1103 1	0.1205	1 8750	1 9955	95
96	0 8407	1.7117	0 01612	392 8	392.8	63.98	1039.5	1103 5	0.1223	1.8706	1.9929	96
97	0 8668	1.7647	0 01612	381 7	381.7	64.97	1038.9	1103.9	0.1241	1.8662	1.9903	97
98	0.8935	1.8192	0 01613	370 9	370 9	65 97	1038 4	1104.4	0 1259	1.8618	1.9877	98
99	0 9210	1 8751	0 01613	360.4	360 5	66 97	1037.8	1104.8	0 1277	1.8575	1.9852	99
100	0 9492	1.9325	0 01613	350.3	350.4	67.97	1037.2	1105 2	0.1295	1.8531	1 9826	100
101	0.9781	1.9915	0.01614	340 6	340.6	68.96	1036.6	1105 6	0.1313	1.8488	1.9801	101

[a] Reprinted by permission from *Thermodynamic Properties of Steam*, by J. H. Keenan and F. G. Keyes, published by John Wiley and Sons, Inc., 1936 edition.

TABLE 9. WEIGHT OF SATURATED AND PARTLY SATURATED AIR[a]

Dry-Bulb Temp Deg F	Weight of Saturated Air for Various Barometric and Hygrometric Conditions—Pounds per Cubic Foot							Approx Average Increase in Weight per Deg Wet-Bulb Depression
	Barometric Pressure Inches of Mercury						Increase In Weight Per 0 1 In Rise in Barometer	
	28 5	29 0	29 5	30 0	30 5	31 0		
30	0.07703	0.07839	0.07974	0.08110	0.08245	0.08381	0.00027	0.000017
32	0.07671	0.07806	0.07940	0.08075	0.08210	0.08345	0.00027	0.000017
34	0.07638	0.07772	0.07907	0.08041	0.08175	0.08310	0.00027	0.000018
36	0.07605	0 07739	0.07873	0.08007	0.08141	0.08274	0.00027	0.000018
38	0.07573	0.07706	0.07840	0.07973	0.08106	0.08239	0.00027	0.000019
40	0.07541	0.07674	0.07806	0.07939	0.08072	0.08205	0.00027	0.000019
42	0.07509	0.07641	0.07773	0.07905	0.08038	0.08170	0.00026	0.000020
44	0.07477	0.07609	0.07740	0.07872	0.08004	0.08135	0.00026	0.000020
46	0.07445	0.07576	0.07707	0.07838	0.07970	0.08101	0.00026	0 000021
48	0.07413	0.07544	0.07674	0.07805	0.07936	0.08066	0.00026	0 000021
50	0.07381	0.07512	0.07642	0.07772	0.07902	0.08032	0.00026	0.000022
52	0.07350	0.07479	0.07609	0.07739	0.07868	0.07998	0.00026	0.000023
54	0.07318	0.07447	0.07576	0.07706	0 07835	0.07964	0.00026	0.000023
56	0.07287	0.07415	0.07544	0.07673	0.07801	0.07930	0.00026	0.000024
58	0.07255	0.07383	0.07512	0.07640	0.07768	0.07896	0.00026	0.000025
60	0.07224	0.07352	0.07479	0.07607	0.07734	0.07862	0.00026	0.000026
62	0.07193	0.07320	0.07447	0.07574	0.07701	0.07828	0.00026	0.000027
64	0.07161	0.07288	0.07414	0.07541	0.07668	0.07794	0.00026	0.000028
66	0.07130	0.07256	0.07382	0.07508	0.07634	0.07760	0.00026	0.000029
68	0.07098	0.07224	0.07350	0.07475	0.07601	0.07727	0.00026	0.000030
70	0.07067	0.07192	0.07317	0.07442	0.07568	0.07693	0.00026	0 000031
72	0.07035	0.07160	0.07285	0.07410	0.07534	0.07659	0.00025	0.000032
74	0.07004	0.07128	0.07252	0.07377	0.07501	0.07625	0.00025	0 000033
76	0.06972	0.07096	0.07220	0.07343	0.07467	0.07591	0.00025	0.000034
78	0.06940	0.07064	0.07187	0.07310	0.07434	0 07557	0.00025	0 000036
80	0.06909	0.07032	0.07155	0.07277	0.07400	0.07523	0 00025	0.000037
82	0.06877	0.07000	0.07122	0.07244	0.07366	0.07489	0.00024	0.000039
84	0.06845	0.06967	0.07089	0.07211	0 07333	0.07454	0.00024	0.000040
86	0.06812	0.06934	0.07056	0.07177	0.07299	0.07420	0.00024	0.000042
88	0.06780	0.06901	0.07022	0.07143	0.07264	0.07385	0.00024	0.000043
90	0.06748	0.06868	0.06989	0.07109	0.07230	0.07351	0.00024	0.000045
92	0.06715	0.06835	0.06955	0.07075	0.07195	0.07316	0.00024	0.000047
94	0.06682	0.06801	0.06921	0.07041	0.07161	0.07280	0.00024	0.000049
96	0.06648	0.06768	0.06887	0.07006	0.07126	0.07245	0.00024	0.000051
98	0.06615	0.06734	0.06853	0.06972	0.07091	0.07209	0.00024	0.000053
100	0.06581	0.06700	0.06818	0.06937	0.07055	0 07174	0 00024	0 000055

[a]Approximate average decrease in weight per 0 1 F rise in dry-bulb temperature equals 0 000017 lb per cubic foot

pound of dry air. Since most air conditioning processes are of the steady flow type in which the thermal energy convected with the fluid is its enthalpy, the third is most simply expressed in terms of enthalpy per pound of dry air. It is clear, therefore, that humidity ratio W and enthalpy per pound of dry air h are fundamental coordinates. Their use for the purpose of graphical representation is due to Mollier [3]. A convenient modification of the Mollier diagram devised by Goff is obtained by taking humidity ratio W as ordinate and *reduced enthalpy* $(h - 1000W)$ as abscissa, as shown in the chart enclosed in the envelope attached to the inside back cover of this book.

The reasons for the use of the difference $(h - 1000W)$ as abscissa instead of h itself in the Mollier Diagram for Moist Air are the following:

(1) it amounts to plotting on oblique coordinates and thus reduces to convenient proportions a diagram which would otherwise take the form of a scroll; (2) by the choice of the factor 1000 the necessary multiplication reduces to shifting the decimal point; (3) the ease with which the ordinate W can be multiplied by 1000 and added to the abscissa to obtain the enthalpy h makes it unnecessary to complicate the chart by a family of isenthalpic lines.

In the Mollier diagram, the lines inclined upward and slightly to the right are lines of constant (dry-bulb) temperature. They are straight under Dalton's Law but actually have slight curvature. The lines inclined upward to the left are lines of constant thermodynamic wet-bulb and are straight by definition. The dry-bulb and wet-bulb lines meet at the saturation curve and coincide in the region to the left of this curve. This region is divided into three sub-regions by the narrow *wedge* with apex at the junction of the 32 F wet-bulb and dry-bulb lines. Above the wedge, the mixture consists of two distinct phases, saturated vapor and saturated liquid. At point A, for example, the temperature is 60 F and the vapor phase contains 0.01103 lb of water vapor per pound of dry air from

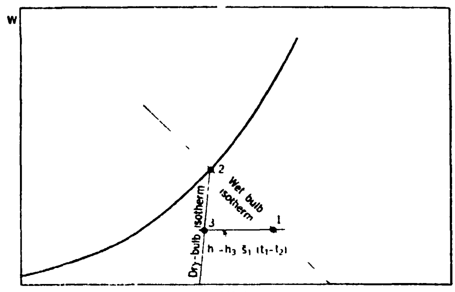

FIG 2 DIAGRAM ILLUSTRATING THERMODYNAMIC WET-BULB TEMPERATURE

Table 6. From the Mollier diagram, $W = 0.016$ lb, leaving 0.00497 lb per pound dry air in the liquid phase. The total enthalpy of the mixture is $10.51 + (1000 \times 0.016) = 26.51$ Btu per pound dry air of which $15.34 + (1000 \times 0.01103) = 26.37$ Btu per pound dry air is contributed by the vapor phase.

Within the wedge, the mixture consists of three distinct phases, saturated vapor, saturated solid and saturated liquid. The temperature is 32 F; and the relative proportions of the three phases depend upon the location of the state point within the wedge. Below the wedge, the mixture consists of saturated vapor and saturated solid.

The curved lines in the single vapor-phase region to the right of the saturation curve are lines of constant per cent saturation. Lines of constant dew-point are, of course, horizontal straight lines of constant humidity ratio. At point B, for example, the dry-bulb temperature is 60 F, the thermodynamic wet-bulb is 50 F, the dew-point is 40.8 F, the degree of saturation is 48.6 per cent, the humidity ratio is 0.00536 lb per pound dry air, and the enthalpy is $14.85 + (1000 \times 0.00536) = 20.21$ Btu per pound dry air.

With the aid of the Mollier diagram, it is easy to throw the definition of thermodynamic wet-bulb, Equation 20, into a more familiar form. Consider the three points 1, 2, 3, Fig. 2. Point 3 is located with respect

to points 1 and 2 so that $W_3 = W_1$ and $t_3 = t_2$. Points 1 and 2, being on a line of constant thermodynamic wet-bulb, satisfy Equation 20; thus,

$$h_1 - h_3 + (W_2 - W_1)\, h'_{w,2} = h_2 - h_3$$

where h_3 has been subtracted from both sides. Under Dalton's Law, $h_2 - h_3 = (W_2 - W_1)\, h_{w,2}$; moreover, $h_1 - h_3$ may be replaced by $\bar{s}_1 (t_1 - t_2)$ where $\bar{s}_1$ is often referred to as *mean humid heat* and may be calculated with good approximation from

$$\bar{s}_1 = 0.240 + 0.444\, W_1 \tag{21}$$

Finally, introducing latent heat of vaporization at the wet-bulb temperature, namely, $(h_{fg})_2 = (h_w - h'_w)_2$, Equation 20 becomes, after omitting the subscript 2,

$$\frac{t_1 - t'}{W_s - W_1} = \frac{h_{fg}}{\bar{s}_1} \tag{22}$$

it being understood that W_s is the saturation humidity ratio and h_{fg}, the latent heat, at the wet-bulb temperature t'. Equation 22 was derived by Carrier [1].

Example 12. Work Example 10 using Equation 22.

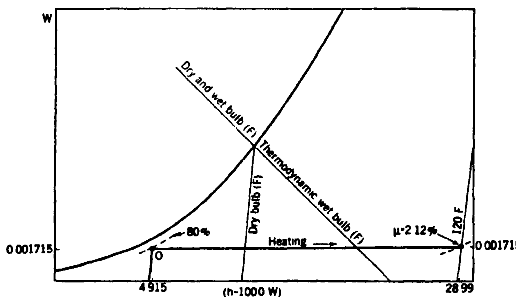

Fig. 3. Diagram Illustrating Example 13

Solution. A trial-by-error method is involved. Taking 48 F as a trial value of t', $(80 - 48) \div (0.007072 - 0) = 4520$; but $1066.7 \div 0.240 = 4440$. The trial value must, therefore, be revised upward, the final solution being 48.26 F as in Example 10.

TYPICAL AIR CONDITIONING PROCESSES

Illustrative Examples. The use of Table 6 and the Mollier diagram in analyzing typical air conditioning processes is best explained by the use of illustrative examples. In each of these examples, the observed pressure is assumed to be standard atmospheric pressure (29.921 in. Hg).

Example 13. Heating. Air at 20 F and 80 per cent saturation is to be heated to 120 F. Analyze the process as illustrated in Fig. 3.

Solution. The initial humidity ratio is $0.80 \times 0.002144 = 0.001715$ lb per pound dry air (table). This same value is read directly on the chart. The initial enthalpy is $4.798 + (0.80 \times 2.290) = 6.630$ Btu per pound dry air (table) or $4\,915 + (1000 \times 0.001715) = 6.630$ (chart).

The final degree of saturation is $0.001715 \div 0.08093 = 0\,0212$ (table); hence the final enthalpy is $28.80 + (0.0212 \times 90.09) = 30.71$ Btu per pound dry air (table) or $28.99 + (1000 \times 0.001715) = 30.71$ (chart).

The increase in enthalpy is the quantity of heat to be supplied, namely, $30.71 - 6\,63 = 24.08$ Btu per pound dry air (table). Since humidity ratio W and therefore $1000W$ is constant, this is also simply the horizontal distance between the representative points on the chart; thus, the heat to be supplied is also $28.99 - 4.915 = 24.08$ Btu per pound dry air (chart).

The final volume is 14.60 + (0.0212 × 1.90) = 14.64 cu ft per pound (table); or direct from the volume chart. Therefore, if 20,000 cfm of heated air is to be supplied, the quantity of heat required is (20,000 ÷ 14.64) × 24.08 = 32,900 Btu per minute.

Example 14. Cooling and Separating. Air at 95 F and 50 per cent saturation is to be cooled to 70 F and the liquid separated out. Analyze the process as shown in Fig. 4.

Solution. The initial humidity ratio is 0.50 × 0 03652 = 0.01826 (table). The initial enthalpy is 22.80 + (0.50 × 40.25) = 42.93 Btu per pound dry air (table) or 24.67 + (1000 × 0.01826) = 42.93 (chart).

The final state is in the two-phase region and consists of 0.01574 lb water per pound dry air in the vapor phase, and 0.00252 lb water per pound dry air in the liquid phase. The final enthalpy is therefor 33.96 + (0.00252 × 38.0) = 34.06 Btu per pound dry air (table) or 15.80 + (1000 × 0.01826) = 34.06 (chart).

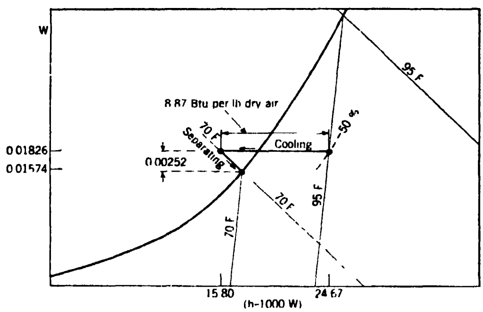

FIG. 4. DIAGRAM ILLUSTRATING EXAMPLE 14

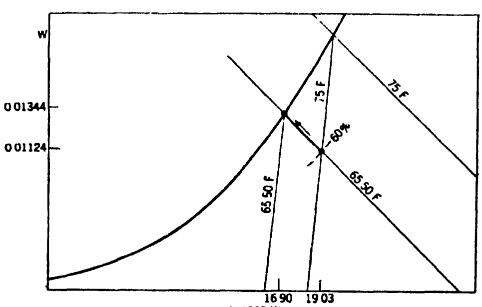

FIG. 5. DIAGRAM ILLUSTRATING EXAMPLE 15

The decrease of enthalpy is the refrigeration to be supplied and is 42.93 − 34.06 = 8.87 Btu per pound dry air (table). Since the weight of water per pound of dry air is constant, this is also the horizontal distance between the representative points on the chart, namely, 24.67 − 15.80 = 8 87 Btu per pound dry air (chart).

The initial volume is 13.97 + (0.50 × 0.82) = 14.38 cu ft per pound (table); or direct from the volume chart. Therefore, if 20,000 cfm of initial air is to be processed, the refrigeration required is (20,000 × 8.87) ÷ (14.38 × 200) = 61.7 tons. The weight of water to be removed is (20,000 × 0.00252) ÷ 14.38 = 3.51 lb per minute.

Example 15. Adiabatic Saturation with Recirculated Spray Water. Air at 75 F and 60 per cent saturation is saturated adiabatically with spray water which is recirculated. Find the resulting temperature and the weight of water added per pound of dry air as outlined in Fig. 5.

Solution. The recirculated water will assume the thermodynamic wet-bulb temperature of the entering air which will also be the temperature of the resulting saturated mixture. The humidity ratio of the entering air is 0.60 × 0.01873 = 0.01124 lb water per pound dry air (table); its enthalpy is 17.99 + (0.60 × 20.47) = 30.27 Btu per pound

dry air (table) or 19.03 + (1000 × 0.01124) = 30.27 Btu per pound dry air (chart). To determine the resulting temperature, the following equation must be solved.

$$30.27 + (W_s - 0.01124)\, h'_w = h_s$$

A trial value is 65 F corresponding to h_s = 30.27 The final value is 65 50 F corresponding to h_s = 30.27 + (0.01320 − 0 01124) × 33.05 = 30.34 Btu per pound dry air. The weight of water to be added is 0.01344 − 0.01124 = 0.00200 lb per pound dry air.

The volume of the entering air is 13.47 + (0.60 × 0.40) = 13.71 cu ft per pound. If 20,000 cfm of entering air is to be saturated, the weight of water to be added per minute is (20,000 × 0 00200) ÷ 13.71 = 2.92 lb per minute.

Adiabatic Mixing of Two Air Streams

A typical process requiring special discussion is the adiabatic mixing of two air streams. Let stream 1 contain M_1 pounds of dry air per minute and let its enthalpy be h_1 and its humidity ratio W_1. Using subscripts 2 and 3 in a similar manner to designate stream 2 and the resulting mixture respectively, write:

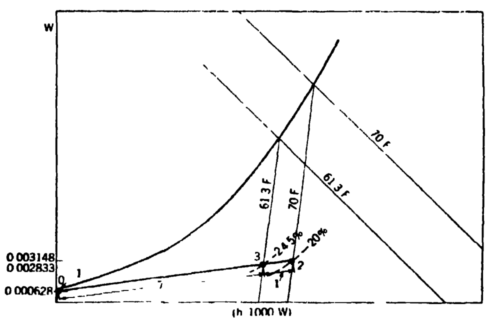

FIG. 6. DIAGRAM ILLUSTRATING EXAMPLE 16

$M_1 + M_2 = M_3$ (weight balance for the dry air)
$M_1W_1 + M_2W_2 = M_3W_3$ (weight balance for the water)
$M_1h_1 + M_2h_2 = M_3h_3$ (energy balance, no heat absorbed)

Eliminating M_3,

$$\frac{W_2 - W_3}{W_3 - W_1} = \frac{h_2 - h_3}{h_3 - h_1} = \frac{M_1}{M_2} \qquad (23)$$

according to which: *on the Mollier Chart the representative point of the resulting mixture lies on the straight line connecting the representative points of the two streams being mixed, and divides the line into two segments which are in the same ratio as the weights of dry air in the two streams.* It must not be forgotten that this analysis assumes *adiabatic* mixing.

Example 16. Outside air at 0 F and 80 per cent saturation is to be mixed adiabatically with recirculated air at 70 F and 20 per cent saturation in the ratio one pound of dry air in the former to seven in the latter. Find the temperature and degree of saturation of the resulting mixture as shown in Fig 6

Solution. The humidity ratio and enthalpy of the resulting mixture satisfy

$$\frac{0.003148 - W_3}{W_3 - 0.000628} = \frac{20\,23 - h_3}{h_3 - 0.666} = \frac{1}{7}$$

whence,

W_3 = 0.002833 lb water per pound dry air.
h_3 = 17.78 Btu per pound dry air.

The corresponding temperature and degree of saturation are 61.3 F and 24.5 per cent as is easily verified by use of Table 6. The numerical solution is somewhat tedious, but the graphical solution is easy.

Adiabatic Mixing with Injected Water

Another typical process is that of injecting water (solid, liquid or vapor) into an air stream to mix adiabatically with it. Let the subscripts 1 and 2 refer to the initial and final conditions, respectively; then write

$$1 + 0 = 1 \quad \text{(weight balance for the dry air)}$$
$$W_1 + (W_2 - W_1) = W_2 \quad \text{(weight balance for the water)}$$
$$h_1 + (W_2 - W_1)\,h_W = h_2 \quad \text{(energy balance, no heat absorbed)}$$

The first two are identities and are incorporated in the third. This may be rewritten as follows:

$$\frac{h_2 - h_1}{W_2 - W_1} = h_W \tag{24}$$

and shows that *the process is represented by a straight line on the Mollier*

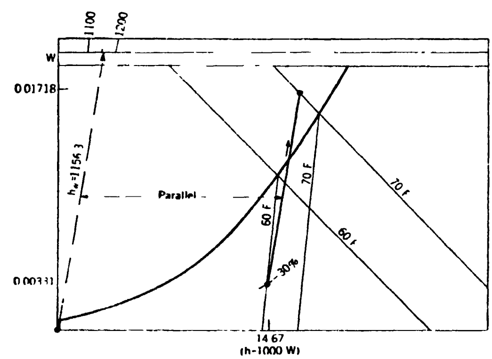

FIG. 7. DIAGRAM ILLUSTRATING EXAMPLE 18

diagram, the slope of the line being determined by the specific enthalpy of the injected water. It must not be forgotten that the analysis assumes adiabatic mixing. *Energy convected with a fluid is not heat.*

Example 17. It is desired to increase the humidity ratio of air at 70 F without changing its temperature. Under what conditions may water be injected in order to accomplish the desired result?

Solution. Under Dalton's Law a line of constant (dry-bulb) temperature is straight on the Mollier diagram and its slope is determined by the specific enthalpy of water *vapor* at the given temperature. At 70 F, h_W = 1092.3 Btu per pound; hence injection of steam having this specific enthalpy will cause the representative point to move in a direction parallel to the 70 F isotherm. Saturated steam at 70 F may not be used because its pressure is only 0 7392 in. Hg and it cannot therefore be injected into air at atmospheric pressure. Saturated steam at 667.4 F, 2488 lb per square inch has the right specific enthalpy and can be throttled into a room at 70 F without altering the room temperature

Border Scale

On the Mollier diagram is placed a border scale to facilitate the graphical solution of problems in which given quantities of energy and water are added (or withdrawn) simultaneously as in the case of adiabatic mixing with injected water. All marks in the upper half of this scale point to the lower left corner of the chart and each shows the *direction* that the representative point will move due to adiabatic mixing with injected water having the indicated specific enthalpy. All marks in the lower half of the scale point to the lower right corner of the chart.

Example 18. If dry saturated steam at 20 lb per square inch absolute is injected into air initially at 60 F and 30 per cent saturation to raise the temperature to 70 F, what is the final degree of saturation and how much water is added per pound dry air? (See Fig. 7.)

Solution. The initial humidity ratio is $0.30 \times 0.01103 = 0.00331$ lb water per pound dry air (or direct from chart). The initial enthalpy is $14.39 + (0.30 \times 11.98) = 17.98$ Btu per pound dry air (table) or $14.67 + (1000 \times 0.00331) = 17.98$ (chart). A preliminary calculation shows that the final mixture contains liquid. The final weight of water per pound of dry air is determined from

$$\frac{33.96 + (W - 0.01574) \times 38.0 - 17.98}{W - 0.00331} = 1156.3$$

where the specific enthalpy of the injected water is 1156.3 Btu per pound. The answer is

$$W = 0.01718 \text{ lb water per pound dry air.}$$

Therefore, the weight of water added is $0.01718 - 0.00331 = 0.01387$ lb per pound dry air as shown in Fig. 7.

Adiabatic Saturation

Any case of adiabatic mixing in which the resulting mixture is saturated may properly be called *adiabatic saturation.* For example, if enough water at 352 F be sprayed into dry air at 80 F to produce a saturated mixture, the resulting enthalpy will be $h_s = 19.19 + (W_s - 0)\ 324$; and since h_s and W_s are functions of the same temperature, this temperature is determined by the equation to be 53.0 F. Thus, adiabatic saturation of dry air at 80 F by injecting liquid water at 352 F results in a temperature of 53.0 F when saturation is reached.

But in practice, much more is usually read into the term adiabatic saturation, it being generally understood that saturation is to be produced by injecting liquid water at such a temperature as will *coincide* with that at which the saturation curve is reached. *With this understanding it may be said that thermodynamic wet-bulb temperature is the result of adiabatic saturation.* Thus, if liquid water at 48.26 F instead of 352 F be injected into dry air at 80 F a saturated mixture at 48.26 F instead of 53.0 F will be produced. Therefore, 48.26 F is the thermodynamic wet-bulb temperature of dry air at 80 F.

It is possible to produce adiabatic saturation, interpreting the term literally, by mixing two air streams neither of which is itself saturated. In order for this to be possible, the straight line connecting the representative points on the Mollier diagram must cut the saturation curve twice.

Cooling Load

In the calculation of the cooling load for an air conditioned space, the problem usually reduces to determining the quantity of inside air that must be withdrawn and the condition to which it must be brought by cooling, separating and possibly reheating so that return of the conditioned air will have the net effect of removing given amounts of energy and water from the air conditioned space.

Let m denote the weight of dry air withdrawn per hour. With it will be withdrawn *energy* of amount mh_1 Btu per hour and *water* of amount mW_1 pounds per hour, where h_1 and W_1 denote enthalpy and humidity ratio, respectively, of inside air. The weight of dry air returned per hour will be the same as that withdrawn but with it must be returned a smaller amount of *energy*, mh Btu per hour, and a smaller quantity of *water*, mW pounds per hour, where h and W denote enthalpy and humidity ratio of conditioned air.

With this understanding, the requirements of the cooling load problem are,

$$mh = mh_1 - \Delta Q$$
$$mW = mW_1 - \Delta W$$

where ΔQ and ΔW are the *given* amounts of energy and water, respectively, to be removed simultaneously. Eliminating m from these equations,

$$\frac{h - h_1}{W - W_1} = \frac{\Delta Q}{\Delta W} = q \tag{25}$$

which says that all possible states for the conditioned air lie on a straight line, on the Mollier Chart, which passes through the state point of the Inside Air with a slope determined by the ratio q (Btu per lb water) of the quantities of energy and water to be removed simultaneously. This straight line is called the *condition line* for the given problem.

If the condition line crosses the saturation curve the intersection is called the *apparatus dew-point* (Chapter 20). For, if the air conditioning apparatus is set to produce a saturated mixture having the temperature corresponding to this point, the introduction of this saturated mixture

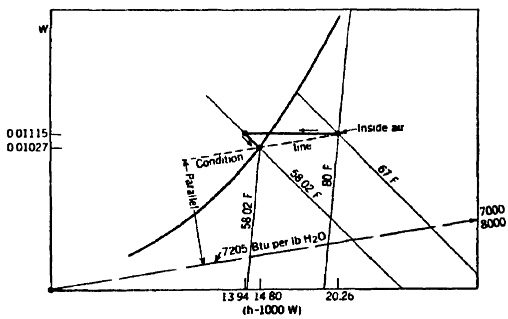

Fig 8. Diagram Illustrating Example 19

into the conditioned space will result in the simultaneous removal of the required amounts of energy and water.

Graphical solution of a cooling load problem is facilitated by the border scale on the Mollier Chart. The numbers around this border scale may be regarded as values of the ratio q (Btu per lb water), each number determining the *direction* of the corresponding condition line.

Example 19 A condition of 80 F dry-bulb, 67 F wet-bulb is to be maintained in a certain clothing store, outside conditions being 95 F dry-bulb, 75 F wet-bulb. The energy gain from normal heat transmission is estimated at 16,000 Btu per hour, that from solar radiation at 48,000 Btu per hour. The energy generated by lights, fans, etc., is estimated at 13,900 Btu per hour. The ventilation requirement is 30,000 cu ft of Outside Air per hour. The number of occupants is 50. Find the apparatus dew-point and the cooling load as analyzed in Fig. 8.

Solution. The thermodynamic properties of Outside Air are: $v = 14.29$ cu ft per pound dry air, $h = 38.26$ Btu per pound dry air, and $W = 0.01402$ lb water per pound dry air. Therefore, the weight of dry air entering with the Ventilating Air is $30,000 \div 14.29 = 2099$ lb dry air per hour. This brings with it energy of amount $2099 \times 38.26 = 80,300$ Btu per hour and water of amount $2099 \times 0.01402 = 29.43$ lb per hour. An equal weight of dry air must be displaced from the store.

The thermodynamic properties of Inside Air are: $h_1 = 31.41$ Btu per pound dry air, and $W_1 = 0.01115$ lb water per pound dry air. Therefore, the energy leaving the store

with the Inside Air displaced by the Ventilating Air is 2099 × 31.41 = 65,900 Btu per hour; the weight of water leaving is 2099 × 0.01115 = 23.41 lb per hour.

Each occupant may be regarded as a *normal person standing at rest* and therefore evaporating 0.198 lb of water per hour at about 79 F (Table 3, Chapter 2). Therefore the energy added to the store by such evaporation is 50 × 0.198 × 1096.2 (enthalpy of saturated vapor at 79 F, Table 8) = 11,000 Btu per hour, the weight of water added being 50 × 0.198 = 9.90 lb per hour. In addition each person loses 225 Btu of heat per hour by conduction, convection and radiation, making a total for 50 persons of 11,300 Btu per hour.

An energy balance shows a net gain of 16,000 + 48,000 + 13,900 + 80,300 − 65,900 + 11,000 + 11,300 = 114,600 Btu per hour. A water balance shows a net gain of 29.43 − 23.41 + 9.90 = 15.92 lb per hour. The slope of the condition line is determined by the ratio q = 114,600 ÷ 15.92 = 7205 Btu per pound of water. The temperature at which the condition line crosses the saturation curve is 58.02 F which is, therefore, the apparatus dew-point. This temperature is found by solving Equation 25,

$$\frac{31.41 - h_s}{0.01115 - W_s} = 7205$$

The fact that a trial-by-error solution is required is not a serious complication.

In order to calculate the cooling load it will be assumed that the air conditioning process consists of cooling and separating The thermodynamic properties entering the calculations are:

	Inside Air	*After Cooling*	*After Separating*
t	80.0	58 02	58 02
W	0.01115	0 01115	0 01027
h	31 41	25 086	25 063

It follows that the refrigeration required is 31.41 − 25.086 = 6.324 Btu per pound dry air. But the weight of dry air involved is 114,600 ÷ (31.41 − 25 063) = 18,056 pounds per hour; hence the total refrigeration required, namely, the cooling load, is 18,056 × 6.324 = 114,185 Btu per hour, or 114,185 ÷ 12,000 (Btu extracted per hour per ton of refrigeration) = 9.49 tons.

The weight of water removed is 18,056 × (0.01115 − 0.01027) = 15.92 lb per hour as required. This water is removed as liquid at 58.02 F and therefore removes energy of amount 15.92 × 26.1 (specific enthalpy of liquid water at 58 02 F, Table 6) = 415 Btu per hour. This plus the refrigeration accounts for the total removal of 114,600 Btu per hour as required.

In practice the point at which the condition line crosses the saturation curve may dictate an excessive number of air changes. If so, it may be necessary to cool to a lower temperature. But, if the requirements of the problem are to be exactly met both as regards removal of energy and removal of water, the mixture returned to the conditioned space must then contain a certain amount of liquid. In other words, its state point must lie on the condition line.

It may be that the condition line does not cross the saturation curve at all, in which case the apparatus dew-point as defined previously does not exist. In this case the actual dew-point of the apparatus can be set at any temperature provided the air is then reheated to a point on the condition line before being returned to the conditioned space.

In actual practice it is rarely possible to obtain complete saturation at the dew-point temperature at which the apparatus is set. This may be due to insufficient contact; or a portion of the air may be deliberately by-passed. But either is equivalent to reheating and, if the final condition still lies on the condition line, the requirements of the problem can be exactly met.

Heating Load

The idea of the condition line is also useful in calculating heating load problems. Its use is best illustrated by means of an illustrative example.

Example 20. The clothing store of Example 19 is to be maintained at 70 F dry-bulb, 50 per cent saturation in winter, with outside design conditions being 0 F dry-bulb, 80 per cent saturation. In order to avoid window condensation with the given inside and outside conditions, double doors and windows are provided. Show windows are sealed. The ventilation requirements of 10 cfm per person for 50 persons, or 30,000 cfh, will build up a slight pressure. For these three reasons, infiltration is reduced to a negligible amount. The normal heat transmission through walls, partition, floor, roof, glass, and doors is estimated at 73,750 Btu per hour. Considerable energy is gained from lights and occupants, but only after the store is raised to the proper conditions; hence this item should be disregarded in figuring the maximum heating load. Analyze the problem as shown in Fig 9

Solution. The thermodynamic properties of Outside Air are: v = 11.59 cu ft per pound of dry air, h = 0 67 Btu per pound of dry air, and W = 0.00063 lb water per pound of dry air. Accordingly, the Ventilating Air introduces dry air of amount 30,000 ÷ 11.59 = 2590 lb per hour, energy of amount 2590 × 0 67 = 1730 Btu per hour, and water of amount 2590 × 0.00063 = 1.63 lb per hour. Since infiltration is negligible, none of the Ventilating Air will be admitted directly to the store, but will enter with the Supply Air after having been processed in the air conditioning apparatus. Nevertheless, it displaces an equal weight of dry air from the store.

The thermodynamic properties of Inside Air are: v = 13.51 cu ft per pound of dry air, h = 25.38 Btu per pound of dry air, and W = 0.00787 lb water per pound of dry air. Accordingly, the Ventilating Air displaces energy of amount 2590 × 25 38 = 65,730 Btu per hour, and water of amount 2590 × 0 00787 = 20.38 lb per hour.

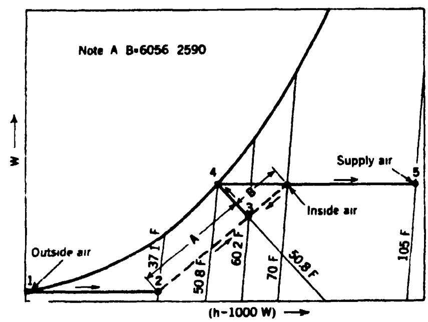

FIG 9. DIAGRAM ILLUSTRATING EXAMPLE 20

Using these data, it appears that the total energy to be added to the store is 73,750 + 65,730 = 139,480 Btu per hour while the total water to be added is 20.38 lb per hour. But it would be a mistake to determine the condition line by the ratio of these two quantities, since the dry air returned with the Supply Air exceeds that recirculated from the store by the amount introduced with the Ventilating Air. It is correct, however, to lump the Recirculated Air and the Displaced Air together, since both leave the store under the conditions of Inside Air. Then the Supply Air must return more energy to the store than both of these remove by an amount equivalent to the normal heat transmission or 73,750 Btu per hour, and more water by amount *zero*. The ratio of these two quantities determines the condition line Since the ratio is infinite, the condition line is horizontal on the Mollier Chart; in other words, the humidity ratio of the Supply Air must be the same as that of Inside Air.

Good practice is to limit the temperature of the Supply Air to 105 F. The condition line crosses the 105 F isotherm at 15.6 per cent saturation. The thermodynamic properties of the Supply Air are: h = 33.91 Btu per pound of dry air, and W= 0.00787 lb water per pound of dry air. Accordingly, the weight of dry air to be recirculated is 73,750 ÷ (33 91 − 25.38) minus 2590 = 6056 lb per hour.

The dew-point temperature of the Supply Air is the same as that of Inside Air, namely, 50 8 F. A suitable conditioning process is to (see Fig. 9): (1-2) preheat the Ventilating Air to temperature t; (2-3) mix it adiabatically with Recirculated Air; (3-4) saturate the resulting mixture adiabatically with recirculated spray; (4-5) reheat to 105 F on the condition line The temperature t must be chosen so that, after preheating, the wet-bulb of the Ventilating Air is 50.8 F.

The determination of the preheating temperature t using the data of Table 6, though straightforward, is somewhat tedious. Graphical solution using the Mollier Chart is easier. The answer 37.1 F is obtained by drawing the line A + B so that the length of

A is in proportion to the length of B as the weight of Recirculated Air 6056 lb is to the weight of Ventilating Air 2590 lb, and where this line crosses (1-2) temperature t results. The data needed to calculate the quantities of heat required for preheating and subsequent reheating may now be assembled.

	Outside Air	*After Preheating*	*After Mixing*	*After Adiabatic Saturation*	*After Reheating*
t	0	37.1	60.2	50.8	105.0
h	0.67	9.59	20.65	20.69	33.91
W	0.00063	0.00063	0.00570	0.00787	0.00787

The quantity of heat required for preheating is 2590 × (9.59 − 0.67) = 23,100 Btu per hour; and that required for reheating is 8646 × (33.91 − 20.69) = 114,300 Btu per hour.

A trial balance for the energy accounting may be made. The Ventilating Air brings in energy 1730 Btu per hour; the heat added by the preheating coil is 23,100 Btu per hour; the energy supplied by the spray is (6056 + 2590) × (20.69 − 20.65) = 340 Btu per hour; the heat added by the reheating coil is 114,300 Btu per hour; and the total is 139,470 Btu per hour. This is in substantial agreement with the stated requirements of the problem.

The Ventilating Air brings in water of amount 1.63 lb per hour; the spray adds (6056 + 2590) × (0.00787 − 0.00570) = 18.76 lb per hour; and the total is 20.39 lb which is in agreement with the stated requirements.

STEADY FLOW ENERGY EQUATION

It was previously stated that, in steady flow, the energy convected by the fluid at any section is the sum of (*a*) kinetic energy due to velocity; (*b*) gravitational energy due to elevation; (*c*) enthalpy due to the condition of pressure, temperature and composition of the fluid. A more detailed discussion of item (*a*) is in order.

Kinetic Energy

There are reasons to believe that the so-called *velocity pressure* h_v read by a Pitot tube is simply the kinetic energy per unit volume of the fluid immediately upstream from the tube, as application of Bernoulli's Equation suggests. Thus

$$V = 1097.3\sqrt{\frac{h_v}{d}} \tag{26}$$

where

V = velocity, feet per minute.
h_v = velocity pressure, inches of water at 60 F.
d = density of fluid, pounds per cubic foot.

In the case of flow through a duct, the velocity pressure is found to vary considerably over the section and a traverse has to be made. The cross-sectional area of the duct is divided into a number of equal concentric areas, and measuring stations are located at centroidal points in each area along two perpendicular diameters. Usually the ultimate object is to determine an *average* velocity $\overline{V}$ from which the weight of fluid crossing the section per unit time can be obtained on multiplying by the cross-sectional area of the duct and by the density of the fluid. This is obtained by simply averaging the square roots of all measured velocity pressures as follows:

$$\overline{V} = \frac{1097.3}{\sqrt{d}}\left(h_v^{1/2}\right)_{av} \tag{27}$$

where

$\overline{V}$ = average velocity, feet per minute.
$(h_v^{1/2})_{av}$ = arithmetic average of the square roots of all measured velocity pressures, inches of water at 60 F.

But the item of present importance is the *average kinetic energy* convected with each pound of fluid. Consistently with the previous discussion, this can be shown to be

$$\overline{KE} = 0.006678\, v \frac{(h_v^{3/2})_{av}}{(h_v^{1/2})_{av}} \tag{28}$$

where

$\overline{KE}$ = average kinetic energy, Btu per pound.
v = specific volume, cubic feet per pound.
$(h_v^{3/2})_{av}$ = arithmetic average of the 3/2-powers of all measured velocity pressures, inches of water at 60 F.

If the velocity pressure were uniform over the section, Equations 27 and 28 could be combined to give

$$\overline{KE} = \left(\frac{\overline{V}}{13{,}430}\right)^2 \tag{29}$$

But, it is interesting to note that if the velocity varies *parabolically* from zero at the walls to maximum at the center as it does in the case of purely viscous flow in a circular duct, *then the average kinetic energy is twice that given by* Equation 29.

Example 20. If 2000 cfm of air flows through an 8 in. diameter circular duct, find the average kinetic energy per pound of air.

Solution. The cross-sectional area of the duct is 0.349 sq ft; hence the average flow velocity is 5730 fpm. If the velocity were uniform over the section, the average kinetic energy would be $(5730 \div 13{,}430)^2 = 0.182$ Btu per pound. But it is more likely that the actual distribution of velocity would approximate that characteristic of viscous flow; hence the average kinetic energy would be more nearly $2 \times 0.182 = 0.364$ Btu per pound.

Gravitational Energy

The potential energy due to elevation Z (feet) above any convenient datum is simply $Z \div 778.3$ Btu per pound of fluid. In the case of moist air,

$$PE = \frac{\overline{Z}\,(1 + W)}{778.3} \tag{30}$$

where

PE = average potential energy, Btu per pound dry air.
$\overline{Z}$ = average elevation, feet.
W = humidity ratio, pound water per pound dry air.

Enthalpy

No further discussion of enthalpy is required. It may be well to emphasize, however, that enthalpies have been figured on the basis of one pound of dry air.

Heat and Shaft Work

Between any two sections 1 and 2 in an apparatus through which steady flow occurs, there may be heat *absorbed* from outside, ${}_1q_2$, Btu per pound of dry air, and shaft work *removed* to outside, ${}_1l_2$, Btu per pound of dry air. If heat is actually rejected to outside, ${}_1q_2$ is intrinsically negative; and if shaft work is actually put in from outside ${}_1l_2$, is intrinsically negative.

Steady-flow Energy Equation

A complete energy accounting takes the form of Equation 31 which is usually referred to as the steady-flow energy equation.

$$_1q_2 = (h_2 + \overline{KE}_2 + \overline{PE}_2) - (h_1 + \overline{KE}_1 + \overline{PE}_1) + {}_1l_2 \tag{31}$$

where

$_1q_2$ = heat *added* from outside *between* sections 1 and 2, Btu per pound dry air.
h_2 = enthalpy of the mixture *at* section 2, Btu per pound dry air.
$\overline{KE}_2$ = average kinetic energy *at* section 2, Btu per pound dry air.
$\overline{PE}_2$ = average potential energy *at* section 2, Btu per pound dry air.
h_1 = enthalpy *at* section 1, Btu per pound dry air.
$\overline{KE}_1$ = average kinetic energy *at* section 1, Btu per pound dry air.
PE_1 = average potential energy *at* section 1, Btu per pound dry air.
$_1l_2$ = shaft work withdrawn *between* sections 1 and 2, Btu per pound dry air.

In Equation 31 all quantities are *per pound of dry air.* If Equation 28 is used in computing average kinetic energy, the result will be in Btu per pound of dry air if v is taken as volume per pound of dry air. If Equation 29 is used, multiplication by $(1 + W)$ as in Equation 30 is required though this is a refinement seldom justified.

Properties of saturated steam are given in Table 8 and for additional definitions refer to Chapter 47.

U. S. STANDARD ATMOSPHERE

The so-called U. S. Standard Atmosphere is an essential standard of reference in aeronautics and as such has become important to the air conditioning engineer who frequently has to simulate atmospheric conditions at high altitudes in connection with aeronautical research. In defining this standard it is first assumed that temperature T varies linearly with altitude Z above sea level, at any rate up to the lower limit of the isothermal layer at 35,332 ft. Thus,

$$T = T_0 - 0.0019812\ Z \tag{32}$$

or

$$\frac{dT}{dZ} = -0.0019812 \text{ (degree Centigrade per foot)} \tag{33}$$

The second assumption is the validity of the perfect gas laws, namely,

$$Pv = BT \tag{34}$$

An horizontal disc of air having unit cross-sectional area (1 sq ft) and vertical thickness dZ (ft) weighs dZ/v (lb). This accounts for the difference of pressure dP (lb per sq ft) between the upper and lower faces of the disc; hence, using Equation 34

$$dZ = -\frac{BT\,dP}{P} \tag{35}$$

Equations 33 and 35 can be combined to eliminate Z and then integrated to obtain the relation between pressure and temperature, namely,

$$\frac{T}{T_0} = \left(\frac{P}{P_0}\right)^{0.1903} \tag{36}$$

The values $T_0 = 288\ K$ and $P_0 = 29.921$ in. Hg are parts of the definition of the standard atmosphere.

TABLE 10. PRESSURE AND TEMPERATURE FOR ALTITUDES IN U. S. STANDARD ATMOSPHERE

ALTITUDE FEET Z	PRESSURE IN OF HG P	TEMP F t
− 1,000	31.02	+62.6
− 500	30.47	+60.8
0	29.921	+59.0
+ 500	29.38	+57.2
+ 1,000	28.86	+55.4
+ 5,000	24.89	+41.2
10,000	20.58	+23.4
15,000	16.88	+ 5.5
20,000	13.75	−12.3
25,000	11.10	−30.1
30,000	8.88	−47.9
35,000	7.04	−65.8
40,000	5.54	−67.0
45,000	4.36	−67.0
50,000	3 436	−67.0

Values of pressure and temperature are listed in Table 10 for altitudes in the standard atmosphere from −1,000 to 50,000 ft above sea level. Values for altitudes below the lower limit of the isothermal layer conform to Equations 32 and 36. For further explanation, reference (11) should be consulted.

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[11] National Advisory Committee for Aeronautics, Technical Report No 538, 1935.

CHAPTER 2

Physiological Principles

Chemical Vitiation of Air, Physical Impurities in Air, Thermal Interchanges Between the Body and Its Environment, Adaptation to Hot Conditions, Relation of Air Conditioning Needs to Metabolism, Acclimatization, Effective Temperature Index, Physiological Objectives of Heating and Ventilation, Summer Comfort, Influence of Humidity, Influence of Air Movement, The Four Vital Factors

VENTILATION is defined in part as *the process of supplying or removing air by natural or mechanical means to or from any space.* (See Chapter 47). The word in itself implies quantity but not necessarily quality. From the standpoint of comfort and health, however, the problem is now considered to be one of securing air of the proper quality rather than of supplying only a given quantity.

The term *air conditioning* in its broadest sense implies control of any or all of the physical or chemical qualities of the air. When applied to comfort air conditioning, however, the A.S.H.V.E. Code of Minimum Requirements of Comfort Air Conditioning[1] defines it "as the process by which simultaneously the temperature, moisture content, movement and quality of the air in enclosed spaces intended for human occupancy may be maintained within required limits. If an installation cannot perform all of these functions, it shall be designated by a name that describes only the function or functions performed."

CHEMICAL VITIATION OF AIR

Under the artificial conditions of indoor life, the air undergoes certain physical and chemical changes which are brought about by the occupants themselves. The oxygen content is somewhat reduced, and the carbon dioxide slightly increased by the respiratory processes. Organic matter, which is usually perceived as odors, comes from the nose, mouth, skin and clothing. The temperature of the air is increased by the metabolic processes, and the humidity raised by the moisture emitted from the skin and lungs.

Contrary to old theories, the usual changes in oxygen and carbon dioxide are of no physiological concern because they are too small to produce appreciable effects even under the worst conditions of normal human occupancy. Only in such unusually air-tight enclosures as submarines and some air raid shelters need the increase in carbon dioxide and the reduction in oxygen be considered. The amount of carbon dioxide in air is often used as an index of odors of human origin, but the information it affords rarely justifies the labor involved in making the observation[2, 3]. Little is known of the identity and physiological effects of the organic matter given off in the process of respiration. The former belief that the discomfort experienced in confined spaces was due to some toxic volatile matter in the expired air is now limited, in the light of numerous researches, to the much less dogmatic view that the presence of such a

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[1]Code of Minimum Requirements for Comfort Air Conditioning (A.S.H.V E. TRANSACTIONS, Vol. 44, 1938, p. 27). Reprints of this code are available at $ 10 a copy

[2]A S.H.V.E. RESEARCH REPORT No 959—Indices of Air Change and Air Distribution, by F. C. Houghten and J. L Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol 39, 1933, p. 261).

[3]A.S.H.V.E. RESEARCH REPORT No. 1031—Ventilation Requirements, by C. P. Yaglou, E C. Riley and D J. Coggins (A.S.H.V.E. TRANSACTIONS, Vol 42, 1936, p. 133).

substance has not been demonstrated. The only certain fact is that expired air may be odorous, offensive, and capable of producing loss of appetite and a disinclination for physical activity. Objectionable body odors have the same effects. These reasons, whether esthetic or physiological, call for the introduction of a certain minimum amount of clean outdoor air to dilute odors from any source, including cooking and other processes, to a concentration which is not objectionable.

In certain industrial processes toxic fumes and gases may be produced whose removal by local exhaust ventilation is essential for the protection of human health. In the ordinary occupied spaces harmful chemical impurities may be contributed from certain types of cooking and heating appliances, including carbon monoxide from imperfect combustion, which may be a serious hazard to life and health.

The control of offensive or hazardous concentrations of chemical vitiation of the air is frequently brought about by the ventilating engineer through dilution. This method has found application when the source of contamination is the human occupant and not something of a particularly hazardous character.

In the case of vitiation by a few hazardous gases such as carbon monoxide from heating and cooking and certain industrial processes, no satisfactory chemical treatment for the elimination of the impurity has been found The only really satisfactory solution is elimination at the source; or, if this is impossible, reduction to a safe concentration by dilution. In the case of contamination by other matter, including volatile vapors and gases, chemical treatment for the removal or chemical reduction of the impurities has been made available through air cleaning methods, which are discussed in Chapter 28 on Air Cleaning Devices. The A.S.H.V.E Research Technical Advisory Committee on Removal of Atmospheric Impurities has outlined[4] means for the reduction of atmospheric impurities.

When the only source of contamination is the human occupant, the minimum quantity of outdoor air needed appears to be that required to remove objectionable body odors, or tobacco smoke. The concentration of body odor in a room, in turn, depends upon a number of factors, including the dietary and hygienic habits of the occupants (frequently reflecting their socio-economic status), the outdoor air supply, air space allowed per person, odor adsorbing capacity of air conditioning processes, and temperature and relative humidity. Perception of odor, like the perception of most of the other senses, is proportional to the logarithmic function of the intensity of the stimulus; or in the case of odors, the perception has been found to vary as the logarithmic function of the odor intensity, or inversely with the logarithmic function of the available air determined by the outdoor air supply and the air space per person.

The relation between air supply and occupancy has been reported by the *Harvard School of Public Health*[5] and the A.S.H.V.E. Research Laboratory[6]. The findings from the Harvard study are given in Table 1. Outdoor air requirements for removal of objectionable tobacco smoke odors are not accurately known but sources of information available and practice indicates the need of 15 cfm per person or more.

The total quantity of outside air to be circulated through an enclosure is governed by both chemical and physical considerations. The physical

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[4]Report of the A S.H.V E. Technical Advisory Committee on Removal of Atmospheric Impurities (Programs of the Research Technical Advisory Committees, June, 1941, p 7)

[5]Loc. Cit. Note 3

[6]Loc Cit Note 2.

requirements for controlling temperature, air distribution and air velocity usually predominate. Other factors which must be taken into consideration include the type and usage of the building, locality, climate, height of rooms, floor area, window area, extent of occupancy, and the operation of the system distributing the air supply. Frequently, some of these factors, particularly the need for air movement and good distribution, may be satisfied by recirculation of inside air rather than by outside air.

It will be noted that, with adequate air space, the rate of air change

TABLE 1. MINIMUM OUTDOOR AIR REQUIREMENTS TO REMOVE OBJECTIONABLE BODY ODORS[7]

Type of Occupants	Air Space per Person Cu Ft	Outdoor Air Supply CFM per Person
Heating season with or without recirculation. Air not conditioned		
Sedentary adults of average socio-economic status	100	25
Sedentary adults of average socio-economic status	200	16
Sedentary adults of average socio-economic status	300	12
Sedentary adults of average socio-economic status	500	7
Laborers	200	23
Grade school children of average class	100	29
Grade school children of average class	200	21
Grade school children of average class	300	17
Grade school children of average class	500	11
Grade school children of poor class	200	38
Grade school children of better class	200	18
Grade school children of best class	100	22
Heating season. Air humidified by means of centrifugal humidifier. Water atomization rate 8 to 10 gph. Total air circulation 30 cfm per person		
Sedentary Adults	200	12
Summer season. Air cooled and dehumidified by means of a spray dehumidifier Spray water changed daily Total air circulation 30 cfm per person.		
Sedentary Adults	200	<4

indicated in Table 1 is from 10 to 30 cfm per person. In rooms occupied by only a few persons such a rate of air change will be automatically attained in cold weather by normal leakage around doors and windows while it can easily be secured in warm weather by the opening of windows. With a space allotment of 400 cu ft per person, only 1½ air changes per hour are necessary to provide an air change of 10 cfm per person. This space allotment is essential for other reasons.

Therefore, in the ordinary dwelling with adequate cubic space allotment, no special provision for controlling chemical purity of the air is necessary (aside from removal of fumes from heating appliances). For

[7]Loc Cit Note 3, p 156

such conditions, the control of air temperature is the major factor to be considered.

In more crowded rooms (large offices, large workrooms, auditoriums), the whole picture changes. Cubic space per person is less and the size of the room makes it impossible to admit untempered outside air without drafts. Here, mechanical ventilation is essential, but as will be noted in a later paragraph, it is even more essential for thermal than for chemical reasons. It is control of the thermal properties of the air in order to effect the removal of the heat produced by human bodies, rather than dilution of chemical poisons, which must govern practice.

The Code of Minimum Requirements for Comfort Air Conditioning[8] prescribes definite minimum requirements which should be familiar to the designing engineer. It should be emphasized, however, that the provisions of the code aim to provide *minimum*, rather than *adequate*, requirements.

Notwithstanding the rapid advance in the field of air conditioning during the past few years, there still remain those who believe in a superior, stimulating quality of outdoor air (particularly country, mountain and seashore air) under ideal weather conditions, as compared with properly conditioned air. While this point of view is usually held by persons not intimately acquainted with the complicated factors involved they nevertheless carry some weight. It is apparent, however, to anyone acquainted with the factors involved and the conditions effecting comfort that in modern air conditioning, like in most other branches of engineering, modern science makes it possible to control the phenomena of nature for the service and comfort of man beyond any possibilities found in nature itself. When the requirements for optimum comfort as determined by the atmospheric environment are known (and the comprehensive studies to date lead us to believe that they are known at least to a high degree), the air conditioning engineer can supply these requirements indoors to the same perfection as may accidentally be found at times outdoors and keep them under control. The freedom of movement, action and thought, together with the variability of stimulae experienced by persons under ideal conditions in the country, mountains or seashore, and the psychological effect of these *wide open spaces* undoubtedly have some stimulating effect, which when compared with the monotony of confinement indoors even in the most favorable atmospheric environment accounts for the contrast. Ultra-violet light and ionization have been suggested but the evidence so far is inconclusive or negative[9].

Ozone has been used with success for the destruction of micro-organisms (molds) in meat packing establishments and the like; and where considerable amounts of organic effluvia are present it may be useful as a deodorant. For ordinary ventilation practice, however, neither of these purposes can be usefully attained, since the concentration of ozone

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[8]Loc Cit Note 1

[9]A S H V.E. RESEARCH REPORT No 921—Changes in Ionic Content in Occupied Rooms, Ventilated by Natural and Mechanical Methods, by C P Yaglou, L C Benjamin and S P Choate (A S H V E TRANSACTIONS, Vol. 38, 1932, p 191) A S H.V.E RESEARCH REPORT No 965—Physiologic Changes During Exposure to Ionized Air, by C. P Yaglou, A. D. Brandt and L C Benjamin (A S.H.V.E TRANSACTIONS, Vol 39, 1933, p 357). A.S H V E RESEARCH REPORT No 985—Diurnal and Seasonal Variations in the Small Ion Content of Outdoor and Indoor Air, by C. P Yaglou and L C Benjamin (A S H V E TRANSACTIONS, Vol 40, 1934, p. 271). The Nature of Ions in Air and Their Possible Physiological Effects, by L B Loeb (A S H V E TRANSACTIONS, Vol 41, 1935, p. 101). The Influence of Ionized Air upon Normal Subjects, by L P. Herrington (*Journal Clinical Investigation*, 14, January, 1935). The Effect of High Concentrations of Light Negative Atmospheric Ions on the Growth and Activity of the Albino Rat, by L P. Herrington and Karl L Smith (*Journal Ind. Hygiene*, 17, November, 1935) Subjective Reactions of Human Beings to Certain Outdoor Atmospheric Conditions, by C.-E. A. Winslow and L P. Herrington (A.S H V E. TRANSACTIONS, Vol 42, 1936, p. 119).

necessary for effectiveness would be likely to transcend the limit of comfort in ordinary occupied rooms. While ozone has been used in the treatment of certain diseases, there is no evidence that it has a tendency to increase comfort or to benefit health under conditions of normal human occupancy. The allowable concentrations in the breathing zone are very small, between 0.01 to 0.05 ppm parts of air. These are much too small to influence bacteria. Higher concentrations are associated with a pungent unpleasant odor and considerable discomfort to the occupants. One part per million causes respiratory discomfort, headaches, depression, a lowering of the metabolic rate and may even lead to coma[10].

PHYSICAL IMPURITIES IN AIR

Dust particles of various types, when present in considerable concentrations, produce an irritant effect upon the mucous membranes of nose and throat and may be associated with high prevalence of acute respiratory diseases such as bronchitis and pneumonia. Dust which contains free silica has special harmful effects, causing a primary disease of the lungs (silicosis) and predisposing the victim in a high degree to tuberculosis. These, however, are special problems of industrial hygiene which will not be discussed in detail in this chapter.

A certain part of the dissemination of disease in confined spaces is caused by the emission of pathogenic organisms from infected persons. Droplets sprayed into the air in talking, coughing, sneezing, etc., do not all fall immediately to the ground within a few feet from the source, as was formerly believed. The large droplets fall, but minute droplets less than 0.1 mm in diameter evaporate to dryness before they fall the height of a man. Nuclear residues from such sources, which may contain infective organisms drift long distances with the air currents and the virus may remain alive long enough to be transmitted to other persons in the same room or building. Droplet nuclei have been recovered from cultures of resistant micro-organisms a week after innoculation into a tight chamber of 3000 cu ft capacity, although the majority of disease germs died out within a few hours[11]. Practical epidemiological evidence indicates that the danger of such atmospheric transmission is slight with the bacterial diseases but may be appreciable with the diseases caused by the much smaller viruses. Avoidance of overcrowding is a major factor in avoiding such dangers. The microbic concentration in the atmosphere may be reduced by air change, but since the rate of contamination may be great at local points over short periods of time the hazardous concentration may not be eliminated quickly enough and may even be spread over larger areas by local drafts. The possibility of sterilizing the air supply at the source, or destroying the micro-organisms at their point of admission to the air by ultra-violet light is being studied and offers considerable promise[12].

While in some instances it may be possible to reduce the physical impurities of the air by dilution from a non-contaminated source, such non-contaminated sources are rarely available. Frequently the outside air contains a higher concentration of physical impurities than that within an enclosure. Therefore, it is usually desirable to reduce the concentra-

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[10]*The British Medical Journal*, Editorial, June 25, 1932, p 1182. See also Loc. Cit. Note 9.

[11]Air-Borne Infection and Sanitary Air Control, by W. F. Wells (*Journal Industrial Hygiene*, November, 1935).

[12]Sanitary Ventilation in Wards, by W. F. Wells (*Heating and Ventilating*, April, 1939, p. 26). Measurement of Sanitary Ventilation, by W F. Wells (*American Journal of Public Health*, Vol. 28, 1938, p 343).

tion of physical impurities by air cleaning methods, as discussed in Chapter 28.

THERMAL INTERCHANGES BETWEEN THE BODY AND ITS ENVIRONMENT

The importance of the thermal factors arises from the profound influence which they exert upon body temperature, comfort and health. Body temperature depends upon the balance between heat production and heat loss. The heat resulting from the oxidation which occurs within the body (metabolism) maintains the body temperature well above that of the surrounding air in a cool or cold environment. At the same time, heat is constantly lost from the body by radiation, convection and evaporation. Since, under ordinary conditions, the body temperature is maintained at its normal level of about 98.6 F, the heat production must be balanced by the heat loss.

In conditioning air for comfort and health it is necessary to know the rate of sensible and latent heat liberation from the human body, which in conjunction with other heat loads (see Chapters 4, 6 and 7) determines the capacity required for proper conditioning. The data in common use are those of the A.S.H.V.E. Research Laboratory[13].

The fundamental thermodynamic processes concerned in heat interchanges between the body and its environment may be described by the equation:

$$M = \pm S + E \pm R \pm C \qquad (1)$$

where

M = rate of metabolism.
S = rate of storage.
E = rate of evaporative heat loss.
R = rate of radiative heat loss or gain.
C = rate of convective heat loss or gain.

Factor M, the rate of metabolism, is always positive. The storage, S, may be either positive or negative, depending upon whether heat is being stored or given off, accompanied by a rise or fall in body temperature. Under ordinary circumstances (when the dew-point of the air is below the body surface temperature) the evaporation loss, E, is always positive; that is, heat from metabolism supplies this loss. R and C are positive when the surface temperature of the body is above that of the walls and air, and negative when it is cooler.

The human body possesses remarkable powers of adaptation to a narrow range of atmospheric conditions around an ideal optimum where storage is zero, and metabolism and skin and tissue temperature are at optimum values. As skin temperature and body-tissue temperature rise or fall above or below an optimum, complex adaptive mechanisms come into play, chiefly associated with redistribution of blood supply between the skin and deeper tissues (in a cold environment) and with sweat

[13]A S H V E. RESEARCH REPORT No 830—Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F C Houghten, W. W Teague, W E. Miller and W P Yant (A.S H V.E TRANSACTIONS, Vol 35, 1929, p 245) Thermal Exchanges Between the Human Body and Its Atmospheric Environment, by F. C. Houghten, W. W Teague, W E Miller and W. P. Yant (*American Journal of Physiology*, Vol 88, 1929, p 386) A S H V E RESEARCH REPORT No 908—Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by F C. Houghten, W. W Teague, W E. Miller and W P. Yant (A S.H V E TRANSACTIONS, Vol 37, 1931, p. 541) Thermal Exchanges Between the Bodies of Men Working and the Atmospheric Environment, by F C. Houghten, W. W. Teague, W. E Miller and W. P. Yant (*American Journal of Hygiene*, Vol XIII, 1931, No. 2, p. 415). A.S H.V.E RESEARCH REPORT No. 1106—Air Conditioning in Industry, by W. L. Fleisher, A E Stacey, Jr, F C Houghten and M B Ferderber (A S H V E TRANSACTIONS, Vol 45, 1939, p 59).

secretion (in a hot environment). Under cold conditions, the need for more heat and shivering or other muscular movements increase metabolism, which is, again, a reaction favorable to temperature regulation; but under very hot conditions metabolism also rises and this reaction is obviously harmful and indicates a balance of purely chemical over physiological control[14] resulting from increased chemical reactions with rise in temperature. In other words, it represents a breakdown or failure of the entire regulative processes. These reactions are governed by nervous or chemical stimuli from both skin and internal tissues. Nerves from the skin, for example, carry the sense impressions to the brain and the response comes back over another set of nerves, the motor nerves, to the musculature and to all the active tissues in the body, including the endocrine glands. In this way, a two-sided mechanism controls the body temperature by (1) regulation of internal heat production (chemical regulation), and (2) regulation of heat loss by means of automatic variation in the rate of cutaneous circulation and the operation of the sweat glands (physical regulation). The reactions involved in cold and in hot environments are on the whole radically different in nature. The mechanisms of adjustment involved are extremely complex and while they are receiving considerable study a complete understanding of their operation is still lacking.

In a certain middle range, normal and easy physiological regulation occurs by slight changes in the distribution of blood carrying heat between the skin and the inner organs, resulting in slight changes in the body surface temperature, and hence, in the rate of heat dissipation to the atmosphere. This easy balance gives a sensation of comfort. Above this range, the blood capillaries near the surface become dilated, allowing more blood and heat to flow into the skin, and thus increase its temperature and consequently its heat loss. If this method of cooling is not in itself sufficient, the stimulus is extended to the sweat glands which allow water to pass through the surface of the skin. This method of cooling is the most effective of all, as long as the vapor pressure and dew-point temperature of the air are sufficiently low to allow for evaporation. In high humidities, where the difference between the dew-point temperature of the air and body temperature is not sufficient to allow rapid evaporation, increase in heat loss may be had by increasing air movement. The body, under hot conditions, is in the *zone of evaporative regulation*, and for moderately extreme conditions perfect balance between heat production and heat loss may be attained, although at the cost of considerable discomfort.

In a cold environment, where environmental conditions are such as to remove heat too rapidly, the organism adapts in some degree by constricting the blood vessels leading to the surface, thereby reducing the blood flow and heat available for dissipation to the environmental surroundings. This adaptation is, however, partial and incomplete, and in an environment too cold for the clothing worn the temperature of the body tissues may fall, with accompanying discomfort and ultimate danger of serious chill. The process may go on for hours. The individual may move about and increase metabolism through muscular activity and thus balance the excessive heat demand of the environment, or he may reduce the loss by greater insulation of his body in the form of clothing.

Some of these phenomena which are important are shown graphically in Fig. 1. The dotted curves, from a study at the *John B. Pierce Labora-*

[14]Loc Cit. Note 13.

tory of Hygiene[15], are for subjects lightly clothed in a semi-reclining position and give the relation between the dry-bulb temperature of the environment (with about 45 per cent relative humidity) and the metabolic rate, the rate of heat dissipation by radiation and convection combined, and the latent heat loss due to evaporation of, perspiration and moisture from the respiratory tract. The smooth line curves, from the work of the A.S.H.V.E. Research Laboratory[16], give the same relationships for healthy, male subjects (18 to 24 years of age), seated at rest and normally clothed for winter-heated and air conditioned occupancy. The data for the semi-reclining subject also include the rate of heat storage (either positive or negative) due to a rise or fall in body temperature. For the normally clothed subjects a curve gives the total heat loss (that is, the

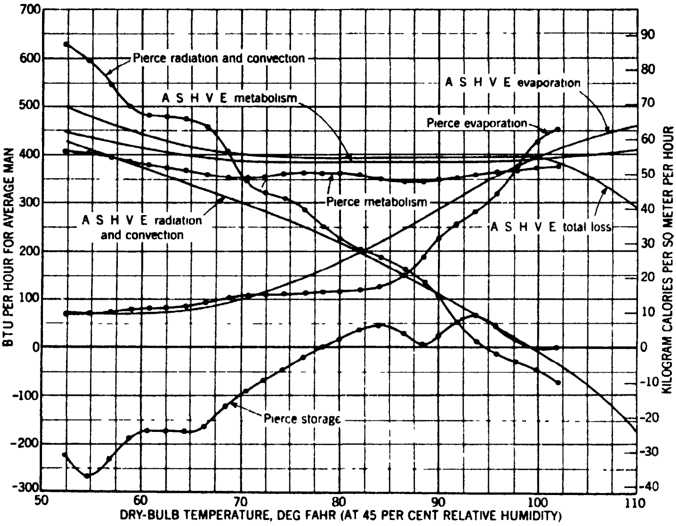

FIG 1 RELATION BETWEEN METABOLISM, STORAGE, EVAPORATION, RADIATION PLUS CONVECTION, AND OPERATIVE TEMPERATURE FOR THE CLOTHED SUBJECT

sum of the radiation, convection and evaporative losses). Here, storage is given by the difference between the metabolism and total heat loss.

The small difference between the metabolic or heat production rates for the two types of subjects may be accounted for by the difference in activity. Heat exchange between the body and the environment by radiation and convection is greater for the lightly clothed subject, both for cool conditions where there is considerable heat loss, and for very warm conditions where there is a sensible transfer from the atmosphere to the body. The two curves for evaporative loss serve to show how physiological control uses evaporation of perspiration to maintain equilibrium, particularly at high temperatures. Below about 75 F for the normally clothed subject, and below about 85 F for the lightly clothed subject, evaporation loss is minimal and probably due to uncontrolled evaporation from the relatively dry skin and from the respiratory tract. Above these temperatures control is had by availability of perspiration for evaporation. The difference in the curves above 75 F is probably largely determined by the difference

[15]A S.H V E. RESEARCH REPORT NO 1107—Recent Advances in Physiological Knowledge and Their Bearing on Ventilation Practice, by C -E. A. Winslow, T Bedford, E F. DuBois, R W. Keeton, A Missenard, R. R Sayers and C. Tasker (A S H V E TRANSACTIONS, Vol 45, 1939, p 111)

[16]Loc Cit Note 13

in clothing and activity. Above temperatures from 95 to 100 F (probably that of the average outside surface of the clothed body) radiation and convection combined changes from positive to negative, and slightly above this temperature even the greatly increased latent heat loss ceases to suffice to take care of the rate of heat production and the negative radiation and convection loss, and storage or a rise in body temperature is the consequence. Above this range, even though there is inability to dissipate heat rapidly enough, metabolism actually increases, which may be accounted for by the predominance of the purely chemical laws of increased chemical reaction with rise in temperature, over physiological control, and indicates the point where a breakdown in thermal equilibrium begins. For higher temperatures life can only survive to the point where these accelerated processes will result in a rise in body temperature to the limiting level of from 106 to 108 F.

Air movement is an important factor in increasing heat loss by either convection or evaporation. The result is accomplished through removal of hot humid air from near the body surface and replacing it with cooler and relatively drier air. This is an important factor in maintaining thermal equilibrium either for persons at rest or at work in hot, humid conditions. For conditions in the comfort zone and below, excessive velocities (particularly localized drafts) should be avoided since differential cooling of one area of the body may produce surprisingly unpleasant reactions in quite different parts of the body. In a recent experiment[17] it was shown that the application of an ice pack to an area of 60 sq cm on the back of the neck for 15 min caused a drop of 17 F in the skin temperature of the fingers and that this low temperature of the fingers persisted for one hour after the ice pack was removed.

ADAPTATION TO HOT CONDITIONS

It will be observed from Fig. 1 that a zone ranging from about 70 to 80 F, at 45 per cent relative humidity (or from 66 to 74 deg ET) the body is adequately able to maintain equilibrium through control of radiation and convection losses combined, and evaporative loss. This corresponds to the zone over which the largest percentage of persons find optimum comfort. For higher temperatures, up to an upper limit in the neighborhood of 100 F, at 45 per cent relative humidity (or approximately 87 deg ET) control is obtained through availability of perspiration on the body surface for evaporation. While a fair degree of temperature equilibrium is maintained over this range it is nevertheless obtained with considerable discomfort.

Studies at the *John B. Pierce Laboratory of Hygiene*[18] have indicated the relation between discomfort and the degree of wetting of the body surface by perspiration for lightly clothed subjects in a semi-reclining position, and the investigators there have designated this as the *zone of evaporative regulation*. Work of the A.S.H.V.E. Research Laboratory[19] over the past two decades has made available data on the relation between sensible perspiration and the atmospheric environment for normal persons at rest and at work.

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[17]The Relative Influence of Radiation and Convection Upon the Temperature Regulation of the Clothed Body, by C.-E. A Winslow, L. P. Herrington and A P Gagge (*American Journal of Physiology*, Vol 124, October, 1938, p. 51).

[18]Relations Between Atmospheric Conditions, Physiological Reactions, and Sensations of Pleasantness, by C.-E. A. Winslow, L. P. Herrington and A P. Gagge (*American Journal of Hygiene*, Vol 26, July, 1937, p. 102). The Reactions of the Clothed Human Body to Variations in Atmospheric Humidity, by C.-E. A Winslow, L. P. Herrington and A P. Gagge (*American Journal of Physiology*, Vol. 124, December, 1938 p. 692).

[19]Loc. Cit. Note 13.

For colder conditions below 70 F, with 45 per cent relative humidity (or about 66 deg ET) thermal equilibrium is maintained, first, by the amount of clothing worn; second, and to a smaller extent, by limiting availability of heat at the surface by decreasing peripheral blood circulation, which results in a drop in skin temperature; and third, by an increase in the metabolic rate. Here again, while thermal equilibrium is fairly well maintained, any drop in the skin temperature is accompanied by a certain degree of discomfort.

Studies at the A.S.H.V.E. Research Laboratory[20] and elsewhere[21] during the past two decades have made available a mass of information dealing with the physiological effects of hot atmospheres on workers and means to alleviate the distress and hazards associated therewith. This interest has been termed *air conditioning in industry*, or *the effects of hot atmospheres in industrial hygiene*, and is a growing factor in air conditioning applications. Table 2 gives some of the physiological responses of men at rest

TABLE 2. PHYSIOLOGICAL RESPONSES TO HEAT OF MEN AT REST AND AT WORK[a]

Effective Temp	Actual Cheek Temp (Deg Fahr)	Men at Rest			Men at Work 90,000 ft-lb of Work per Hour			
		Rise in Rectal Temp (Deg Fahr per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Weight by Perspiration (Lb per Hr)	Total Work Accomplished (Ft-Lb)	Rise in Body Temp (Deg Fahr per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Wt by Perspiration (Lb per Hr)
60	....	....	....	....	225,000	0.0	6	0.5
70	....	0.0	0	0.2	225,000	0.1	7	0.6
80	96.1	0.0	0	0.3	209,000	0.3	11	0.8
85	96.6	0.1	1	0 4	190,000	0.6	17	1.1
90	97.0	0.3	4	0.5	153,000	1.2	31	1.5
95	97 6	0 9	15	0.9	102,000	2.3	61	2.0
100	99.6	2.2	40	1 7	67,000	4 0[b]	103[b]	2.7[b]
105	104.7	4.0	83	2.7	49,000	6.0[b]	158[b]	3.5[b]
110	....	5.9[b]	137[b]	4.0[b]	37,000	8.5[b]	237[b]	4.4[b]

[a]Data by A S.H.V.E. Research Laboratory.
[b]Computed value from exposures lasting less than one hour.

and at work, to hot environments. Recent physiological studies[22] indicate that frequent and continued exposure of workers to hot environments results in not only violent but subtle physiological derangement, affecting

[20]A S.H.V E. RESEARCH REPORT NO. 651—Some Physiological Reactions to High Temperatures and Humidities, by W. J. McConnell and F C Houghten (A.S.H V E TRANSACTIONS, Vol 29, 1923, p. 129). A S H.V E RESEARCH REPORT NO. 672—Further Study of Physiological Reactions, by W. J. McConnell, F C. Houghten and F. M Phillips (A S H V E. TRANSACTIONS, Vol 29, 1923, p 353). A.S H V.E RESEARCH REPORT NO. 690—Air Motion, High Temperatures and Various Humidities—Reactions on Human Beings, by W. J. McConnell, F C. Houghten and C P. Yaglou (A S.H.V E TRANSACTIONS, Vol 30, 1924, p. 167). A S H.V E. RESEARCH REPORT NO. 718—Work Tests Conducted in Atmospheres of High Temperatures and Various Humidities in Still and Moving Air, by W. J. McConnell and C. P. Yaglou (A S H V E TRANSACTIONS, Vol. 31, 1925, p. 101). A S H V E. RESEARCH REPORT NO. 719—Basal Metabolism Before and After Exposure to High Temperatures and Various Humidities, by W. J. McConnell, C. P. Yaglou and W. B. Fulton (A.S.H.V E. TRANSACTIONS, Vol 31, 1925, p. 123). A.S.H.V.E RESEARCH REPORT NO. 908—Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by F. C Houghten, W. W. Teague, W. E Miller and W. P. Yant (A S.H V.E. TRANSACTIONS, Vol 37, 1931, p 541) A S H V.E. RESEARCH REPORT No 1106—Air Conditioning in Industry—Physiological Reactions of Individual Workers to High Effective Temperatures, by W. L. Fleisher, A E Stacey, Jr, F. C Houghten and M. B. Ferderber (A S H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 59). A S H.V E. RESEARCH REPORT No 1153—Seasonal Variation in Reactions to Hot Atmospheres, by F. C. Houghten, A. A. Rosenberg and M. B Ferderber (A.S H V.E. TRANSACTIONS, Vol. 46, 1940, p. 185). Physiologic Effects of Hot Atmospheres, by F. C. Houghten, M. B Ferderber and A. A. Rosenberg (*Industrial Medicine*, January, 1940, p. 7).

[21]A S.H.V E RESEARCH REPORT NO. 1151—The Peripheral Type of Circulatory Failure in Experimental Heat Exhaustion, by R. W. Keeton, F K. Hick, Nathaniel Glickman and M. M. Montgomery (A S H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 157).

[22]A S.H V.E. RESEARCH REPORT NO. 1153—Seasonal Variation in Reactions to Hot Atmospheres, by F. C. Houghten, A. A. Rosenberg and M. B. Ferderber (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p 185)

the leucocyte count of the blood and other factors dealing with man's mechanism of defense against infection.

Another of the deleterious effects of high temperatures is that the blood is diverted from the internal organs to the surface capillaries, in order to serve in the process of cooling. This affects the stomach, heart, lungs and other vital organs, and it is suggested that the feeling of lassitude and discomfort experienced is due in part to the anaemic condition of the brain. The stomach loses some of its power to act upon the food, owing to a diminished secretion of gastric juice, and there is a corresponding loss in the antiseptic and antifermentive action which favors the growth of bacteria in the intestinal tract[23]. These are considered to be the potent factors in the increased susceptibility to gastro-intestinal disorders in hot summer weather.

In warm atmospheres, particularly during physical work, a considerable amount of chloride is lost from the system through sweating. The loss of this substance may lead to attacks of cramps, unless the salts are replaced in the drinking water. In order to relieve both cramps and fatigue, it is recommended that 6 g of sodium chloride and 4 g of potassium chloride be added to a gallon of water[24].

The deleterious physiologic effects of high temperatures exert a powerful influence upon physical activity, accidents, sickness and mortality. Both laboratory and field data show that physical work in warm atmospheres is a great effort, and that production falls progressively as the temperature rises. The incidence of industrial accidents reaches a minimum at about 68 F, increasing above and below that temperature. Sickness and mortality rates increase progressively as the temperature rises.

The need of air conditioning for workers in hot industries is growing rapidly and this should become an important field for the air conditioning engineer. The hot conditions may be remedied by any of the recognized comfort cooling applications. The choice of the type of system and cycle to be used in a given instance must be determined by the air conditioning engineer after a study of surrounding conditions.

In some hot industries where a small number of workers are engaged in spaces of large volumetric capacity the worker himself, rather than the entire environment, can be cooled by either placing him in a small cooled and ventilated booth, by blowing cooled air over him, or by circulating cooled air through a loose-fitting suit [25]

RELATION OF AIR CONDITIONING NEEDS TO METABOLISM

The major objective of heating and ventilation is to balance heat losses from the human body. The basic factor is metabolism. The desirable environment, from the standpoint of heat loss, depends directly on the heat produced in the body and this heat may be over ten times as great when a man is exercising violently as when he is reclining and at rest. Therefore, there is no absolute optimum of air temperature or other environmental conditions, which will meet all cases. With moderate, (45 per cent) relative humidity and minimum air movement, an air temperature of 80 F has been found ideal for the lightly clothed subject at rest in a semi-reclining position, while normally clothed, healthy persons

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[23] Influence of Effective Temperature upon Bactericidal Action of Gastro-Intestinal Tract, by Arnold and Brody (*Proceedings Society Exp. Biol. Med.*, Vol. 24, 1927, p 832)

[24] Some Effects of High Air Temperatures Upon the Miner, by K N Moss (*Transactions Institute of Mining Engineers*, Vol. 66, 1924, p. 284)

[25] A.S.H.V.E. RESEARCH REPORT No 1188—Local Cooling of Workers in Hot Industry, by F. C Houghten, M. B. Ferderber and Carl Gutberlet (A S.H.V.E TRANSACTIONS, Vol 47, 1941, p 403)

have been found comfortable at 72 and 77 F with 45 per cent relative humidity (or 67 and 71 deg ET, respectively) for winter and summer conditions. In factories where light work is performed in summer time, the ideal has been found to be about 76 F. For children (who have a high

TABLE 3. RELATION BETWEEN METABOLIC RATE AND ACTIVITY[a]

Activity	Hourly Metabolic Rate for Avg Person or Total Heat Dissipated, Btu per hour	Hourly Sensible Heat Dissipated, at 79 F, Btu per Hour	Hourly Latent Heat Dissipated, at 79 F, Btu per Hour	Moisture Dissipated per Hour per Person	
				Grains	Pounds
Basal	291	145	145	978	0.140
Seated at Rest	384	225	159	1072	0.153
Reading Aloud (Seated)	420	225	195	1315	0.188
Standing at Rest	431	225	206	1389	0.198
Hand Sewing (Seated)	441	225	216	1457	0.208
Knitting 23 stitches per minute on Sweater	462	225	237	1598	0.228
Dressing and Undressing	468	225	243	1639	0.234
Tailor	482	225	257	1733	0.248
Singing	486	225	261	1760	0.251
Office Worker Moderately Active	490	225	265	1787	0.255
Light Work Standing	549	225	324	2185	0.312
Typewriting Rapidly	558	225	333	2246	0.321
Ironing with 5 lb iron	570	225	345	2326	0.332
Dishwashing—Plates, Bowls, Cups and Saucers	600	225	375	2529	0.361
Clerk Moderately Active Standing at Counter	600	225	375	2529	0.361
Book Binder	626	225	401	2704	0.386
Shoemaker	661	225	436	2940	0.420
Sweeping Bare Floor 38 Strokes per Minute	672	229	443	2987	0.427
Pool Player	680	230	450	3055	0.434
Walking 2 mph, Light Dancing	761	250	511	3446	0.492
Light Metal Worker (at Bench)	862	277	585	3945	0.564
Painter of Furniture (at Bench)	876	280	596	4019	0.574
Carpenter	954	307	647	4363	0.623
Restaurant Serving	1000	325	675	4552	0.650
Pulling Weight	1041	335	708	4774	0.682
Walking 3 mph	1050	339	711	4795	0.685
Walking 4 mph, Active Dancing, Roller Skating	1390	452	938	6325	0.904
Walking Down Stairs	1444	467	977	6588	0.941
Stone Mason	1490	485	1005	6777	0.968
Bowling	1500	490	1010	6811	0.973
Man Sawing Wood	1800	590	1210	8160	1.166
Swimming	1986	......	......	......	......
Running 5.3 mph	2268	......	......	......	......
Walking 5 mph	2330	......	......	......	......
Walking Very Fast 5.3 mph	2580	......	......	......	......
Walking Upstairs	4365	......	......	......	......
Maximum Exertion Different People	3000-4800	......	......	......	......

[a]These metabolic rates were compiled by the A S H V.E Research Laboratory from actual tests, from other authoritative sources, and from estimates based upon various considerations Division of the total heat dissipation into latent and sensible rates is based on actual test data and on various considerations for metabolic rates up to 1250 Btu per hour, and extrapolated for higher rates Values for total heat dissipation for a person at rest apply for a dry-bulb temperature range from approximately 70 to 90 F; for other than rest conditions the values apply for a similar but lower temperature range Below these temperature ranges metabolic rates and total rates of heat dissipation increase, while above these ranges metabolic rates increase slightly and total heat dissipation rates decrease rapidly Division of total dissipation rates into sensible and latent heat holds only for a dry-bulb temperature of 79 F For lower temperatures, sensible heat dissipation increases and latent heat decreases, while for higher temperatures the reverse is true

metabolism) at school, in winter clothing, 70 F has been considered correct; while in a gymnasium, 55 F has been recommended.

The wide variations in metabolic activities with which the engineer must be prepared to cope and the influence of such variations in metabolism on the heat load contributed by the human body to the environment

are given in Table 3 and in Figs. 2, 3 and 4. It should be noted that metabolism and heat dissipation values are proportional to the body surface areas of the persons considered, and that the data referred to are only for persons having an average surface area of 19.5 sq ft or that of the average adult American male, 5 ft 8 in. in height and weighing 150 lb, and will therefore not apply to many audiences made up largely of women or younger persons. Fig. 5, taken from the work of Du Bois[26], gives the relation of body surface area to height and weight, and may serve to

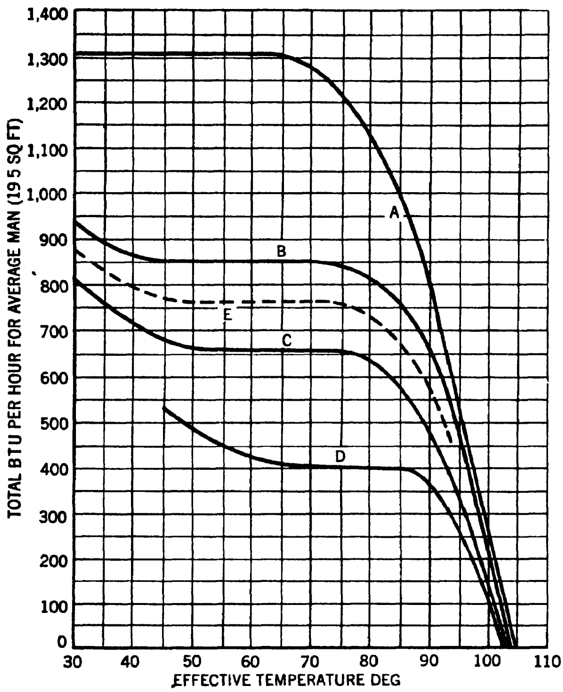

FIG. 2. RELATION BETWEEN TOTAL HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE FOR STILL AIR[a]

[a]Curve *A*—Persons working so as to have a metabolic rate of 1310 Btu per hour Curve *B*—Persons working so as to have a metabolic rate of 850 Btu per hour Curve *C*—Persons working so as to have a metabolic rate of 660 Btu per hour Curve *D*—Persons seated at rest, or with a metabolic rate of 400 Btu per hour. Curves *B* and *D* based on test data covering a wide temperature range Curves *A* and *C* based on test data at an Effective Temperature of 70 deg and extrapolation of Curves *B* and *D* All curves are averages of values for high and low relative humidities which apply with satisfactory accuracy for most considerations For special problems requiring a higher degree of accuracy see more detailed A S H V E Research Laboratory reports

correct the data for audiences other than adult men. The curves in the figures are based on certain averages of test results with different humidities, and are sufficiently accurate for most practical applications. Where greater precision in the applications of the results is required, or for extreme variations in temperature and humidity, the reports[27] covering the A.S.H.V.E. Laboratory work may be consulted.

The curves in Figs. 2, 3 and 4, and proper interpolation between these

[26]DuBois, D. and E. F. (*Archives of Internal Medicine*, 1916, Vol. 18, p. 865).
[27]Loc. Cit. Note 13.

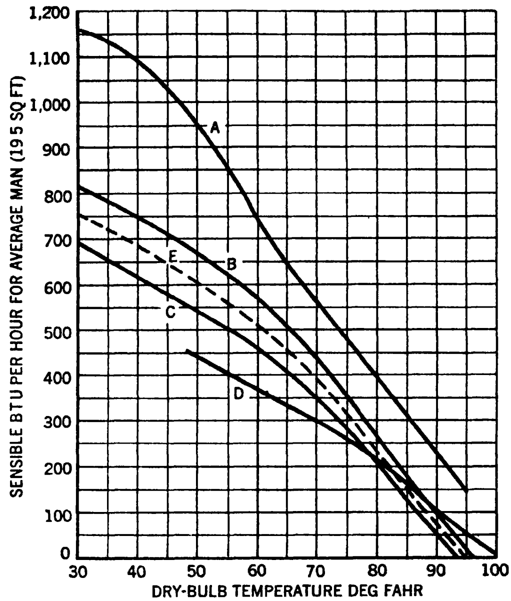

FIG. 3. RELATION BETWEEN SENSIBLE HEAT LOSS FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AIR[a]

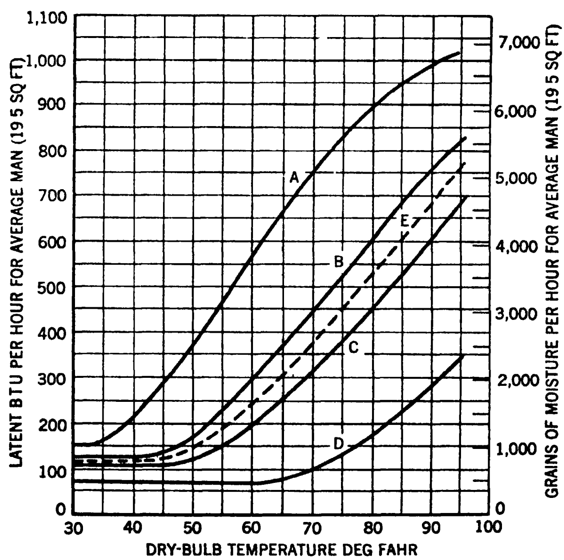

FIG. 4. LATENT HEAT AND MOISTURE LOSS FROM THE HUMAN BODY BY EVAPORATION, IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS[a]

[a]Loc. Cit. See footnote a, Fig. 2.

curves make it possible to apply the data to persons engaged in any type of work or physical activity, providing the resulting metabolic rate is known. As an example, if it is found that a certain type of work results in a metabolic rate of approximately 760 Btu per hour for an average person working in an atmosphere of 70 ET, then his total rate of heat dissipation to atmospheres of various temperatures will be approximately as given by the broken-line curve in Fig. 2. The broken line curves in Figs. 3 and 4 give the rate of sensible and latent heat dissipation of the person for different dry-bulb temperatures.

ACCLIMATIZATION

Acclimatization and the factor of psychology are two important influences in air conditioning which cannot be ignored. The first is man's ability to adapt himself to changes in air conditions; the second is an intangible matter of habit and suggestion.

Some persons regard the unnecessary endurance of cold as a virtue.

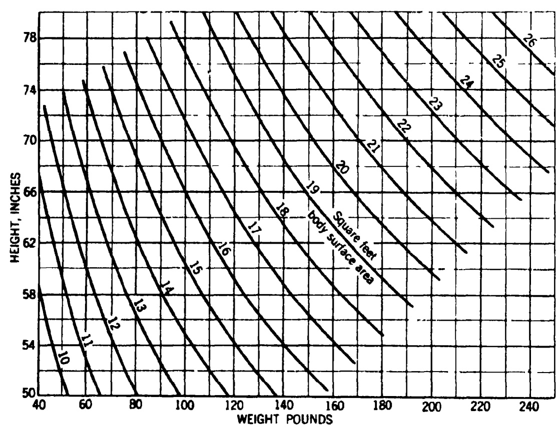

FIG. 5. CHART FOR DETERMINING SURFACE AREA OF INDIVIDUALS FOR HEIGHT AND WEIGHT GIVEN

They believe that the human organism can adapt itself to a wide range of air conditions with no apparent discomfort or injury to health. In the light of present knowledge of air conditioning these views are not justified. Acclimatization to extreme conditions involves a strain upon the heat regulating system and interferes with the normal physiologic functions of the human body. Thousands of years in the heat of Africa do not seem to have acclimatized the Negro to a temperature exceeding 80 F. The same holds true of northern races with respect to cold, although the effects are mitigated by artificial control. An environment averaging 64 F for the 24-hour period has been indicated as associated with minimal mortality[28].

Within limits, however, there does occur a definite adaptation to external temperature level. People and animals raised under conditions of tropical moist heat stand chilling poorly as they are unable quickly to increase internal combustion to keep up the body temperature. For this

[28]Civilization and Climate, by Ellsworth Huntington, Yale University Press, 1928.

reason they have trouble standing the cold, stormy weather of the temperate zones, and when exposed to it are very susceptible to respiratory infections. Likewise, people living in cool climates suffer greatly in the moist heat of the tropics until their adaptive mechanism has been adjusted.

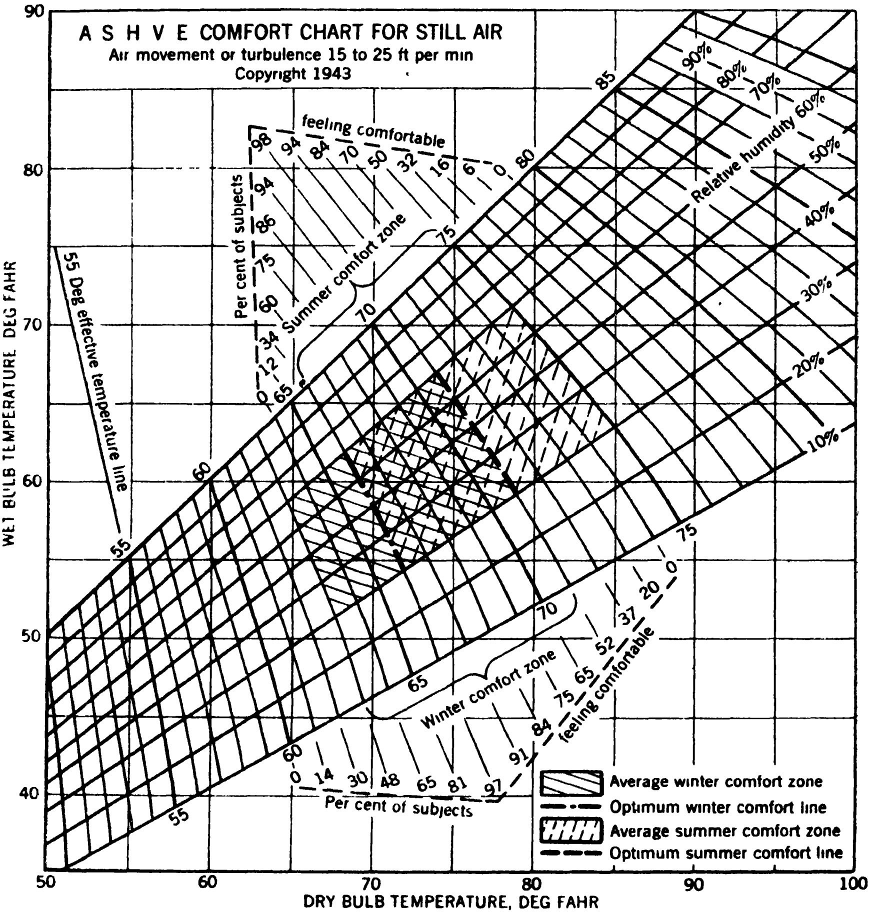

FIG. 6. A.S H.V.E. COMFORT CHART FOR STILL AIR

Note —Both summer and winter comfort zones apply to inhabitants of the United States only. Application of winter comfort line is further limited to rooms heated by central station systems of the convection type. The line does not apply to rooms heated by radiant methods. Application of summer comfort line is limited to homes, offices and the like, where the occupants become fully adapted to the artificial air conditions The line does not apply to theaters, department stores, and the like where the exposure is less than 3 hours. The optimum summer comfort line shown pertains to Pittsburgh and to other cities in the northern portion of the United States and Southern Canada, and at elevations not in excess of 1000 ft above sea level An increase of one deg ET should be made approximately per 5 deg reduction in north latitude.

Within a few years, however, they find themselves reacting as natives to the new environment.

The adaptive level changes somewhat with the season[29]. There are also marked differences between the sexes. In the cold zone the thickness of

[29]The Reactions of the Clothed Human Body to Variations in Atmospheric Humidity, by C.-E. A. Winslow, L. P. Herrington and A P. Gagge (*American Journal of Physiology*, Vol. 124, December, 1938, p. 692).

thermal insulating tissues of women is almost double that of men, although the sensory responses to cold are similar. In the hot zone, the threshold of sweating and skin temperature levels are higher for women.

Finally, the thickness and insulating value of the clothing worn is an important factor in the determination of the comfort level.

EFFECTIVE TEMPERATURE INDEX

Sensations of warmth or cold depend, not only on the temperature of the surrounding air as registered by a dry-bulb thermometer, but also upon the temperature indicated by a wet-bulb thermometer, upon air movement and upon radiation effects. Dry air at a relatively high temperature may feel cooler than air of considerably lower temperature with a high moisture content. Air motion makes any moderate condition feel cooler. Radiation to cold or from warm surfaces is another important factor under certain conditions.

Combinations of temperature, humidity, and air movement which induce the same feeling of warmth are called thermo-equivalent conditions. A series of studies[30] at the A.S.H.V.E. Research Laboratory, Pittsburgh, established the equivalent conditions met with in general air conditioning work. This scale of thermo-equivalent conditions not only indicates the sensation of warmth, but also to a considerable degree determines the physiological *effects* on the body induced by heat or cold. For this reason, it is called the *effective temperature* scale or index, and it denotes sensory heat level.

Effective temperature is an empirically determined index of the degree of warmth perceived on exposure to different combinations of temperature, humidity, and air movement. It was determined by trained subjects who compared the relative warmth of various air conditions in two adjoining conditioned rooms by passing back and forth from one room to the other.

The numerical value of the index for any given air conditions is fixed by the temperature of slowly moving (15 to 25 fpm air movement) saturated air which induces a like sensation of warmth or cold. Thus, any air condition has an effective temperature of 60 deg, when it induces a sensation of warmth like that experienced in slowly moving air at 60 deg saturated with moisture. The *effective temperature index* cannot be measured directly but is determined by the dry- and wet-bulb temperature observations and by reference to the Comfort Chart (see Figs. 6, 8 and 9) or tables. The relation of winter and summer sensations of comfort to wet- and dry-bulb temperature at low air movement is shown in Fig. 6 This chart, published by an A.S.H.V.E. Technical Advisory Committee[31], is based on research prior to 1932. Later studies by the A.S.H.V.E. Research Laboratory indicate somewhat higher temperatures for winter comfort, while Fig. 7 shows some variation in the requirements for comfort in summer cooled and air conditioned space. Relations between moisture content and various dry-bulb temperatures to wet-bulb readings and effective temperatures are depicted in Fig. 8. Effective temperatures

[30] A S H.V.E RESEARCH REPORT NO. 673—Determination of the Comfort Zone, by F. C Houghten and C. P. Yaglou (A S H V E. TRANSACTIONS, Vol. 29, 1923, p. 361). A S H V.E RESEARCH REPORT No 691—Cooling Effect on Human Beings Produced by Various Air Velocities, by F. C. Houghten and C P Yaglou (A S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p 193). A S H.V E RESEARCH REPORT NO 717—Effective Temperature with Clothing, by C. P Yaglou and W. E. Miller (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p 89). A S H.V.E RESEARCH REPORT No. 755—Effective Temperature for Persons Lightly Clothed and Working in Still Air, by F. C. Houghten, W. W Teague and W E. Miller (A S H V.E. TRANSACTIONS, Vol. 32, 1926, p. 315).

[31] How to Use the Effective Temperature Index and Comfort Charts, by C P. Yaglou, W H Carrier, Dr E V. Hill, F. C. Houghten and J. H. Walker (A.S.H.V.E TRANSACTIONS, Vol. 38, 1932, p 410).

for various combinations of wet- and dry-bulb temperatures and air movement are given in Fig. 9.

A long series of studies have been made to determine the optimum effective temperature for comfort of normal persons in both winter and summer air conditioned space, in different geographical regions and for different age groups of men and women. A group of these studies[32] was made between 1935 and 1940 by the A.S.H.V.E. Laboratory in Pittsburgh, and in several metropolitan districts of the United States in cooperation with the managements of offices employing large numbers of workers. Some of the results are shown in Fig. 7. Taking all of these studies together, women of all age groups studied indicate an average effective temperature for comfort 1.1 deg higher than for men. All men and

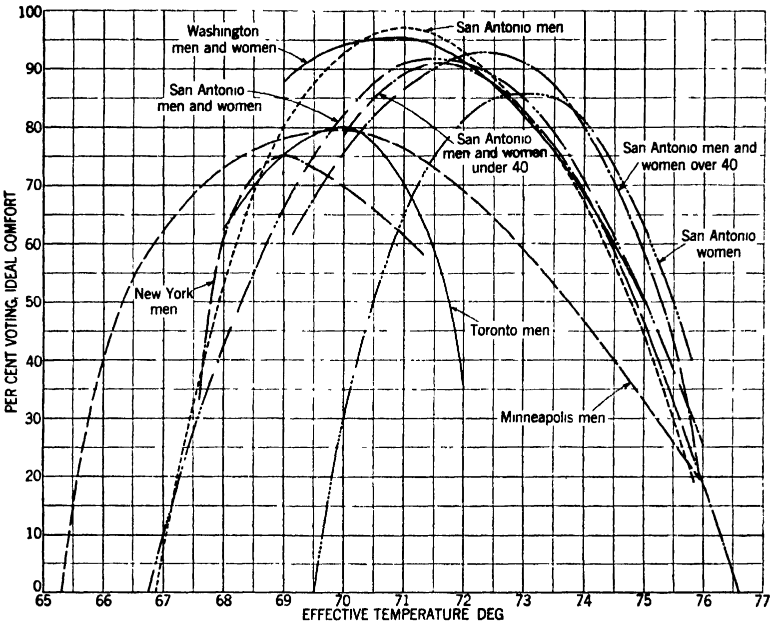

Fig. 7. Relation Between Effective Temperature and Percentage Observations Indicating Comfort

women, beyond the age of 40 years, show an average desire for 0.9 deg ET higher than those below this age; while the men and women, respectively, beyond 40 desired effective temperatures of 0.8 and 1.2 deg higher than those below 40. The persons serving in all of these studies were representative of office workers clothed for air conditioned space in the summer

[32] A S H.V E Research Report No 1035—Comfort Standards for Summer Air Conditioning, by F C Houghten and Carl Gutberlet (A S H V E Transactions, Vol 42, 1936, p 215) A S H V E Research Report No 1055—Cooling Requirements for Summer Comfort Air Conditioning, by F C Houghten, F E Giesecke, C. Tasker and Carl Gutberlet (A S H V.E. Transactions, Vol 43, 1937, p 145). A S.H V E. Research Report No 1088—Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A B. Newton, F C Houghten, Carl Gutberlet and R W. Qualley (A S.H V E Transactions, Vol. 44, 1938, p 337) Cooling Requirements for Summer Comfort Air Conditioning in Toronto, by C Tasker (A.S H V E. Transactions, Vol. 44, 1938, p 549) A S H V E Research Report No. 1127—Reactions of Office Workers to Air Conditioning in South Texas, by A J Rummel, F. E Giesecke, W. H Badgett and A T. Moses (A S H V E. Transactions, Vol 45, 1939, p 459) A S H V E Research Report No 1136—Summer Cooling Requirements in Washington, D C, and Other Metropolitan Districts, by F C Houghten, Carl Gutberlet and Albert A Rosenberg (A S H.V E Transactions, Vol. 45, 1939, p 577). A S H V E Research Report No 1160—Reactions of 745 Clerks to Summer Air Conditioning, by W J McConnell and M Spiegelman (A S H V E Transactions, Vol 46, 1940, p 291).

season and engaged in the customary sedentary activity of office workers.

The 66 deg ET indicated as giving optimum comfort for winter conditions in Fig. 6 and determined prior to 1932, has more recently been checked; first, by occasional observations, and second, by consistent laboratory study[33], indicating that a higher optimum of 67 deg ET was preferred by a majority of the eight male college students whose opinions were ascertained. This can be checked with larger groups when conditions permit.

On the basis of present knowledge, for different geographical regions

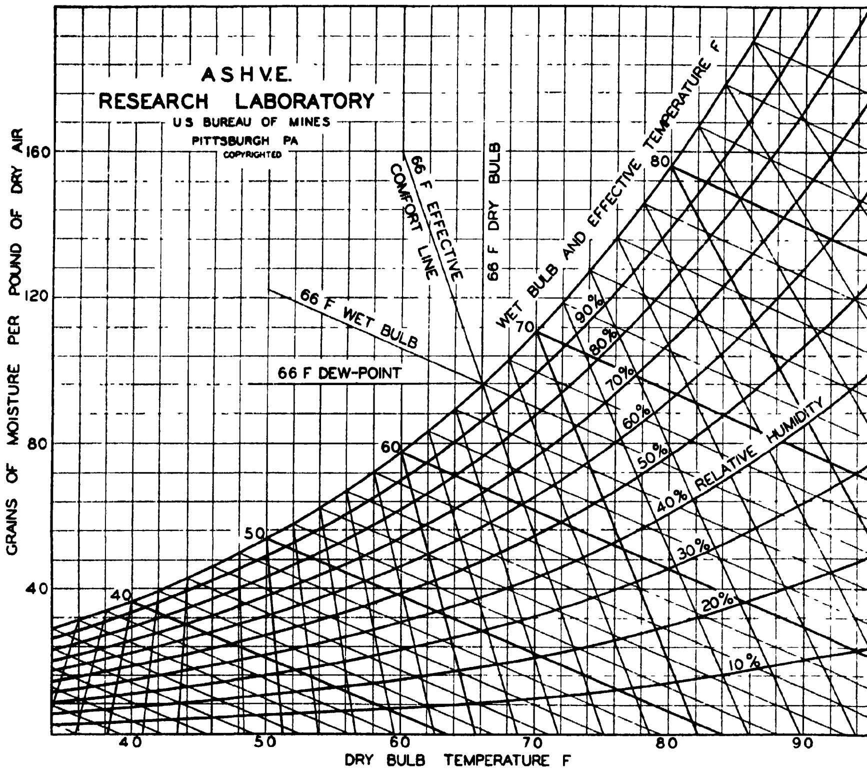

FIG. 8. PSYCHROMETRIC CHART, PERSONS AT REST, NORMALLY CLOTHED, IN STILL AIR

and age groups, the total spread in optimum comfort conditions ranges from a low of 66 deg ET for winter heating and air conditioning, to a high of 73 deg ET for summer cooling and air conditioning.

The spread for summer cooling and air conditioning for optimum comfort is confined entirely to an effective temperature range of from 69 to 73 deg, and it may be presumed that for winter conditioning a like spread would be had; while for inter-seasonal conditions there will be a fluctuation between these two ranges.

Recent studies[34] indicate that for the average individual a temperature

[33]A S H.V.E. RESEARCH REPORT NO. 1172—Radiation as a Factor in the Sensation of Warmth, by F. C. Houghten, S. B. Gunst and J. Suciu, Jr. (A.S.H V.E TRANSACTIONS, Vol 47, 1941, p 93)
[34]Loc Cit. Note 33

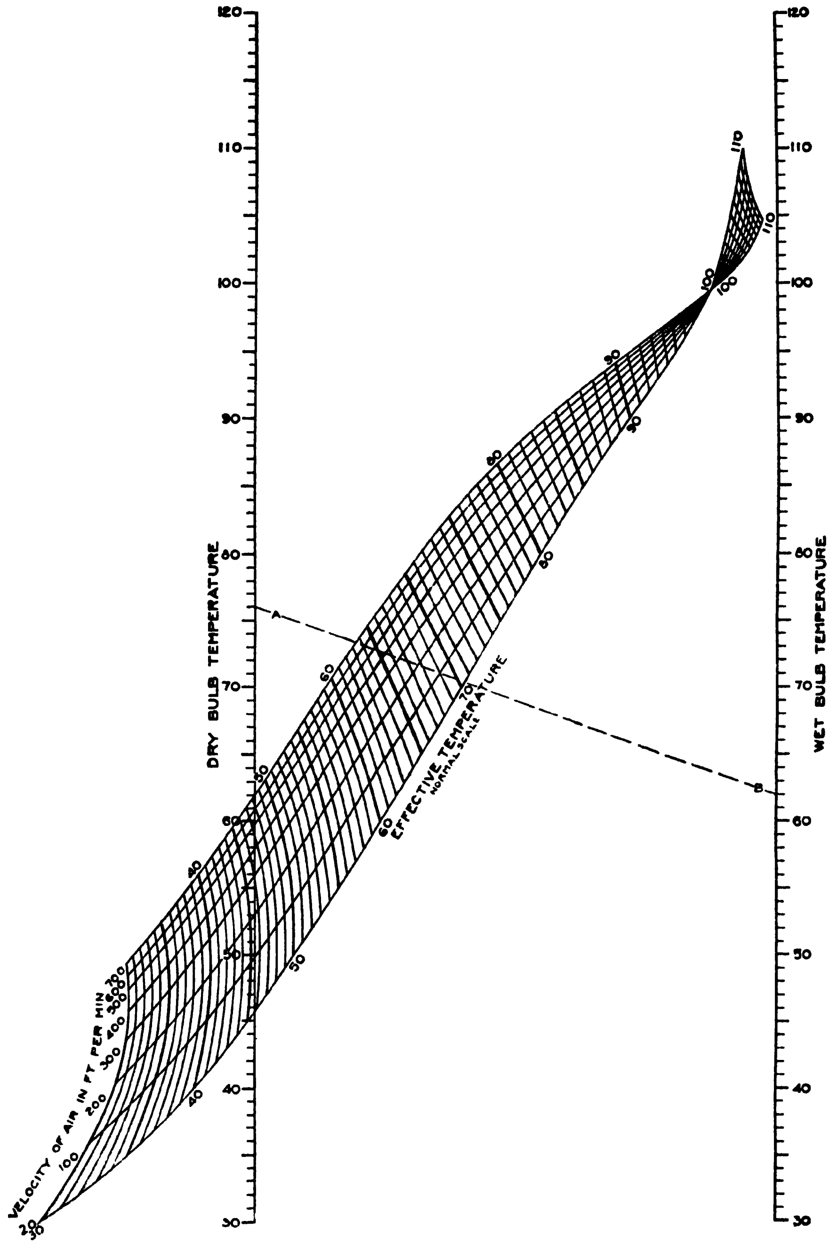

FIG. 9. EFFECTIVE TEMPERATURE CHART SHOWING NORMAL SCALE OF EFFECTIVE TEMPERATURE. APPLICABLE TO INHABITANTS OF THE UNITED STATES UNDER FOLLOWING CONDITIONS:

A Clothing: Customary indoor clothing *B Activity* Sedentary or light muscular work *C. Heating Methods*: Convection type, i.e., warm air, direct steam or hot water radiators, plenum systems

change of about 3 deg ET is required to change a person's sensation from *ideally comfortable* to *cool* or *warm*. From this it may be observed that necessary variations in the effective temperature of air conditioned space for optimum comfort for most persons, need little differentiation for either the winter season or the summer season, and not more than about 4 deg on the average between seasons.

Acclimatization and habits of clothing and diet account for these variations. As a result of a recent analysis[35] of all of the evidence available by the A.S.H.V.E. Technical Advisory Committee on Sensations of Comfort, a variation of 3 deg spread in optimum effective temperature for summer cooling and air conditioning with geographical location has been proposed. However, it should be recognized that variations in sensation of comfort among individuals may be greater for any given location, as shown in Fig. 7, than for variations due to a difference in geographical location. The available information indicates rather clearly that changes in weather conditions over a period of a few days do not acclimate people to a desire for different indoor conditions, but in general, people experiencing low temperatures over an extended period of time become acclimated to desiring lower indoor temperatures, while those experiencing higher temperatures become acclimated to a desire for higher indoor temperatures. It is obvious that a person spending a considerable portion of his time in space conditioned to his comfort will become acclimated to his indoor environment. While few people spend more than a small percentage of the total time within an air conditioned enclosure, there is some evidence that persons experiencing comfort air conditioning a large part of the time tend to become acclimated to about 70 or 71 deg ET.

The entering shock to occupants of summer cooled and air conditioned space may at times be important, and is due to the rapid evaporation of perspiration on the skin, accumulated during the incoming occupant's previous subjection to hot and humid outside conditions. While recent studies[36] show that for healthy individuals this shock is usually a pleasant experience, for others it may result in unpleasant or even harmful chills. This fact should be taken into consideration in applying summer cooling and air conditioning, particularly to spaces where a large number of the occupants may enter for only a short time, 15 min or less. Such occupants may be satisfied with less cooling. For long occupancy very little deviation from the optimum effective temperature is desirable.

Radiation between the occupant of an enclosure and the surfaces of the room itself and objects within the room, including windows, heating and cooling equipment, and other occupants, has an important bearing on the feeling of warmth and may alter to some measurable degree the optimum conditions for comfort indicated previously. Fig. 10[37] shows the necessary elevation in the dry-bulb temperature of the air to compensate for the lower temperature of three of four side-wall surfaces, and indicates that for this condition each degree reduction in the average of the three wall surface temperatures requires an elevation of 0.3 deg in the dry-bulb temperature of the air to compensate. Recent studies by the A.S.H.V.E.

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[35] A.S.H.V.E. RESEARCH PAPER—Comfort with Summer Air Conditioning, by Thomas Chester, N. D. Adams, C. R. Bellamy, G. D. Fife, E. P. Heckel, Dr. W. J. McConnell, F. C. McIntosh, A. B. Newton, B. F. Raber and C. Tasker (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 107).

[36] A.S.H.V.E. RESEARCH REPORT No. 1102—Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley and M. C. W. Tomlinson (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 571).

[37] A.S.H.V.E. RESEARCH REPORT No. 946—Cold Walls and Their Relation to Feeling of Warmth, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 83).

Research Laboratory[38] on the effect of radiation within an enclosure, including the effect of panel heating, indicate that each degree elevation or depression of the mean radiant temperature above or below the air temperature requires about 0.5 deg counterchange in effective temperature of the air. Since the mean radiant temperature of the surroundings is affected by cold, uninsulated walls and windows, particularly single glazed windows, as well as by heating units placed within the room, including panel heaters, these factors must be compensated. Likewise, in densely occupied spaces, such as classrooms, theaters and auditoriums, somewhat lower temperatures may be necessary than those indicated by the comfort line on account of counter-radiation between the bodies of

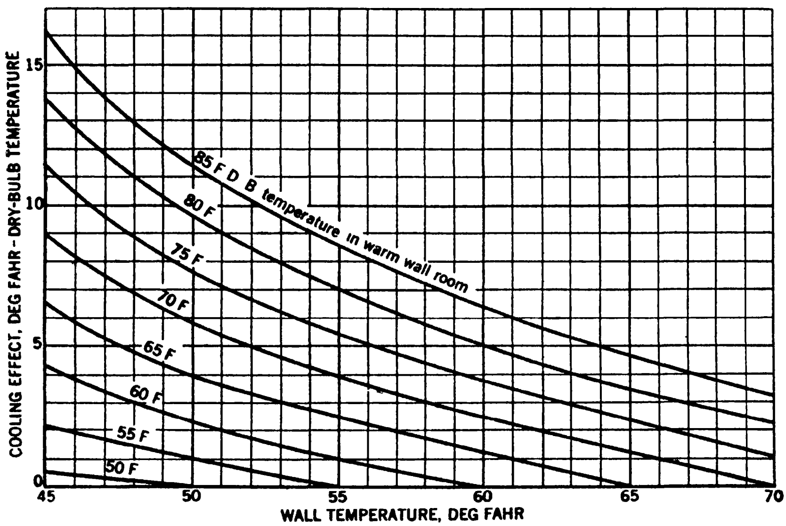

Fig. 10. Cooling Effect of Three Cold Walls in a Small Experimental Room, as Determined by Comparison with Sensations in a Room of Uniform Wall and Air Temperature

occupants in close proximity to each other which also will elevate the mean radiant temperature of the room.

The sensation of comfort, insofar as the physical environment is concerned, is not absolute but varies considerably among certain individuals. Therefore, in applying the air conditions indicated, it should not be expected that all the occupants of a room will feel perfectly comfortable. The curves in Fig. 7 indicate that some persons require temperatures as much as 4 and 6 deg lower and higher than the optimum for the average. In this connection it is of interest to note from the characteristic shape of the curves that in general people will object more quickly to a few degrees drop in temperature from the average optimum than will be the case for the same number of degrees overheating. However, when optimum comfort temperatures are applied in accordance with foregoing recommendations, the majority of the occupants should be comfortable, and it should be expected that there will be a few too warm and a few too cold. These individual differences among the minority should be counteracted by suitable clothing.

[38]Loc. Cit. Note 33.

Satisfactory comfort conditions for persons at work[39] are found to vary depending upon the rate of work and the amount of clothing worn. In general, the greater the degree of activity, the lower the effective temperature necessary for optimum comfort. However, recent work by the A.S.H.V.E. Research Laboratory[40] indicates that under certain conditions moderate activity on the part of a person standing up and moving about may result in a slightly higher optimum effective temperature than for a person seated at rest, because of the larger body surface area exposed to heat elimination and the increase in effective air movement over his body. Where few workers occupy a large space in hot industries, recent work by the A.S.H.V.E. Research Laboratory[41] shows that they may be made reasonably comfortable by blowing relatively small volumes of slightly cooled air over them or through their clothing.

For prematurely born infants, the optimum temperature varies from 100 to 75 F, depending upon the stage of development. The optimum relative humidity for these infants is placed at 65 per cent[42]. No data are yet available on the optimum air conditions for full term infants and young children up to school age. Satisfactory air conditions for these age groups are assumed to vary from 75 to 68 F with natural indoor humidities. For school children, the studies of the New York State Commission on Ventilation place the optimum air conditions at 66 to 68 F temperature with a moderate humidity and a moderate but not excessive amount of air movement[43]. A great number of persons seem to be fairly content with a higher plane of indoor temperature, particularly when the matter of first cost and operating cost of a cooling plant is given due consideration. Recent studies by the University of Illinois[44] in cooperation with the A.S.H.V.E. Committee on Research indicate that effective temperatures as high as 74.5 deg are acceptable in the living quarters of a residence, and while this condition is not representative of optimum comfort it provides sufficient relief in hot weather to be acceptable to the majority of users. It should be emphasized, however, that these are borderline cases that may be acceptable largely in the interest of economy. Comprehensive studies by the A.S.H.V.E. Research Laboratory[45] in cooperation with office staffs in widely distributed regions, including San Antonio, Minneapolis, Washington, D. C., and New York City (see Fig. 7), show conclusively that lower effective temperatures are required for optimum comfort.

PHYSIOLOGICAL OBJECTIVES OF HEATING AND VENTILATION

Aside from the removal of toxic fumes and dusts from heating appliances and industrial processes, the chief task of the heating and ventilating engineer is to keep his clients warm in winter and cool in summer.

For the normally vigorous person, normally clothed, and at rest, an air temperature of 65 F should be provided at knee-height, 18 in. in order to

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[39]A.S.H.V.E. RESEARCH REPORT No. 755—Effective Temperature for Persons Lightly Clothed and Working in Still Air, by F. C. Houghten, W. W. Teague and W. E. Miller (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926, p. 315).

[40]A.S.H.V.E. RESEARCH REPORT No. 1106—Air Conditioning in Industry, by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 59).

[41]Loc. Cit. Note 25.

[42]Application of Air Conditioning to Premature Nurseries in Hospitals, by C. P. Yaglou, Philip Drinker and K. D. Blackfan (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 383).

[43]Ventilation Report of the New York State Commission on Ventilation (E. P. Dutton Co., N. Y., 1923).

[44]A.S.H.V.E. RESEARCH REPORT No. 1012—Study of Summer Cooling in the Research Residence for the Summer of 1934, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 207).

[45]Loc. Cit. Note 32.

prevent chilling of the legs and feet. With some heating systems, this will correspond to 70 F at a 5 ft height. Air temperature may be increased or decreased in order to compensate for deviations of *mean radiant temperature* above or below air temperature.

In rooms occupied by persons of sub-normal vitality, knee-height temperatures must be higher than 65 F. Since dwellings are designed for occupancy by old people and children, the heating system should be able to provide a temperature of 70 F at knee-height under ordinary winter conditions.

The maintenance of such conditions as these in winter depends on three major factors, the heat produced in the occupied space, the heat absorbed from the sun and the heat loss through the walls, floor and ceiling of the structure to cold air and earth. Taking these up in the order in which they occur, in planning a new structure it is essential to remember the important effect of orientation and fenestration of the building with respect to the absorption of radiant heat from the sun. It has recently been shown that, in the vicinity of New York, effective sun-heat on a wall facing south is almost five times as great in winter as in summer, but on a wall facing west-north-west it is six times as great in summer as in winter[46]. The orientation of the same one-story house (in a laboratory model) was changed from a position in which its principal rooms faced northwest to a position in which these rooms (with rearranged and slightly increased fenestration) faced west of south. This change decreased average summer sun-heat to one-ninth and increased average winter sun-heat to fourfold of its value with the original orientation.

The choice between the various methods of heating depends, of course, on many engineering and other factors. From the standpoint of human health and comfort, however, it is important to minimize floor-ceiling differentials as far as possible to avoid hot heads and cold feet. Furthermore, when the problem is a heating one, low air movement is desirable, since air temperature must be raised to balance the cooling effect of air motion.

Where occupants are closely congregated, a new problem comes in, the removal of the excess heat and water vapor produced by the human body. If the temperature of such a space be correctly adjusted when the occupants enter, it will rise steadily during the period of occupancy as a result of the heat given off by the occupants. Of the 400 Btu per hour given off 100 would perhaps be lost in evaporation, leaving 300 Btu per person per hour to warm the air. In a room containing many persons, the effects of this body heat can be neutralized by admission of outside air without producing unpleasant and dangerous drafts on those near the windows or other inlets. The supply of air before it reaches the occupant should be so tempered as to avoid drafts but in an amount and at a temperature which will remove the sensible heat produced by metabolism. With no heat loss through walls (as in an interior auditorium) this will require 28 cfm of air per person when admitted at 60 F, and an average temperature of 70 F for air leaving the room. Under practical conditions, with one or more cold walls, and a room containing a moderate number of occupants and ample cubic space, window ventilation with deflectors and a gravity exhaust duct may suffice. With crowded rooms, and with any rooms containing 50 or more occupants, forced ventilation will be essential.

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[46]Solar Radiation as Related to Winter Heating in Residences, by H N. Wright (*Report of John B Pierce Foundation*, January 20, 1936)

SUMMER COMFORT

The problem of keeping cool in summer is physiologically as important as keeping warm in winter. In summer the relative humidity of the atmosphere is of great importance, along with air temperature, air movement, and wall temperature. There is no very practical method of cooling walls, but summer comfort can be promoted by modifying any one of the other three factors involved.

Increase of comfort by air movement may be had by the promotion of natural circulation by cross or through ventilation; and here the architect is responsible for providing fenestration which will make such natural ventilation possible. In the lowest cost housing this should be considered as essential.

The direct control of air temperature and humidity is, of course, the ideal solution where the cost of a complete air conditioning equipment can be met. Where this objective is attained, there are two schools of thought concerning the relation between temperature and humidity to be maintained For a given effective temperature some engineers favor comparatively low temperature with a high humidity as this results in a reduction of refrigeration requirements. Preliminary experiments at the A.S.H.V.E. Laboratory[47] would seem to indicate not much impairment of comfort with relative humidities of 70 per cent and somewhat higher, provided the effective temperature is between 70 and 73 deg. Until this subject is fully investigated it is desirable not to exceed 70 per cent and in general relative humidities of 60 per cent or less give more satisfactory results[48].

The second school favors a higher dry-bulb temperature, according to the prevailing outdoor dry-bulb, with a comparatively low humidity (well below 50 per cent), the main purpose being an assumed reduction in temperature contrasts upon entering and leaving the cooled space and to keep the clothing and skin dry. This second scheme requires more refrigeration with the present conventional type of apparatus.

INFLUENCE OF HUMIDITY

The limitation of the comfort zones in Fig 6 with respect to humidity is not final Relative humidities below 30 per cent may prove satisfactory from the standpoint of comfort. In mild weather comparatively high relative humidities seem to be entirely feasible, but in cold weather they are objectionable on account of condensation and frosting on the windows Information on this subject is given in Chapter 4

As to the effects of dryness of the air, *per se*, and irrespective of thermal effects, there is a common belief that dry air in itself exerts a harmful effect upon the skin and mucous membranes; but there is no convincing evidence that the increase of atmospheric moisture which can practically be introduced by humidification into the air of cool occupied rooms has any effect upon health and comfort. All controlled experiments on this point have yielded negative results; and the respiratory membranes of industrial workers exposed to hot moist air are distinctly abnormal compared with those of workers exposed to hot dry air[49].

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[47]A S H.V E RESEARCH REPORT No 1035—Comfort Standards for Summer Air Conditioning, by F C Houghten and Carl Gutberlet (A S H V E TRANSACTIONS, Vol 42, 1936, p 215) A S H V E RESEARCH REPORT No 1055—Cooling Requirements for Summer Air Conditioning, by F C. Houghten, F. E Giesecke, C Tasker and Carl Gutberlet (A S H.V E TRANSACTIONS, Vol 43, 1937, p 145)

[48]Loc Cit. Note 35.

[49]Loc Cit. Note 39.

For the premature infant, a high relative humidity of about 65 per cent is demonstrably beneficial to health and growth[50] until the infants reach a weight of about 5 lb. No such clear-cut evidence exists in the case of adults. In the comfort zone experiments of the A.S.H.V.E. Research Laboratory, the relative humidity was varied between the limits of 30 and 70 per cent approximately, but the most comfortable range has not been determined. In similar experiments at the *Harvard School of Public Health*, the majority of the subjects were unable to detect sensations of humidity (*i.e.*, too high, too low, or medium) when the relative humidity was between 30 per cent and 60 per cent with ordinary room temperatures which is in accord with other studies[51,52].

INFLUENCE OF AIR MOVEMENT

Air movement has a powerful influence on the factors involved in thermal equilibrium of the body. An understanding of the phenomena involved is best obtained through a consideration of the purely physical factors involved in the effect of air movement on heat dissipation from inanimate surfaces by radiation, convection and evaporation. Thermal equilibrium of the human body is more complex because of the physiological control exercised in permitting the body surface temperature to drop when factors influencing heat loss are unavoidably increased without additional clothing and by the making available of perspiration for evaporation.

Air movement does not affect radiation loss, provided there is no change in the skin temperature. However, if there is excessive cooling and lowering of the skin temperature due to increased convection loss, then radiation loss (which varies as the differences of the fourth power of the absolute temperatures of the radiator and receiver) decreases. It has been shown by the work at the *John B. Pierce Laboratory of Hygiene*[53] and by the A.S.H.V.E. Research Laboratory[54] that radiation may thus actually descrease due to air movement in relatively cool atmospheres.

Convection loss from any surface, including that of the clothed body, is greatly increased by air movement, provided the surface temperature remains the same. In cool atmospheres, unless increased clothing is worn, heat loss due to air movement may be accompanied by a drop in body surface temperature.

Heat loss by evaporation is greatly increased by air movement, provided surface temperature and moisture available for evaporation (or the wetness of the surface) are constant. However, since in the human body perspiration is only made available when there is need for increased evaporative heat loss due to reduction in convection loss, increased air movement is accompanied by decreased perspiration and evaporative cooling in moderately cool atmospheres. In very hot atmospheres, particularly with low vapor pressure, evaporative cooling may be increased by air movement so as to increase the maximum temperature level at which thermal equilibrium may be maintained. Results of studies at

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[50]Loc. Cit Note 42

[51]Humidity and Comfort, by W. H Howell (*The Science Press*, April, 1931).

[52]Effect of Variation in Relative Humidity upon Skin Temperature and Sense of Comfort, by U. Miura (*American Journal of Hygiene*, Vol. 13, 1931, p. 432).

[53]Loc. Cit. Note 15.

[54]Loc. Cit. Note 13.

the A.S.H.V.E. Research Laboratory[55] and at the *John B. Pierce Laboratory of Hygiene*[56] give data on the effect of air movement on heat dissipation for normally clothed, standing and seated subjects, and for lightly clothed and semi-reclining subjects, respectively. Fig. 9, resulting from A.S.H.V.E. research, shows the increase in dry-bulb and wet-bulb temperatures for the same effective temperature with air velocities ranging from 20 to 700 fpm.

Air velocities may be used for effective cooling; however, great care must be exercised to avoid drafts due to uneven cooling of the body surface. During the heating season air velocities in excess of 25 to 30 fpm usually give undesirable effects. With summer cooling and air conditioning higher velocities up to 40 or 50 fpm, if properly controlled, seem to give satisfactory conditions free from sensation of draft, while with higher ambient temperatures even higher air velocities may be used. In this connection it may be emphasized that drafts are interpreted[57] as local sensations of excessive coolness, and that even while very high air movement in relatively warm air increases the rate of heat loss from local parts of the body, it may improve the comfort of the occupant, so long as that part of his body surface is not excessively cooled.

THE FOUR VITAL FACTORS

From the preceding discussion it is clear that thermal environment cannot properly be adjusted to the requirements of human health and comfort without control of all the four basic factors:

1. Air temperature (free from radiation effects)
2. Air movement.
3. Humidity.
4. Mean radiant temperature of surrounding surfaces.

According to the recommendations of the Sub-Committee on the Hygiene of Environmental Conditions in the Dwelling[58], it is of great importance in all research studies to make an accurate record of each of the four independent factors governing bodily heat exchanges, temperature, movement and humidity of the air, and mean radiant temperature of the surrounding surfaces. For this purpose the committee suggested in the interest of comparability the use of the following four types of instruments or others yielding similar data:

1. Silvered dry-bulb thermometers or hair-pin thermometers (Bargeboer)
2. Silvered dry Kata thermometers or the hot-wire anemometer.
3. Psychrometer, wet- and dry-bulb, whirling or ventilated.
4. Globe thermometer (Vernon) or the dry resultant thermometer (Missenard).

In this country, the shielded thermometer, or a very fine wire thermocouple, has been found more convenient for determining the true dry-

[55]A.S.H.V.E. Research Report No. 691—Cooling Effects on Human Beings Produced by Various Air Velocities, by F. C. Houghten and C. P. Yaglou (A.S.H.V.E. Transactions, Vol. 30, 1924, p. 193). A.S.H.V.E. Research Report No. 717—Effective Temperature with Clothing, by C. P. Yaglou and W. E. Miller (A.S.H.V.E. Transactions, Vol. 31, 1925, p. 89). A.S.H.V.E. Research Report No. 690—Air Motion, High Temperatures and Various Humidities-Reactions on Human Beings, by W. J. McConnell, F. C. Houghten and C. P. Yaglou (A.S.H.V.E. Transactions, Vol. 30, 1924, p. 167). A.S.H.V.E. Research Report No. 718—Work Tests Conducted in Atmospheres of High Temperatures and Various Humidities in Still and Moving Air, by W. J. McConnell and C. P. Yaglou (A.S.H.V.E. Transactions, Vol. 31, 1925, p. 101).

[56]The Influence of Air Movement on Heat Losses for the Clothed Human Body, by C.-E. A. Winslow, A. P. Gagge and L. P. Herrington (*American Journal of Physiology*, October, 1939, Vol. 127, p. 505).

[57]A.S.H.V.E. Research Report No. 1086—Draft Temperatures and Velocities in Relation to Skin Temperature and Feeling of Warmth, by F. C. Houghten, Carl Gutberlet and Edward Witkowski (A.S.H.V.E. Transactions, Vol. 44, 1938, p. 289).

[58]Housing Commission of the League of Nations, adopted at Geneva, June 25, 1937.

bulb temperature. Fine, hot-wire anemometers are rapidly replacing the use of the Kata thermometer for measuring low air velocities, while some adaptations of the Vernon Globe thermometer, incorporating thermocouples rather than mercury thermometers, have been found more satisfactory and convenient.

Such instruments as these, when properly calibrated and their readings are compared, can be used for determining the four basic physical factors concerned separately or in certain combinations. The results of the four physical measurements thus determined can generally be translated into the terms of any special instrument combining two or more of them.

The work of the A.S.H.V.E. Research Laboratory has made available psychrometric charts with effective temperature scale superimposed thereon, including Figs. 8 and 9, and others[59], while recent studies[60] have indicated the degree to which mean radiant temperature of the surroundings modifies the effective temperature index.

In some instances it may be important to record not only the movement and temperature of the air at various levels, but also the temperature of each wall and window, of the flooring, and of the ceiling, and to measure the total effective radiation of the surroundings in 6 directions; in order to trace the exact causes of defects in the building which have an unfavorable influence on the heat exchanges of its inhabitants. Facts of this type are of great practical importance.

[59]A S H V E Research Report No 691—Cooling Effects on Human Beings Produced by Various Air Velocities, by F C Houghten and C P Yaglou (A S H V.E. Transactions, Vol 30, 1924, p 193)

[60]Loc Cit Note 31

CHAPTER 3

Fundamentals of Heat Transfer

Conduction, Convection, Radiation, Combined Convection and Radiation, Heat-Flow Resistance, Electrical Analogies, Practical Heat Transfer Problems, Unit Conductances for Convection Flow Systems, Radiation Factors or Emissivities, Solutions for Steady-State Conduction Problems

HEAT is that form of energy which is transferred from place to place by virtue of an existing temperature difference. The temperature difference is the potential which causes the transfer, the latter in turn being resisted by the thermal properties of the material combined in a simple term and known as the resistance. Energy exchange associated with mass transfer from place to place (evaporation, condensation, etc.) due to concentration differences will be treated elsewhere such as the section on *cooling tower design* in Chapter 26. The objectives of this chapter are to:

1. Describe the mechanisms and present the rate equations for the different modes of heat transfer.
2. Illustrate the application of the basic concepts to steady-state problems (temperature independent of time or a cyclic variable thereof) by means of several typical solutions of heat transfer systems.

Further applications to specific systems will be found throughout THE GUIDE.

CONDUCTION, CONVECTION AND RADIATION

Thermal conduction is the term applied to the mechanism of heat transfer whereby in fluids the molecules of higher random kinetic energies transmit by direct molecular collision part of their energy to adjacent molecules of lower random kinetic energy. Since the temperature is proportional to the random kinetic energy of the molecules, thermal transfer will occur in the direction of decreasing temperature. The molecules oscillate about a mean position at fairly high velocities and frequencies, but there is no net material flow associated with the conduction mechanism.

In solids the significant mechanism of heat transport is thermal conduction and is ascribed to a transfer mechanism associated with the free electrons[1]. Even in the case of fluids, thermal conduction is significant in the region very close to a solid boundary or wall, for in this region the flow is laminar, parallel with the wall surface, and there are practically no cross currents in the direction of the heat transfer.

Contrasted to the thermal conduction mechanism, *thermal convection* involves energy transfer by eddy mixing and diffusion[2] in addition to conduction. This condition is pictured schematically in Fig. 1 which exhibits transfer from a pipe wall at surface temperature t_s to a colder fluid at a bulk temperature t_f. (Bulk temperature is that which would be attained if the fluid stream were drawn off at a certain section and mixed. It is therefore slightly higher than the lowest temperature in the stream. In the *laminar* sublayer, immediately adjacent to the wall, the heat transfer occurs by thermal conduction; in the *transition* region, which is

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[1]The Metallic State, by H. Hume-Rothery (Oxford Press, 1931).

[2]Absorption and Extraction, by T. K. Sherwood (McGraw-Hill Co , 1937)

called the buffer layer, eddy mixing as well as conduction effects are significant; in the *eddy* or *turbulent* region the major fraction of the transfer occurs by eddy mixing.

In most commercial equipment the main body of the fluid is in turbulent flow, and the laminar film exists at the solid walls only, as shown in Fig. 1. But in cases of low-velocity flow in small tubes, or with viscous liquids such as heavy oil (low Reynolds' numbers), the entire flow may be laminar. In these latter cases there is no transition or eddy region.

When the fluid currents are induced by sources external to the heat transfer region, as for example a pump, the described solid to fluid heat transfer is termed *forced convection*. In contrast, if the fluid currents are internally generated, as a result of non-homogeneous densities arising from the temperature variations, the heat transfer is termed *free convection*.

In the conduction and convection mechanisms heat is transferred as internal energy, *i.e.*, the random molecular kinetic energy associated with the material temperature. For radiant heat transfer, however, a change

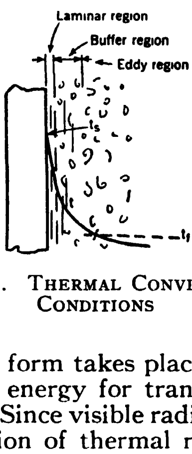

FIG. 1. THERMAL CONVECTION CONDITIONS

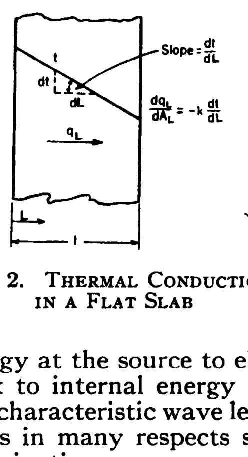

FIG. 2. THERMAL CONDUCTION IN A FLAT SLAB

in energy form takes place from internal energy at the source to electromagnetic energy for transmission, then back to internal energy at the receiver. Since visible radiant energy exhibits characteristic wave lengths, the solution of thermal radiation problems is in many respects similar to the solution of problems in the field of illumination.

The rate of thermal current flow (*i.e.*, rate of heat transfer) corresponding to the transfer mechanisms previously described, may be expressed by three rate equations. These are similar to Ohm's Law for electrical flow, the current flow through a resistance being proportional to the potential difference.

Thermal Conduction Equation

Equation 1 states symbolically that the thermal conduction current per unit transfer area normal to the flow, $(dq)/(dA)$, Btu per hour per square foot, is proportional to the temperature gradient $(dt)/(dL)$, degree Fahrenheit per foot. The proportionality factor is termed the *thermal conductivity*, k, Btu per hour per square foot per degree Fahrenheit per foot of thickness.

$$\frac{dq}{dA} = -k\frac{dt}{dL} \tag{1}$$

The minus sign on the right side of the equation is introduced to indicate positive current flow in the direction of decreasing temperature.

The physical significance of indicated quantities is illustrated further by the schematic diagram Fig. 2.

It should be emphasized that the thermal conductivity used should be expressed in consistent units; either using the inch or foot throughout.

Expressions of conductivity used in the heating field are usually inconsistent in this sense, in that it is customary to refer to the conductivity *per square foot* but for one inch of thickness. This custom has been adopted for the reason that wall thicknesses are usually expressed in inches, whereas if expressed in feet, decimal or fractional thicknesses would result. When dealing with flat walls no complication is involved in using the inconsistent expression of conductivity. However, when curved or spherical walls are considered, considerable complication is involved. Therefore, in this discussion the consistent units of conductivity expressed in *Btu per hour per square foot per degree Fahrenheit for one foot of thickness* are used throughout. *Conductivity values obtained from Chapter 4 or Table 1 in this chapter, which are expressed in inconsistent units, must therefore be converted for use in the calculations of this chapter by dividing by 12.* As an example, the conductivity of brick, expressed in inconsistent units as 5.0 in Table 3 of Chapter 4, becomes 0.42 when used in the calculations of this chapter. Also, it should be emphasized that in order to make the calculations and applications consistent in this chapter, *all dimensions of thickness must be expressed in feet.*

TABLE 1 APPROXIMATE UNIT THERMAL CONDUCTIVITIES OF MISCELLANEOUS MATERIALS[a]

MATERIAL	CONDUCTIVITY, *k* BTU PER HOUR PER SQ FT PER DEG F FOR ONE INCH THICKNESS
Air	0.168
Aluminum	1416 0
Brass (70 − 30)	720.0
Cast-Iron	336 0
Copper	2640 0
Glass	3 6—7.32
Lead	240 0
Nickel	408 0
Soil	2.4—12.0
Steel, mild	312 0
Water, liquid	4 08

[a]Thermal conductivities depend to some extent on temperature The above magnitudes are approximate only Refer to Heat Transmission, by W H McAdams (McGraw-Hill Co , 1942) for additional values

Thermal Convection Equation

$$\frac{dq}{dA} = h_c\,(t_s - t_f) \qquad (2)$$

This rate equation states that the thermal convection current per unit transfer area $(dq)/(dA)$, Btu per hour per square foot, is proportional to the temperature difference, $(t_s - t_f)$ which is the temperature of the surface less that of the fluid[3]. The proportionality factor is termed the *unit convection conductance* (sometimes called the film coefficient for convection), h_c, Btu per hour per square foot per degree Fahrenheit. These convection conditions are illustrated in Fig. 1.

[3]The particular fluid temperature to use for a given system will be noted under the discussion of that system.

The heat transmission by free or natural convection for objects surrounded by air can be conveniently expressed as in Equation 2a:

$$q_c = C\left(\frac{1}{D}\right)^{0.2}\left(\frac{1}{T_{av}}\right)^{0.181}(t_s - t_f)^{1.27} \tag{2a}$$

where

q_c = heat transmission by convection, Btu per square foot per hour.
C = a constant depending upon the surface shape.
D = diameter of pipe or circular duct or height of vertical wall, inches. (Effect of diameter or height becomes constant at 24 in.)
T_{av} = average of wall surface and surrounding air temperature, degrees Fahrenheit absolute.
$t_s - t_f$ = temperature excess between wall surface and surrounding air, degrees Fahrenheit.

For horizontal cylinders, the value of C = 1.016 has been well established by various investigations. For vertical plates, the value of C = 1.394 has been fairly well established. A value of C = 1.79 for horizontal plates warmer than the surrounding air facing upward and 0.89 for horizontal plates warmer than air facing downward is indicated by recent investigations[4].

TABLE 2. HEAT TRANSMISSION BY FREE CONVECTION FOR LARGE VERTICAL SURFACES

Expressed in Btu per square foot per hour

Temp Deg F	Temperature Difference between Body and Surrounding Still Air at 80 F													
	0	10	20	30	40	50	60	70	80	90	100	110	120	130
0	0	4.4	10.4	17.4	25.0	33.2	41.8	50.6	59.9	69.4	79.4	89.2	99.4	109.8
1	0.3	4.9	11.1	18.1	25.8	34.1	42.6	51.5	60.8	70.3	80.4	90.2	100.4	110.9
2	0.6	5.5	11.8	18.9	26.7	34.9	43.5	52.4	61.8	71.3	81.4	91.2	101.5	112.0
3	1.0	6.0	12.5	19.7	27.5	35.7	44.3	53.4	62.7	72.3	82.4	92.2	102.6	113.0
4	1.4	6.6	13.2	20.5	28.3	36.6	45.2	54.3	63.7	73.3	83.3	93.3	103.6	114.1
5	1.8	7.3	13.9	21.2	29.2	37.4	46.1	55.2	64.6	74.3	84.2	94.3	104.7	115.2
6	2.3	7.9	14.6	22.0	30.0	38.3	47.0	56.1	65.6	75.3	85.2	95.3	105.7	116.3
7	2.8	8.5	15.3	22.7	30.8	39.1	47.8	57.1	66.5	76.3	86.2	96.3	106.7	117.3
8	3.3	9.1	16.0	23.5	31.6	40.0	48.7	58.0	67.5	77.4	87.2	97.4	107.8	118.4
9	3.8	9.7	16.7	24.3	32.4	40.9	49.7	59.0	68.4	78.4	88.2	98.4	108.8	119.5

The heat transmission by free convection from vertical walls ***24 in. or more in height*** is given in Table 2 as calculated from Equation 2a for ambient air temperature of 80 F. The values in Table 2 will not be changed appreciably by a considerable change in air temperature for a given temperature excess. For instance, a change in air temperature from 80 to 40 F will increase the heat transmission given in Table 2 by only 1.3 per cent.

Table 2 can also be used for calculating the free convection rate of transmission for various commercial shapes such as pipes and ducts. These calculations are simplified by the use of the factors in Tables 3 and 4. Table 3 gives factors by which the values in Table 2 must be multiplied to obtain the free convective transfer from various shapes whose characteristic dimensions are 24 in. or over, and Table 4 gives the factors to be used in conjunction with the factors in Table 3 for obtaining the free convection from Table 2 for pipes and ducts *whose characteristic dimensions are less than 24 in.*

[4]The Transmission of Heat by Radiation and Convection, by Griffith and Davis (*Special Report* No. 9, 1922, *Department of Scientific and Industrial Research*, His Majesty's Stationery Office, London, England).

For example, the free convection transfer from a 3 in. O.D. horizontal cylinder for a temperature difference of 40 F = 25.0 × 0.73 × 1.52 = 27.7 Btu per square foot per hour.

Problems in either forced convection or natural convection may be solved by the simple first-power equation if the convection coefficient is expressed as a unit conductance:

$$q_c = h\,A\,(t_1 - t_2) \tag{2b}$$

where

q_c = heat transmission by convection, Btu per hour
A = surface area, square feet.
$t_1 - t_2$ = temperature difference between the surface and the fluid degrees Fahrenheit.
h = unit conductance, from Table 5, Btu per square foot per hour per degree Fahrenheit temperature difference.

TABLE 3. FREE CONVECTION FACTORS FOR VARIOUS SHAPES

SHAPES	FACTOR
Horizontal cylinders 24 in. in diam. or over	0.73
Long vertical cylinders 24 in. in diam. or over	0.88
Vertical plates 24 in. in height or over	1.00
Horizontal plates warmer than air facing upward	1.28
Horizontal plates warmer than air facing downward	0.64
Horizontal plates cooler than air facing upward	0.64
Horizontal plates cooler than air facing downward	1.28

TABLE 4. FREE CONVECTION FACTORS FOR VARIOUS DIAMETER PIPES OR VARIOUS HEIGHT PLATES

Actual O.D., or height, in.	1	2	3	4	5	6	7	8
Factor	1.88	1.64	1.52	1.43	1.37	1.32	1.28	1.25
Actual O.D., or height, in.	9	10	12	14	16	18	20	22
Factor	1.22	1.19	1.15	1.11	1.09	1.06	1.04	1.02

NOMENCLATURE AND DIMENSIONS FOR TABLE 5

c_p = fluid unit heat capacity at constant pressure, Btu per pound per degree Fahrenheit.
D = cylinder diameter, feet.
G = 3600 $V_s\rho$ = fluid mass velocity, pounds per hour per square foot of flow cross-section.
ρ = density, pounds per cubic foot.
h_c = unit conductance for thermal convection, Btu per hour per square foot per degree Fahrenheit.
k = unit thermal conductivity of the fluid, Btu per hour per square foot per degree Fahrenheit for one foot thickness.
R_H = hydraulic radius of the flow cross-section = flow cross-section area per wetted perimeter, feet.
s = fin spacing, feet
t = average fluid film temperature, degree Fahrenheit.
$t_1 - t_2$ = temperature difference surface to main fluid, degree Fahrenheit.
V_s = fluid velocity, feet per second.
μ = fluid viscosity, pounds per hour per foot = viscosity in centipoises × 2.42.

TABLE 5. APPROXIMATE UNIT CONDUCTANCES FOR THERMAL CONVECTION FOR SEVERAL FLOW SYSTEMS[a]

Expressed in Convenient Empirical Form

CASE	SYSTEM	UNIT CONDUCTANCE EQUATION[b]
	FORCED CONVECTION	
1	Longitudinal flow in cylinders, turbulent region Fluid being heated[c]	$\frac{h_c D}{k} = 0.0225 \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{c_p \mu}{k}\right)^{0.4}$ For $\left(\frac{DG}{\mu}\right) > 3000$
2	For longitudinal air flow in cylinders case 1 reduces to[c]	$h_c = 0.0036\ G^{0.8}/D^{0.2}$ For $\left(\frac{DG}{\mu}\right) > 3000$
3	For longitudinal water flow in cylinders case 1 reduces to[c].	$h_c = 0.00486\ (1 + 0.01t) \frac{G^{0.8}}{D^{0.2}}$ For $\left(\frac{DG}{\mu}\right) > 3000$
4	Air flow normal to a single right circular cylinder.	$h_c = 0.45 \left(\frac{k}{D}\right) + 0.178\ G^{0.56} \left(\frac{k}{D}\right)^{0.44}$
5	Air flow over staggered pipe banks	$h_c = 0.061 \left(\frac{k}{D}\right)^{0.31} G^{0.69}$
6	Air flow over single spheres	$h_c = 0.010 \frac{G^{0.52}}{D^{0.48}}$ $0 < t < 250$ F
7	Air flow over plane surfaces	$h_c = 1 + 0.22\ V_s$ For $V_s < 16$ fps or $h_c = 0.53\ V_s^{0.8}$ 16 fps $< V_s < 100$ fps
8	Air flow normal to finned cylinders	$h_c = 6.2 \left(\frac{G}{3600}\right)^{0.8} \frac{s^{0.32}}{D^{0.52}}$ $0 < t < 250$ F
	FREE CONVECTION[d]	
9	Single horizontal right circular cylinder in air	$h_c = 0.23 \left(\frac{t_1 - t_2}{D}\right)^{0.25}$
10	Vertical surfaces in air	$h_c = 0.3\ (t_1 - t_2)^{0.25}$
11	Top surface of horizontal plates to air	$h_c = 0.4\ (t_1 - t_2)^{0.25}$
12	Bottom surface of horizontal plates to air	$h_c = 0.2\ (t_1 - t_2)^{0.25}$

[a]Heat Transmission, by W. H. McAdams

[b]Fluid properties should be evaluated at the arithmetic mean fluid temperature, $t_f = (t_{surface} + t_{fluid})$ divided by 2.

[c]These expressions are applicable to longitudinal flow in other than right circular cylinders provided the hydraulic radius is employed as the conduit dimension parameter. For non-circular cross-sections $D = 4\ R_H$.

[d]For low rates of heat transfer by free convection the exponent decreases towards zero, and for higher rates increases towards 0 33 The following equations employing an exponent equal to 0 25 are applicable in the intermediate range.

Thermal Radiation Equation

The relation shown in Equation 3 is usually applicable to systems in which radiant exchange takes place between the surfaces of solids, as sche-

$$q_r = \sigma A_1 F_A F_E (T_1^4 - T_2^4) \tag{3}$$

matically shown in Fig. 3. Gaseous and luminous radiation are not considered in this discussion. Equation 3 states that the net radiation current per unit transfer area of surface 1, q_r/A Btu per hour per square foot, which *sees* surface 2 through a non-absorbing medium, is proportional to the

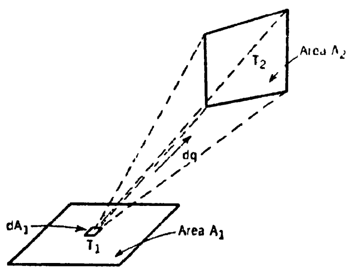

FIG. 3 RADIATION BETWEEN SURFACES

difference of the fourth powers of the absolute surface temperatures $(T_1^4 - T_2^4)$. The proportionality factor $(\sigma\ F_A F_E)$ may be conveniently separated into three parts:

σ = the Stefan-Boltzmann radiation constant
= 1730×10^{-12} Btu per hour per square foot per degree Fahrenheit absolute temperature to the fourth power

F_A = the configuration factor is dimensionless and ≤ 1. This factor accounts for the shape and relative position of the two surfaces. The value of $F_A = 1$ may be used in the cases of large parallel planes, long concentric cylinders or smaller bodies in large enclosures. (For other values see References.)

F_E = the emissivity factor is also dimensionless and ≤ 1. This factor accounts for the absorption and emission characteristics of the surfaces for the radiation which exists. Individual emissivities (e) should be taken from Table 6 and applied, for either radiation or absorption, as follows:

a. For a small body in a large enclosure, use the emissivity of the small body only: $F_E = \varepsilon_1$.

b. For rectangles or disks, either parallel or perpendicular and with a common side, use the product of the emissivities: $F_E = \varepsilon_1 \times \varepsilon_2$.

c. For large parallel planes, long concentric cylinders or large enclosed bodies, use both emissivities in the equation:

$$F_E = \frac{1}{\dfrac{1}{\varepsilon_1} + \dfrac{1}{\varepsilon_2} - 1}$$

The radiation under black-body conditions, or for an emissivity of 1.0, is given in Table 7[5] for cold surfaces as low as −39 F to warmer surfaces as high as 139 F. The emissivities of a number of surfaces ordinarily encountered in engineering practice are shown in Table 6. For radiation table at higher temperatures, and further discussion of radiation calculations, see Chapter 45.

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[5]Heat Insulation in Air Conditioning, by R. H. Heilman (*Industrial and Engineering Chemistry*, Vol. 28, July 1936, p. 782).

Combined Convection and Radiation

It should be noted that the previous equations and tables give the heat transfer by convection and by radiation computed separately. In many practical cases it is desirable to treat convection and radiation as a single combined process, using a first-power equation:

$$q_{rc} = h_{rc} A (t_1 - t_2) \tag{4}$$

where q_{rc} is the total heat flow due to radiation and convection, in Btu per hour. Values of h_{rc}, the surface or film conductance for combined

TABLE 6 RADIATION FACTORS OR EMISSIVITIES, ε

For the determination of factor F_E in Equation 3

CLASS	SURFACES	FRACTION OF BLACK-BODY RADIATION		
		At 50–100 F	At 1000 F	Solar Radiation
1	A small hole in a large box, sphere, furnace, or enclosure	0 97 to 0 99	0 97 to 0 99	0 97 to 0 99
2	Black non-metallic surfaces such as asphalt, carbon, slate, paint, paper	0 90 to 0 98	0 90 to 0 98	0.85 to 0 98
3	Red brick and tile, concrete and stone, rusty steel and iron, dark paints (red, brown, green, etc.)	0.85 to 0.95	0 75 to 0 90	0 65 to 0 80
4	Yellow and buff brick and stone, firebrick, fire clay	0 85 to 0.95	0 70 to 0 85	0 50 to 0 70
5	White or light-cream brick, tile, paint or paper, plaster, whitewash	0.85 to 0 95	0.60 to 0.75	0 3 to 0.5
6	Window glass	0 90 to 0 95	.. ..	Transparent
7	Bright aluminum paint; gilt or bronze paint	0 4 to 0 6	.. ...	0 3 to 0 5
8	Dull brass, copper, or aluminum, galvanized steel, polished iron	0.2 to 0.3	0 3 to 0.5	0 4 to 0 65
9	Polished brass, copper, monel metal	0.02 to 0 05	0 05 to 0.15	0 3 to 0 5
10	Highly polished aluminum, tin plate, nickel, chromium	0.02 to 0 04	0 05 to 0 10	0 10 to 0 40

radiation and convection, are given in Chapter 4, Table 1 and Fig. 1. Complete tables for the combined heat transfer of steam and hot water radiators, pipes, coverings, etc., will be found in the appropriate chapters.

When dealing with the effect of operating temperatures upon the combined heat transfer of a given piece of equipment (as for instance a steam radiator), another form of equation is frequently used:

$$q_{rc} = B A (t_1 - t_2)^n \tag{5}$$

Values of n in this equation usually range from 1.3 to 1.5 (see Chapter 13). The chief advantage of this equation is the convenience of representing

heat transfer performance on logarithmic coordinates, and the factor B should be regarded as a simple constant of proportionality.

HEAT-FLOW RESISTANCE

In most of the steady-state heat transfer problems encountered in air conditioning applications, more than one of the heat transfer mechanisms are effective, and the thermal current flows through several resistances in series or in parallel. In using the resistance concept the calculations involved are analogous to the application of Ohm's Law in electricity, *viz.*, the heat flow or thermal current is directly proportional to the thermal

TABLE 7. HEAT TRANSMISSION BY RADIATION FOR BLACK-BODY CONDITIONS[a]

Expressed in Btu per square foot per hour

TEMP DEG F	0	−1	−2	−3	−4	−5	−6	−7	−8	−9
−30	59.3	58.7	58.2	57.7	57.2	56.7	56.2	55.7	55.2	54.7
−20	65.2	64.7	64.1	63.5	62.9	62.3	61.7	61.1	60 5	59.9
−10	71.4	70.8	70.1	69.5	68.9	68.3	67 7	67.1	66.4	65.8
0	78.0	77.4	76.7	76.0	75.4	74.7	74.0	73.4	72.7	72.1
	0	+1	+2	+3	+4	+5	+6	+7	+8	+9
0	78.0	78.7	79.4	80.1	80.8	81.5	82.2	82.9	83.6	84.3
10	85.0	85.7	86.5	97.2	88.0	88.7	89.4	90.2	90.9	91.7
20	92.4	93.3	94.0	94.8	95.6	96.4	97.2	98.0	98.8	99.6
30	100	101	102	103	104	105	105	106	107	108
40	109	110	111	112	112	113	114	115	116	117
50	118	119	120	121	122	123	123	124	125	126
60	127	128	129	130	131	132	133	134	135	136
70	137	138	139	140	142	143	144	145	146	147
80	148	149	150	151	152	153	154	155	156	157
90	159	160	161	162	163	164	166	167	168	169
100	170	171	173	174	175	176	178	179	180	182
110	183	184	185	187	188	189	191	192	193	195
120	196	197	199	200	201	203	204	206	207	209
130	211	212	214	215	217	218	220	221	222	224

[a]*Example:* Radiation from walls of room at 32 F to surface at − 25 F for effective emissivity of 0 95 = (102 − 62 3) 0 95 = 37 7 Btu per square foot per hour

potential or temperature difference, and inversely proportional to the thermal resistance:

$$q_{rc} = \frac{t_1 - t_2}{R} \tag{6}$$

Following the electrical analogy, when there is a thermal current flowing through several *resistances in series*, the resistances are additive:

$$R_T = R_1 + R_2 + R_3 + \ldots . \; + R_n \tag{7}$$

Similarly, conductance is the reciprocal of resistance, and for heat flow through several *resistances in parallel*, the conductances are additive:

$$C_T = \frac{1}{R_T} = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} + \ldots\ldots + \frac{1}{R_n} \tag{8}$$

Practical Heat Transfer Problems

The use of these simple relations for resistance and conductance simplifies many practical heat transfer problems. As discussed in Chapters 4,

17 and 25, the practical analyses of heat transfer in building walls, in fin-tube coils and in pipe coverings, are usually computed by this method.

The same resistance analysis may be applied to complicated steady-state *conduction* problems. Table 8 indicates the solutions in six common cases of steady-state conduction.

A complete analysis by the resistance method is best illustrated by considering the heat transfer from the air outside to the cold water inside of an insulated pipe. The temperature gradients and the nature of the resistance analysis are indicated by the two sketches of Fig. 4.

Since air is sensibly transparent to radiation, there will be some heat transfer by both radiation and convection to the outer insulation surface. The mechanisms act in parallel on the air side. The total current by radiation and convection then passes through the insulating layer and the pipe wall by thermal conduction, and thence by convection into main cold water streams. Radiation is not significant on the water side as

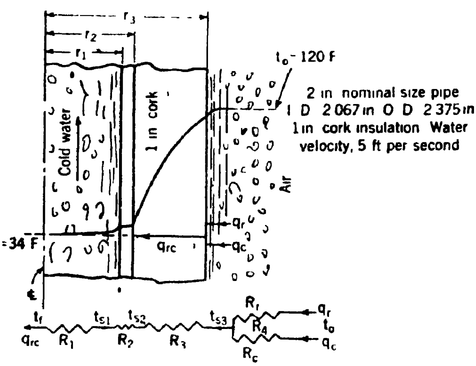


FIG. 4. HEAT TRANSFER CONDITIONS IN THE INSULATED COLD WATER LINE

liquids are sensibly opaque to radiation, although water transmits energy in the visible region. The contact resistance between the insulation and the pipe wall is presumed to be equal to zero.

Referring to Fig. 4, the thermal current for a given length N of pipe, q_{rc} Btu per hour, may be thought of as flowing through the parallel resistances R_r and R_c, associated with the insulation surface radiation and convection transfer. Then the flow is through the resistance offered to thermal conduction by the insulation, R_3, through the pipe wall resistance, R_2, and into the water stream through the convection resistance, R_1. Note the analogy to the direct current electrical circuit problem. A temperature (potential) drop is required to overcome these resistances to the flow of thermal current. The total resistance to heat transfer, R_T, hour degrees Fahrenheit per Btu, is the summation of the individual resistances:

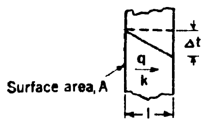
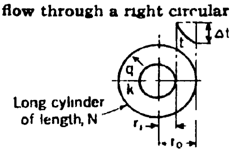
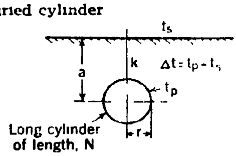
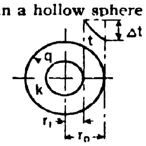
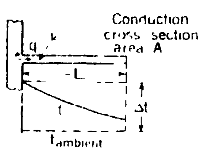
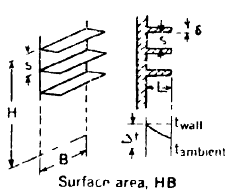
$$R_T = R_1 + R_2 + R_3 + R_4 \tag{9}$$

where the resultant parallel resistance R_4 is obtained from:

$$\frac{1}{R_4} = \frac{1}{R_r} + \frac{1}{R_c}$$

Provided the individual resistances may be evaluated, the total resistance

Table 8. Solutions for Some Steady-State Thermal Conduction Problems[a,b]

No.	System	Expressions for the resistance R entering into the equation: $q = \Delta t/R$ (Btu per hour)
1.	Flat wall or curved wall if curvature is small (wall thickness less than 0.1 of inside diameter) 	$R = \dfrac{L}{kA}$
2.	Radial flow through a right circular cylinder 	$R = \dfrac{\log_e \dfrac{r_o}{r_i}}{2\pi kN}$ (See footnote c)
3	The buried cylinder 	$R = \dfrac{\log_e \dfrac{2a}{r}}{2\pi kN}$, $R = \dfrac{\cosh^{-1} \dfrac{a}{r}}{2\pi kN}$ for $\dfrac{a}{r} \geqq 3$ (See footnote c)
4	Radial flow in a hollow sphere 	$R = \dfrac{\dfrac{1}{r_i} - \dfrac{1}{r_o}}{4\pi k}$
5	The straight fin or rod heated at one end 	$R = \dfrac{m}{h_s p \tanh mL}$ (see footnotes d and e) For $ml > 2.3$, $\tanh mL \approx 1$ $m = \sqrt{h_s p/kA}$ A = conduction cross-section area p = perimeter of cross-section h_s = unit conductance to the surroundings from the fin surface k = thermal conductivity fin material Δt = wall temperature—ambient temperature
6.	Finned surface of area HB 	$R = \dfrac{(s + \delta)}{h_s \left(\dfrac{2}{m} \tanh m\,l + s \right) HB}$ $m = \sqrt{\dfrac{h_s p}{kA}} = \sqrt{\dfrac{2\,h_s}{k\delta}}$ Δt defined as in Case 5 above

[a] The dimensions to be employed in these solutions are: length of dimension p, L, r = feet; units of k = Btu per hour per square foot per degree Fahrenheit for one foot thickness; units of h, Btu per hour per square foot per degree Fahrenheit, units of area, A = square feet.

[b] The thermal conductivity, k, in these solutions should be taken at the average material temperature (see Table 5).

[c] $\log_e x = 2.303 \log_{10} x$

[d] This expression can also be employed as an approximation for tapered fins or of annular fins by employing average magnitudes of A and p.

[e] Tanh is the hyperbolic tangent

can be obtained from this relation. Then the heat transfer current for the length of pipe (N, ft) can be established by the relation:

$$q_{rc} \text{ (Btu per hour)} = \frac{t_o - t_f}{R_T} \tag{10}$$

For a unit length of the pipe the heat transfer rate is:

$$\frac{q_{rc}}{N} \text{ (Btu per hour foot)} = \frac{(t_o - t_f)}{R_T N} \tag{11}$$

The temperature drop, Δt, through an individual resistance may then be calculated from the relation:

$$\Delta t = R\, q_{rc}$$

where R is the resistance in question.

The problem is now reduced to one of evaluating the individual resistances of the system. This entails suitable integration of the rate Equations 1, 2 and 3 to produce expressions of the form:

$$q = \frac{\Delta t}{R} \tag{12}$$

where q is the heat transfer rate, and Δt is the potential drop or temperature difference through the resistance R. Table 8 lists such solutions for six different conduction systems. Table 2 in Chapter 4 and Table 1 of this chapter indicate the magnitudes of the thermal conductivities, k, to be employed in the expressions of Table 8.

The solution applicable to the problem depicted in Fig. 4, for the calculation of R_2 and R_3, is case 2 in Table 8. Thus for a *1 ft length* of 2 in. nominal size pipe (I. D. = 2 067 in., O. D. = 2.375 in.) insulated with 1 in. of cork:

$$R_2 = \frac{\log_e \dfrac{1.188}{1.033}}{2\pi \times 26 \times 1} = 8.5 \times 10^{-4} \text{ hr degree Fahrenheit per Btu.}$$

$$R_3 = \frac{\log_e \dfrac{2.188}{1\ 188}}{2\pi \times 0.025 \times 1} = 3\ 9 \text{ hr degree Fahrenheit per Btu.}$$

The convection resistances to heat transfer from the pipe wall to the cold water, R_1, and from the air to the surface of the insulating material, R_c, are dependent on the flow conditions prevailing at these surfaces, and on the thermal properties of the fluids. The unit conductances for thermal convection, h, Btu per hour per square foot per degree Fahrenheit, have been determined by test for many flow systems. These data may be employed to predict the conductances for *similar flow systems*. Table 5 summarizes some empirical equations expressing such test results.

For the problem under consideration (Fig. 4) case 3 of Table 5 is applicable for the calculation of the cold water side convection resistance R_1. Corresponding to the water velocity of 5 fps, the mass velocity is:

G = 5 (ft per sec) $\times$ 62.4 (lb per cu ft) $\times$ 3600 (sec per hr) = $11\ 2 \times 10^5$ lb per hour per square foot.

The inside diameter of the pipe D is $\dfrac{2.067}{12} = 0.1725$ ft.

The average water film temperature will be estimated as 36 F (mixed mean fluid temperature of 34 F). Then case 3, Table 5 yields:

$$h = 0.00486\ (1 + 0.36)\ \frac{(11.2 \times 10^5)^{0.8}}{(0.1725)^{0.2}} = 650 \text{ Btu per hour per square foot per degree Fahrenheit.}$$

The transfer area on which this conductance is based is the inside tube area. Associated with 1 ft length of pipe there are:

$$\pi \times \frac{2.067}{12} \times 1 = 0.542 \text{ sq ft.}$$

Thus the resistance for 1 ft of tube length is:

$$R_1 = \frac{1}{h\pi D \times 1} = \frac{1}{650 \times 0.542} = 2.8 \times 10^{-3} \text{ hr degree Fahrenheit per Btu.}$$

Case 9, Table 5 is applicable for calculating the free thermal convection resistance, R_c, existing between the surrounding air and the insulation. The air temperature is given as 120 F. As an approximation a 20 F temperature difference between the air and the pipe surface will be assumed. Then case 9 yields:

$$D = \frac{4.375}{12} = 0.364 \text{ ft.}$$

$$h = 0.23 \left(\frac{20}{0.364}\right)^{0.25} = 0.63 \text{ Btu per hour per square foot per degree Fahrenheit.} \quad (13)$$

This result may not be deemed conservative inasmuch as the expression is for *still* air. If, however, the air is not still, but flows at approximately 5 mph or 7 fps the mass velocity corresponds to:

$$G = 7 \times 0.07 \times 3600 = 1770 \text{ lb air per hour per square foot.}$$

A magnitude of $k = 0.014$ Btu per hour per square foot per degree Fahrenheit for one foot thickness applied to case 4 yields:

$$h = 0.45 \left(\frac{0.014}{0.364}\right) + 0.178\ (1770)^{0.56} \left(\frac{0.014}{0.364}\right)^{0.44}$$

$$= 0.017 + 2.8 = 2.8 \text{ Btu per hour per square foot per degree Fahrenheit.}$$

This conductance is based on 1 sq ft of outside lagging area. Thus, since there are $\pi \times \frac{4.375}{12} = 1.14$ sq ft of outside lagging area associated with 1 ft length of pipe:

$$R_c = \frac{1}{2.8 \times 1.14} = 0.312 \text{ hr degree Fahrenheit per Btu.}$$

The radiation resistance, R_r, which acts in parallel with the convection resistance, R_c, for the transfer of heat to the surface of the insulation, may be calculated. For the purposes of this illustrative problem it will be assumed that the insulated pipe is exposed to (*sees*) surroundings, which exist at 120 F. Then the angle factor, F_A, is unity and for an estimated surface emissivity of 0.9 (see Table 6), $F_e = 0.9$. As a first approximation the insulation surface temperature will be estimated as 20 F lower than the surroundings at 120 F. Then the radiation per degree of temperature

difference, by Equation 3 (or more conveniently by Table 8) divided by the temperature difference will be:

$$h_r = \frac{(196 - 170)\,0.95}{20} = 1.17 \text{ Btu per hour per square foot per degree Fahrenheit.}$$

The outside surface area of the insulation associated with 1 ft of pipe length was previously calculated as 1.14 sq ft. Thus:

$$R_r = \frac{1}{1.17 \times 1.14} = 0.75 \text{ hr degree Fahrenheit per Btu.}$$

The resultant resistance of R_c and R_r acting in parallel (see Fig. 4) can now be evaluated as:

$$\frac{1}{R_4} = \frac{1}{R_c} + \frac{1}{R_r} = \frac{1}{0.312} + \frac{1}{0.75} = 4.54 \text{ Btu per hour per degree Fahrenheit.}$$

$$R_4 = 0.22 \text{ hr degree Fahrenheit per Btu.}$$

The individual resistances for a 1 ft length of pipe applying to the illustrative problem depicted in Fig. 4 have now been calculated and are summarized as follows:

R_1 convection from the pipe wall to the cold water $= 2.8 \times 10^{-3}$ hr degree Fahrenheit per Btu.
R_2 conduction through the pipe wall $= 8.5 \times 10^{-4}$ hr degree Fahrenheit per Btu.
R_3 conduction through the cork insulation $= 3.9$ hr degree Fahrenheit per Btu.
R_4 parallel convection and radiation from the surroundings $= 0.22$ hr per degree Fahrenheit per Btu.

Then:

R_T = the over-all resistance surroundings to cold water $= R_1 + R_2 + R_3 + R_4 = 4.1$ hr degree Fahrenheit per Btu.

Note that the controlling resistances are R_3 and R_4. That is, the neglect of R_1 and R_2 would not significantly influence the total resistance, R_T.

On the basis of this resistance calculation the heat transfer from the surroundings to the cold water may be evaluated as:

$$\frac{q_{rc}}{N} = \frac{t_o - t_f}{R_T} = \frac{120 - 34}{4.1} = 21 \text{ Btu per hour per foot}$$

or about 0.175 tons of refrigeration per 100 ft of pipe.

Since the calculation is based on a 1 ft pipe length:

$$q_{rc} = 21 \text{ Btu per hour}$$

The temperature drops through the various resistances are now readily evaluated by Equation 12 as:

$t_o - t_{s3}$ air to insulation surface $= R_4\, q_{rc} = 0.22 \times 21 = 4.6$ F.
$t_{s3} - t_{s2}$ through the insulation $= R_3\, q_{rc} = 3.9 \times 21 = 82$ F.
$t_{s2} - t_{s1}$ through the pipe wall $= R_2\, q_{rc} = 8.5 \times 10^{-4} \times 21 = 0.02$ F.
$t_{s1} - t_f$ pipe wall to cold water $= R_1\, q_{rc} = 2.8 \times 10^{-3} \times 21 = 0.06$ F.

The solution was obtained on the assumption that the air temperature and the outside temperature differed by 20 F. In order to obtain a slightly better estimate of the rate of heat transfer the numerical solution should be repeated using the temperatures calculated from the previous listed temperature differences.

The foregoing problem serves to illustrate a general method of solving

steady-state heat transfer problems. There are many problems which cannot be approximated by steady-state solutions. For instance, the problem of pipe line insulation in transient service; the behavior of automatically controlled thermoflow circuits; or the periodic absorption of solar energy by roof and wall structures during the day and nocturnal radiation to the *cold* sky at night. The transient heat transfer problem differs from the steady-state in that energy storage rates need to be considered. Thus thermal capacity in addition to resistance effects are significant. The vector sum of the thermal capacitance and resistance is the thermal impedance. It is not within the scope of this chapter to deal with many of these problems. There are, however, solutions available in graphical form for certain special cases. Also a general approximate method may be employed which is analogous to the treatment of capacity-resistance lumped parameter electrical circuits.

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CHAPTER 4

Heat Transmission Coefficients

Transfer Through Building Surfaces, Heat Transfer Symbols, Formulas for Calculating Over-all Coefficients, Conductivity of Homogeneous Materials, Surface Conductance Coefficients, Air Space Conductance, Practical Coefficients, Computed Transmission Coefficients, Combined Coefficients of Transmission, Basement Floor and Wall Coefficients, Condensation in Buildings

IN order to calculate the heat transfer through walls, ceilings, floors and other parts of a structure it is necessary to know the *rate* of heat transfer through these surfaces. This rate of heat transfer is designated as the coefficient of transmission and can be determined by test in the guarded hot box apparatus, or calculated if certain constants are known. Because of the many possible combinations of materials in building construction, it is impractical to test each individual construction. Instead the over-all coefficients of transmission are calculated from the individual or component conductivities and conductances according to the procedure described in this chapter.

TRANSFER THROUGH BUILDING SURFACES

A general discussion of the three methods of heat transfer—conduction, convection and radiation—will be found in Chapter 3. The heat transmission between the air on the two sides of a structure takes place by a combination of the three methods. In a simple wall built up of two layers of homogeneous materials separated to give an air space between them, heat will be received from the high temperature surface by radiation, convection and conduction. It will then be conducted through the homogeneous interior section by conduction and carried across to the opposite surface of the air space by radiation, conduction and convection. From here it will be carried by conduction through to the outer surface and leave the outer surface by radiation, convection and conduction.

HEAT TRANSFER SYMBOLS

The symbols representing the various coefficients of heat transmission and their definitions are:

U = over-all coefficient of heat transmission; the amount of heat expressed in Btu transmitted in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 F between the air on the inside and that on the outside of the wall, floor, roof or ceiling.

k = thermal conductivity; the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a homogeneous material 1 in. thick for a difference in temperature of 1 F between the two surfaces of the material. The conductivity of any material depends on the structure of the material and its density. Heavy or dense materials, the weight of which per cubic foot is high, usually transmit more heat than light or less dense materials, the weight of which per cubic foot is low.

C = thermal conductance; the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a material for the thickness or type under consideration for a difference in temperature of 1 F between the two surfaces of the material. Conductance is usually used to designate the heat transmitted through such heterogeneous materials as plasterboard and hollow clay tile.

f = film or surface conductance; the amount of heat expressed in Btu transmitted by radiation, conduction and convection from a surface to the air surrounding it, or vice versa, in one hour per square foot of the surface for a difference in temperature of 1 F between the surface and the surrounding air. To differentiate between inside and outside

wall (or floor, roof or ceiling) surfaces, f_i is used to designate the inside film or surface conductance and f_o the outside film or surface conductance.

a = thermal conductance of an air space; the amount of heat expressed in Btu transmitted by radiation, conduction and convection in one hour through an area of 1 sq ft of an air space for a temperature difference of 1 F. The conductance of an air space depends on the mean absolute temperature, the width, the position and the character of the materials enclosing it.

R = resistance or resistivity which is the reciprocal of transmission, conductance, or conductivity, *i.e.*:

$\frac{1}{U}$ = over-all or air-to-air resistance.

$\frac{1}{k}$ = internal resistivity.

$\frac{1}{C}$ = internal resistance.

$\frac{1}{f}$ = film or surface resistance.

$\frac{1}{a}$ = air space resistance.

Examples of the application of the over-all coefficient U for determining the heat transfer by transmission, are given in Chapter 6.

FORMULAS FOR CALCULATING OVER-ALL COEFFICIENTS

The simplest method of combining the coefficients for the individual parts of the wall is to use the reciprocals of the coefficients and treat them as resistance units. The total over-all resistance of a wall is equal numerically to the sum of the resistances of the various parts, and the reciprocal of the over-all resistance is likewise the over-all heat transmission coefficient of the wall. For a wall built up of a single homogeneous material of conductivity k and x inches thick the over-all resistance,

$$R = \frac{1}{U} = \frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_o} \tag{1}$$

If the coefficients f_i, f_o and k, together with the thickness of the material x are known, the over-all coefficient U may be readily calculated as the reciprocal of the total heat resistance.

For a compound wall built up of three homogeneous materials having conductivities k_1, k_2 and k_3 and thicknesses x_1, x_2 and x_3 respectively, and laid together without air spaces, the total resistance,

$$R = \frac{1}{U} = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{f_o} \tag{2}$$

For a wall with air space construction consisting of two homogeneous materials of thicknesses x_1 and x_2, and conductivities k_1 and k_2, respectively, separated to form an air space of conductance a, the over-all resistance,

$$R = \frac{1}{U} = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{1}{a} + \frac{x_2}{k_2} + \frac{1}{f_o} \tag{3}$$

Likewise any combination of homogeneous materials and air spaces can be put into the wall and the over-all resistance of the combination may be calculated by adding the resistances of the individual sections of the wall. In certain special forms of construction such as tile with irregular air spaces it is necessary to consider the conductance C of the unit as built instead of the unit conductivity k, and the resistance of the section is equal to $\frac{1}{C}$. The method of calculating the over-all heat transmission

coefficient for a given wall is comparatively simple, but the selection of the proper coefficients is often complicated. In some cases the construction of the wall is such that the substituting of coefficients in the accepted formula will give erroneous results. This is the case with irregular cored out air spaces in concrete and tile blocks, and walls in which there are parallel paths for heat flow through materials having different heat resistances. In such cases it is necessary to resort to test methods to check the calculations, and in practically all cases it has been necessary to determine fundamental coefficients by test methods.

Conductivity of Homogeneous Materials

The thermal conductivity of homogeneous materials is affected by several factors. Among these are the density of the material, the amount of moisture present, the mean temperature at which the coefficient is determined, the size of fibers or particles, and their arrangements in the material, and possibly position. There are many materials used in building construction and considered as homogeneous for the purpose of calculation, whereas they are not really homogeneous but are merely considered so as a matter of convenience. In general, the thermal conductivity varies with the density of the material, increases with the amount of moisture present, and increases with the mean temperature at which the coefficient is determined. The rate of change for these various factors is not the same for all materials, and in assigning proper coefficients one should make certain that they apply for the conditions under which the material is to be used in a wall. Failure to do this may result in serious errors in the final coefficient.

With respect to position, convection within the material may have an effect on the over-all heat transmission. According to one investigator[1] the actual rate of heat flow through loose insulating material may be somewhat greater than that indicated by the hot plate test, but in another investigation[2], there was found to be a negligible difference between horizontal and vertical upward heat flow through loose materials.

Surface Conductance Coefficients

Heat is transmitted to or from the surface of a wall by a combination of radiation, convection and conduction. The coefficient will be affected by any factor which has an influence on any one of these three methods of transfer. The amount of heat by radiation is controlled by the character of the surface and the temperature difference between it and the surrounding objects. The amount of heat by conduction and convection is controlled largely by the roughness of the surface, by the air movement over the surface and by the temperature difference between the air and the surface. Because of these variables the surface coefficients may be subject to wide fluctuations for different materials and different conditions. The inside and outside coefficients f_i and f_o are in general affected to the same extent by these various factors and test coefficients determined for inside surfaces will apply equally well to outside surfaces under like conditions. Values for f in still and moving air at different mean temperatures have been determined for various building materials[3].

[1]The Effect of Convection in Ceiling Insulation, by G B Wilkes and L R Vianey (A S H.V E TRANSACTIONS, Vol 49, 1943, p 196)

[2]Heat Transmission Through Insulation as Affected by Orientation of Wall, by F. B Rowley and C E Lund (A S H V E TRANSACTIONS, Vol. 49, 1943, p 331).

[3]A.S H.V E RESEARCH REPORT No. 869—Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B Algren and J L. Blackshaw (A S H V E. TRANSACTIONS, Vol 36, 1930, p. 429).

The surface conductances for different materials at mean temperatures of 20 F are shown in Fig. 1. These values were obtained with air flow parallel to the surface and from other tests in which the angle of incidence between the direction of air flow and the surface was varied from zero to 90 degrees it would appear that values might be lowered approximately 15 per cent for average conditions. While for average building materials there is a difference due to mean temperature, the greatest variation in these coefficients is caused by the character of the surface and the wind velocity. If other surfaces, such as aluminum foil with low emissivity coefficients were substituted, a large part of the radiant heat would be eliminated. This would reduce the total coefficient for all wind veloci-

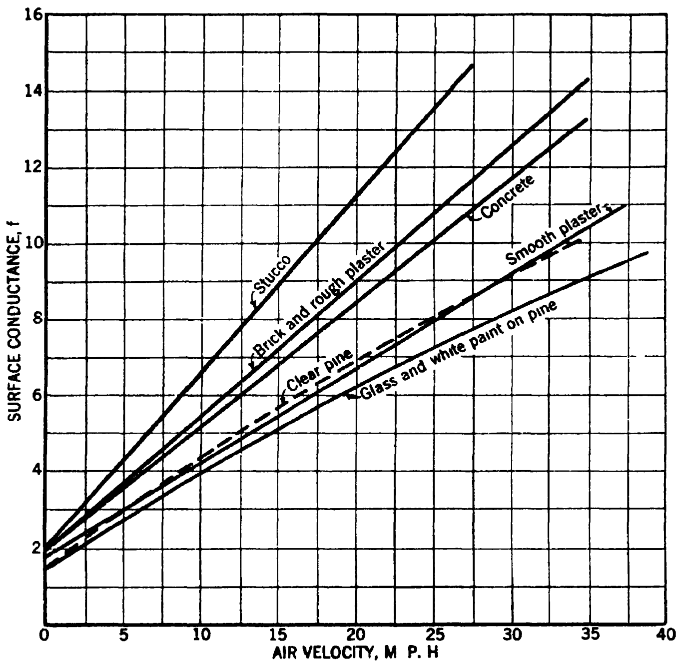


FIG. 1. CURVES SHOWING RELATION BETWEEN SURFACE CONDUCTANCES FOR DIFFERENT SURFACES AT 20 F MEAN TEMPERATURE

ties by about 0.7 Btu and would make but very little difference for the higher wind velocities. In many cases in building construction the heat resistance of the internal parts of the wall is high as compared with the surface resistance and the surface factors become of small importance. In other cases such as single glass windows the surface resistances constitute practically the entire resistance of the structure, and therefore become important factors. Due to the wide variation in surface coefficients for different conditions their selection for a practical building becomes a matter of judgment. In calculating the over-all coefficients for the walls of Tables 4 to 16, 1.65 *has been selected as an average inside coefficient* and 6.0 *as an average outside coefficient* for a 15-mile wind velocity. In special cases where surface coefficients become important factors in the over-all rate of heat transfer more selective coefficients may be required.

The surface conductance values given in Table 1, **Section A** are based on recent tests and are for still air conditions and emissivities of 0.83 and 0.05 respectively, and may be used where it is desirable to differentiate between vertical and horizontal surfaces or where coefficients applicable to low-emissivity surfaces are required.

Air Space Conductance

Heat is conducted across an air space by a combination of radiation, conduction and convection. The amount of heat by radiation is governed largely by the nature of the surface and the temperature difference between the boundary surfaces of the air space. Conduction and convection are controlled largely by the width and shape of the air space and the roughness of the boundary surfaces.

The conductances of vertical air spaces bounded by such materials as paper, wood, plaster, etc., are given in Table 1, **Section B,** having emissivity coefficients of 0.8 or higher, and with extended parallel surfaces perpendicular to the direction of heat flow. A conductance of 1.10 Btu per hour per square foot per degree Fahrenheit temperature difference (resistance = 0 91) based on this table was used for calculating the over-all coefficients given in Tables 4 to 16 inclusive for air spaces ¾ in. or more in width. Air space tests[4] reported by Wilkes and Peterson resulted in comparable values For 3⅝ in. horizontal air spaces having an effective emissivity of 0.83, the conductance for heat flow upward was 1.32 and for heat flow downward, 0.94. The conductance for a similar vertical air space was 1 17, the resistances of course being the reciprocals of these values in each case.

A large part of the heat transferred across air spaces bounded by ordinary materials is by radiation. Therefore, if such air spaces are faced with metallic surfaces such as aluminum foil, coated sheet steel or other low-emissivity, infra-red reflective metal surfaces, the radiant heat transfer will be substantially reduced, thus causing the major portion of the remaining transmitted heat to be by convection. Table 1, **Section C,** gives conductances and resistances for air spaces *bounded by one reflective surface having an emissivity of 0 05.* It will be noted that the conductance values given in this table are a function of the temperature differences across the space rather than mean temperature, the larger the temperature difference, the larger the conductance. The radiant heat transfer is the same regardless of whether the low emissivity surface is on the high or low temperature surface of the space, and is independent of the width of the space. To minimize the convection transfer the vertical air space should be at least ¾ in. in width.

When referring to reflective heat-insulating surfaces, the term *brightness* which deals with visible light has no specific meaning and should be avoided[5]. *Emissivity* and *reflectivity* definitely define the radiating and reflecting properties and values may be determined directly for long wavelength radiation corresponding to room temperature. As previously stated, the values in Table 1, **Section C,** are based on an emissivity of the reflective surface of 0 05. Obviously for reflective materials having higher emissivity, the conductance will increase accordingly.

Where reflective insulating materials are involved the possible increase

[4]Radiation and Convection Across Air Spaces in Frame Construction, by G B Wilkes and C M F Peterson (A.S H V E TRANSACTIONS, Vol 43, 1937, p 351)

[5]Some Reflection and Radiation Characteristics of Aluminum, by C S Taylor and J D Edwards (A S H V E TRANSACTIONS, Vol 45, 1939, p 179)

Table 1. Conductances (*C*) for Surfaces and Air Spaces

All conductance values expressed in Btu per hour per square foot per degree Fahrenheit temperature difference

Section A. Surface Conductances for Still Air[a]

Position of Surface	Direction of Heat Flow	Surface Emissivity	
		e = 0 83	e = 0 05
Horizontal	Upward	1.95	1.16
Horizontal	Downward	1.21	0 44
Vertical		1.52*	0 74

Section B. Conductance of Vertical Spaces at Various Mean Temperatures[b]

Mean Temp Deg Fahr	Conductances of Air Spaces for Various Widths in Inches						
	0 128	0.250	0 364	0 493	0 713	1.00	1 500
20	2 300	1.370	1.180	1.100	1 040	1 030	1.022
30	2.385	1.425	1.234	1.148	1 080	1 070	1.065
40	2 470	1.480	1.288	1.193	1 125	1 112	1 105
50	2 560	1.535	1.340	1.242	1 168	1.152	1.149
60	2.650	1.590	1.390	1.295	1.210	1.195	1 188
70	2.730	1.648	1.440	1 340	1.250	1 240	1.228
80	2 819	1.702	1.492	1.390	1 295	1 280	1.270
90	2 908	1.757	1.547	1.433	1 340	1 320	1 310
100	2 990	1.813	1.600	1 486	1 380	1.362	1 350
110	3.078	1.870	1.650	1 534	1.425	1 402	1.392
120	3.167	1.928	1.700	1 580	1.467	1.445	1.435
130	3.250	1 980	1.750	1 630	1.510	1.485	1 475
140	3.340	2.035	1 800	1 680	1.550	1 530	1 519
150	3 425	2 090	1.852	1 728	1 592	1 569	1 559

Section C. Conductances and Resistances of Air Spaces Faced on One Surface with Reflective Insulation[c]

Location and Position of Air Space	Direction of Heat Flow	Temp[d] Diff Deg Fahr		Conductance[e] (C)			Resistance[e] $\left(\frac{1}{C}\right)$		
				No of Air Spaces			No of Air Spaces		
		Winter	Summer	1	2	3	1	2	3
Rafter Space (8 in)									
Horizontal	Down	45			0 10	0 07		10 00	14 29
Horizontal	Up	45			0 27	0 17		3 70	5 88
Horizontal	Down		25		0 09	0 06		11 11	16 67
Horizontal	Up		25		0 24	0 16		4 17	6.25
30 deg slope	Down	45			0 15	0 10		6 67	10 00
30 deg slope	Up	45			0 25	0 17		4 00	5 88
30 deg slope	Down		25		0 13	0 09		7 69	11 11
30 deg slope	Up		25		0 23	0 14		4 35	7 14
Stud Space (3⅝ in)									
Vertical[f]		30		0 34			2 94		
Vertical		40			0 23	0 13		4 35	7 69
Vertical[f]			15	0 32			3 13		
Vertical			20		0 18	0 11		5 56	9 09
Vertical[g]		30		0.46			2 17		

[a]Radiation and Convection from Surfaces in Various Positions, by G B Wilkes and C M. F Peterson (A S H V E Transactions, Vol. 44, 1938, p 513)

[b]A S H.V E. Research Report No 825—Thermal Resistance of Air Spaces, by F. B Rowley and A. B Algren (A.S.H V.E Transactions, Vol 35, 1929, p 165).

[c]Thermal Test Coefficients of Aluminum Insulation for Buildings, by G B Wilkes, F. G Hechler and E R Queer (A S H.V.E. Transactions, Vol 46, 1940)

[d]Temperature difference is based on total space between plaster base and sheathing, flooring or roofing.

[e]These air space conductance and resistance values are based on one reflective surface (aluminum) having an emissivity of 0 05 facing each space and are based on total space between plaster base and sheathing, flooring or roofing. The rafter and stud spaces are divided into equal spaces.

[f]Stud space is lined on plaster base side with loose paper with aluminum on surface facing air space. The resistance of the small air space between the plaster base and paper was 0 43.

[g]Radiation and Convection Across Air Spaces in Frame Construction, by G B Wilkes and C M F. Peterson (A S H.V.E. Transactions, Vol 43, 1937, p. 351).

*The recommended surface conductance for calculating heat losses for still air for non-reflective surfaces is 1.65 Btu. For a 15 mph wind velocity, the recommended value is 6 0 Btu. These coefficients were derived from Fig. 1 which was based on tests conducted at the University of Minnesota, and apply to vertical surfaces

in the emissivity coefficient due to surface coatings or chemical action[6] should be investigated as to the permanence of the reflective surface for the conditions under which this material will be used. In making installations of this material the partitions between air spaces should be tight, particularly at the top and bottom so that air cannot circulate between adjacent spaces.

When reflective insulating materials are installed with multiple air spaces, the position (vertical, horizontal or inclined) of the material in the structure must be taken into consideration. For example, the resistance to heat flow upward is about one-third that of downward flow in a horizontal position in the same construction, as will be apparent from Table 1, **Section C.** However, the difference between upward heat flow through single horizontal or sloping air spaces and through single vertical air spaces is comparatively small for the same temperature difference. Consequently the same conductance value (0.46) may be used for computing coefficients involving upward heat flow through horizontal and sloping air spaces bounded on one side by aluminum foil applied to plasterboard, as for similar vertical air spaces.

As already stated, a conductance value of 1.10 was similarly used in all cases for calculating the coefficients of construction involving vertical, horizontal and sloping air spaces bounded on both sides by ordinary building materials.

PRACTICAL COEFFICIENTS

For practical purposes it is necessary to have average coefficients that may be applied to various materials and types of construction without the necessity of making tests on the individual material or combination of materials. In Table 2 coefficients are given for a group of materials which have been selected from various sources. Wherever possible the properties of material and conditions of tests are given. However, in selecting and applying these values to any construction a reasonable amount of caution is necessary; variations will be found in the coefficients for the same materials, which may be partly due to different test methods used, but which are largely due to variations in materials. It is recommended that future determinations of thermal conductivity be made in accordance with the code, Standard Method of Test for Thermal Conductivity of Materials by Means of the Guarded Hot Plate[7]. The coefficients which have been used for the calculation of over-all coefficients are given in Table 3.

It should be recognized that in these tables of calculated coefficients space limitations will not permit the inclusion of all the combinations of materials that are used in building construction and the varied applications of insulating materials to these constructions. Typical examples are given of combinations frequently used, but any special construction not given in Tables 4 to 16 can generally be computed by using the conductivity values given in Table 3 and the fundamental heat transfer formulas. For example, the tabulation of all of the values for multiple layers of insulating materials would present extensive and detailed problems of calculations for the varied application combinations, but the engineer having the fundamental conductivity values can quickly obtain the proper coefficients.

[6]Thermal Test Coefficients of Aluminum Insulation for Buildings, by G B Wilkes, F G. Hechler and E R. Queer (A S H V E TRANSACTIONS, Vol 46, 1940, p 109)

[7]Sponsored by A S H V.E, *A S T M*, *A S R E* and *N R C* and approved as a Tentative Code by A S H.V E and *A S T M* in 1942 (*A.S T M.* Designation of Code is C177-42T).

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING AND INSULATING MATERIALS

These constants are expressed in Btu per hour per square foot per degree Fahrenheit temperature difference Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
BUILDING BOARDS (NON-INSULATING)	Compressed cement and asbestos sheets	123	86	2.70		0.37		(1)
	Corrugated asbestos board	20.4	110	0.48		2.08		(2)
	Pressed asbestos mill board	60.5	86	0.84		1.19		(3)
	Gypsum board—gypsum between layers of heavy paper	62.8	70	1.41		0.71*		(3)
	⅜ in gypsum board				3.73		0.27*	
	½ in gypsum board				2.82		0.35	
	½ in gypsum board	53.5	90		2.60		0.38	(1)
FRAME CONSTRUCTION COMBINATIONS	1 in fir sheathing and building paper		30		0.86		1.16*	(4)
	1 in fir sheathing, building paper and yellow pine lap siding		20		0.50		2.00*	(4)
	1 in fir sheathing, building paper and stucco		20		0.82		1.22	(4)
	Pine lap siding and building paper, siding 4 in wide		16		0.85		1.18*	(4)
	Yellow pine lap siding				1.28		0.78*	(4)
MASONRY MATERIALS Brick	Damp or wet			5.0*		0.20*		(2)
	Common yellow clay brick[a]			4.8		0.21		(4)
	One tier yellow common clay brick, one tier face brick, approx 8 in thick[a]				0.77		1.30	(4)
Clay Tile, Hollow	2 in Tile, ½ in plaster both sides	120.0	110		1.00		1.00	(2)
	4 in Tile, ½ in plaster both sides	127.0	100		0.60		1.67	(2)
	6 in Tile, ½ in plaster both sides	124.3	105		0.47		2.13	(2)
	8 in Tile, average of 8 types (Walls No 59, 63, 64, 66, 67, 90, 91, 92[a])				0.52		1.92	(4)
	12 in Clay tile wall 8 in x 5 in x 12 in and 4 in x 5 in x 12 in[a]				0.26		3.84	(4)
Concrete	Sand and gravel aggregate, various ages and mixes			11.35 to 16.36		0.09 to 0.06		(5)
	Sand and gravel aggregate	142	75	12.6		0.08		(4)
	Limestone aggregate	132	75	10.8		0.09		(4)
	Cinder aggregate	97	75	4.9		0.22		(4)
	Steam treated limestone slag aggregate[a]	74.6	75	2.27		0.44		(4)
	Pumice (Mined in California) aggregate[a]	65.0	75	2.42		0.41		(4)
	Expanded burned clay aggregate[a]	59.9	75	2.28		0.44		(4)
	Burned clay aggregate[a]	67.1	75	2.86		0.35		(4)
	Blast furnace slag aggregate	76.0	70	1.6		0.63		(3)
	Expanded vermiculite aggregate	20	90	0.68		1.47		(3)
	Expanded vermiculite aggregate	26.7	90	0.76		1.32		(3)

AUTHORITIES:

[1]U. S. Bureau of Standards, tests based on samples submitted by manufacturers

[2]A C. Willard, L C Lichty and L A Harding, tests conducted at the University of Illinois

[3]J C Peebles, tests conducted at Armour Institute of Technology, based on samples submitted by manufacturers

[4]F. B. Rowley, et al, tests conducted at the University of Minnesota

[5]A S H V.E Research Laboratory

[6]E A Allcut, tests conducted at the University of Toronto

[a]See Thermal Conductivity of Building Materials, by F B Rowley and A B Algren (University of Minnesota *Engineering Experiment Station Bulletin* No 12)

[b]Heat Transmission Through Insulation as Affected by Orientation of Wall, by F B Rowley and C E Lund (A S H V E TRANSACTIONS, Vol 49, 1943, p 331)

[c]The Effect of Convection in Ceiling Insulation, by G B Wilkes and L R Vianey (A S H V E TRANSACTIONS, Vol 49, 1943, p 196).

[d]See A S H V E. RESEARCH REPORT No. 915—Conductivity of Concrete, by F C. Houghten and Carl Gutberlet (A S H V E TRANSACTIONS, Vol. 38, 1932, p. 47).

[e]Recommended value (See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932).

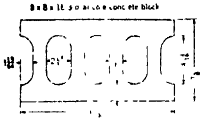
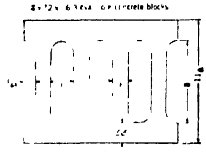
[f]See BMS13, U. S Department of Commerce, National Bureau of Standards, Washington, D. C

[g]Roofing, 0.15 in thick (1 34 lb per square foot), covered with gravel (0 83 lb per square foot), combined thickness assumed 0.25

*Recommended Value for computing heat transmission coefficients See also Table 3

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING AND INSULATING MATERIALS—Continued

These constants are expressed in Btu per hour per square foot per degree Fahrenheit temperature difference. Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
MASONRY MATERIALS—(Continued) CONCRETE—(Continued)	Expanded vermiculite aggregate	35	90	0.86		1.16		(3)
	Expanded vermiculite aggregate	50	90	1.10		0.91		(3)
	Concrete plank	76	75	2.5		0.40		(3)
	Cellular concrete	40.0	75	1.06		0.94		(3)
	Cellular concrete	50.0	75	1.44		0.69		(3)
	Cellular concrete	60.0	75	1.80		0.56		(3)
	Cellular concrete	70.0	75	2.18		0.46		(3)
8 In. Concrete Blocks 	8 in. three oval core, sand and gravel aggregate[a]	126.4	40		0.90		1.11	(4)
	8 in. three oval core, crushed limestone aggregate[a]	134.3	40		0.86		1.16	(4)
	8 in. three oval core, cinder aggregate[a]	86.2	40		0.58		1.73	(4)
	8 in. three oval core, burned clay aggregate[a]	67.7	40		0.50		2.00	(4)
	8 in. three oval core, expanded blast furnace slag aggregate[a]		40		0.49		2.04	(4)
12 In. Concrete Blocks 	12 in. three oval core, sand and gravel aggregate[a]	124.9	40		0.78		1.28	(4)
	12 in. three oval core, cinder aggregate[a]	86.2	40		0.53		1.88	(4)
	12 in. three oval core, burned clay aggregate[a]	76.7	40		0.47		2.13	(4)
GYPSUM	3 in. solid gypsum partition tile[a]			2.41		0.42		(4)
	3 in. three cell gypsum partition tile[a]				0.74		1.35	(4)
	4 in. three cell gypsum partition tile[a]				0.60		1.67	(4)
	87½ per cent gypsum, 12½ per cent wood chips	51.2	74	1.66		0.60		(4)
	Gypsum plaster			3.30		0.30		
PLASTERING MATERIALS	Gypsum plaster, ⅜ in. thick		73		8.80		0.11	(4)
	Cement plaster			8.00		0.13		(2)
	Wood, lath and plaster, total thickness ¾ in.		70		2.50		0.40*	(4)
	Gypsum plaster and expanded vermiculite, 4 to 1 mix	39.9	75	0.85		1.18		(3)
	Insulating plaster 0.9 in. thick applied to ⅜ in. gypsum board	54.0	75		1.07	..	0.93	(3)
ROOFING	Asbestos shingles	65.0	75		6.0		0.17*	(3)
	Asphalt, composition or prepared	70.0	75		6.5		0.15*	(3)
	Asphalt shingles	70.0	75		6.5		0.15*	(3)
	Built-up roofing, bitumen or felt, gravel or slag surfaced[a]			1.33		0.75		(2)
	Slate			10.00		0.10*		
	Wood shingles				1.28		0.78*	
WOODS	Balsa	20.0	90	0.58		1.72		(1)
	Balsa	8.8	90	0.38		2.63		(1)
	Balsa	7.3	90	0.33		3.03		(1)
	California redwood, 0 per cent moisture[a]	28.0	75	0.70		1.43		(4)
	Cypress	28.7	86	0.67		1.49		(1)
	Douglas fir, 0 per cent moisture[a]	34.0	75	0.67		1.49		(4)
	Eastern hemlock, 0 per cent moisture[a]	30.0	75	0.76		1.32		(4)
	Long leaf yellow pine, 0 per cent moisture[a]	40.0	75	0.86		1.16		(4)
	Mahogany	34.3	86	0.90		1.11		(1)

See notes on Page 88.

Table 2. Conductivities (k) and Conductances (C) of Building and Insulating Materials—Continued

These constants are expressed in Btu per hour per square foot per degree Fahrenheit temperature difference. Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
WOODS—(Continued)	Hard maple, 0 per cent moisture[a]	46 0	75	1 05		0 95		(4)
	Maple	44 3	86	1.10		0 91		(1)
	Maple, across grain	40 0	75	1 20		0 83		(3)
	Norway pine, 0 per cent moisture[a]	32.0	75	0.74		1 35		(4)
	Red cypress, 0 per cent moisture[a]	32 0	75	0 79		1 27		(4)
	Red oak, 0 per cent moisture[a]	48 0	75	1 18		0 85		(4)
	Short leaf yellow pine, 0 per cent moisture[a]	36 0	75	0 91		1 10		(4)
	Soft elm, 0 per cent moisture[a]	34 0	75	0 88		1 14		(4)
	Soft maple, 0 per cent moisture[a]	42 0	75	0 95		1 05		(4)
	Sugar pine, 0 per cent moisture[a]	28 0	75	0 64		1 56		(4)
	Virginia pine	34 3	86	0 96		1 04		(1)
	West coast hemlock, 0 per cent moisture[a]	30 0	75	0.79		1 27		(4)
	White pine	31 2	86	0 78		1.28		(1)
	Yellow pine			1 00		1 00		(3)
	Sawdust, various	12 0	90	0 41		2 44		(1)
	Shavings, various from planer	8 8	90	0 41		2 44		(1)
	Shavings, from maple beech and birch (coarse)	13 2	90	0 36		2 78		(1)
INSULATING MATERIALS Blanket and Bat Insulations	Chemically treated wood fibers held between layers of strong paper	3 62	70	0 25		4 00		(3)
	Eel grass between strong paper	4 60	90	0 26		3 85		(1)
	Eel grass between strong paper	3 40	90	0 25		4 00		(1)
	Flax fibers between strong paper	4 90	90	0 28		3 57		(1)
	Chemically treated hog hair between kraft paper	5 76	71	0 26		3 85		(3)
	Chemically treated hog hair between kraft paper and asbestos paper	7 70	71	0 28		3 57		(3)
	Hair felt between layers of paper	11 00	75	0.25		4 00		(3)
	Kapok between burlap or paper	1 00	90	0 24		4 17		(1)
	Stitched and creped expanding fibrous blanket	1 50	70	0 27		3 70		(3)
	Paper and asbestos fiber with emulsified asphalt binder	4 2	94	0 28		3 57		(1)
	Cotton insulating bat	0 875	72	0.24		4 17		(3)
	Felted cattle hair	13 00	90	0 26		3 84		(1)
	Felted cattle hair	11 00	90	0 26		3 84		(1)
	Felted hair and asbestos	7 80	90	0 28		3 57		(1)
	Ground paper between two layers, each ⅜ in thick made up of two layers of kraft paper (sample ¾ in thick)	12 1	75		0 40		2 50	(4)
Reflective	See Table 1, **Section C**							
Insulating Board	Made from sugar cane fiber	13 5	70	0 33		3 03		(3)
	Made from corn stalks	15 00	71	0 33		3 03		(3)
	Made from exploded wood fibers	17 90	78	0 32		3 12		(4)
	Made from hard wood fibers	15 20	70	0 32		3 12		(3)
	Made from wood fiber	15 90	72	0 33		3 03		(3)
	Made from wood fiber	15 00	70	0 33		3 03		(3)
	Made from wood fiber		52	0 33		3 03		(6)
	Made from wood fiber	8 50	72	0 29		3 45		(3)
	Made from wood fiber	15 20		0 33		3 03		(3)
	Made from wood fiber	16 90	90	0 34		2 94		(1)
	Made from licorice root	16 1	81	0 34		2 94		(3)
	½ in insulating boards without special finish[f] (eleven samples)	16 5 to 21 8	90	0 33 to 0 40		3 03 to 2 50		(1)
	1 in insulating board[a]	13 2		0 34		2 94		(4)

See notes on Page 88.

Table 2. Conductivities (k) and Conductances (C) of Building and Insulating Materials—Concluded

These constants are expressed in Btu per hour per square foot per degree Fahrenheit temperature difference Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness.

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
INSULATING MATERIALS —Continued								
Loose Fill Type	Made from ceiba fibers	1.90	75	0.23		4.35		(3)
	Made from ceiba fibers	1.60	75	0.24		4.17		(3)
	Cotton fibers							
	Short Staple Linters, Fireproofed	6.25	90	0.25		4.00		(1)
	Short Staple Linters, Fireproofed	4.50	90	0.24		4.17		(1)
	Short Staple Linters, Fireproofed	2.45	90	0.24		4.17		(1)
	Short Staple Linters, Fireproofed	1.60	90	0.26		3.85		(1)
	Short Staple Linters, Fireproofed	0.85	90	0.29		3.45		(1)
	Short Staple Linters, Fireproofed	0.65	90	0.30		3.33		(1)
	Fibrous material made from dolomite and silica	1.50	75	0.27		3.70		(3)
	Fibrous material made from slag	9.40	103	0.27		3.70		(1)
	Redwood bark	3.00	90	0.31		3.22		(1)
	Redwood bark	5.00	75	0.26		3.84		(3)
	Glass wool fibers 0.0003 in to 0.006 in in diameter	1.50	75	0.27		3.70		(3)
	Granular insulation made from combined silicate of lime and alumina	4.20	72	0.24		4.17		(3)
	Expanded vermiculite			0.48		2.08		(1)
	Expanded vermiculite, particle size— −3 + 14	6.2		0.32		3.12		(3)
	Regranulated cork about 3/16 in particles	8.10	90	0.31		3.22		(1)
	Hand applied granular mineral wool 2 in to 6 in thick, horizontal position[b] No covering	6.05 to 7.13		0.30 to 0.33		3.33 to 3.03		(4)
	4 in machine blown granular mineral wool, horizontal position[b] No covering	5.74		0.30		3.33		(4)
	Rock wool	10.0	90	0.27		3.70		(1)
Slab Insulations	Corkboard, no added binder	14.0	90	0.34		2.94		(1)
	Corkboard, no added binder	10.6	90	0.30		3.33*		(1)
	Corkboard, no added binder	7.0	90	0.27		3.70		(1)
	Corkboard, no added binder	5.4	90	0.25		4.00		(1)
	Corkboard[a]	8.7		0.29		3.45		(4)
	Corkboard, asphaltic binder	14.5	90	0.32		3.12		(1)
	Chemically treated hog hair with film of asphalt	10.0	75	0.28		3.57		(3)
	Sugar cane fiber insulation blocks encased in asphalt membrane	13.8	70	0.30		3.33		(3)
	Made from 85 per cent magnesia and 15 per cent asbestos	19.3	86	0.51		1.96		(1)
	Made from shredded wood and cement	24.2	72	0.46		2.17		(3)
	Made from shredded wood and cement[a]	29.8		0.77		1.30		(4)

See notes on Page 88.

In order to determine the benefit derived from the addition of insulation to a given wall, ceiling, floor, or roof construction, it is necessary to compare the over-all coefficient of heat transmission (U_1) of the insulated construction with the coefficient (U) of the construction without insulation. Attention is called to the necessity of applying the insulation in accordance with the manufacturer's specifications. Certain types of blanket insulations are designed to be installed between the studs of a frame building in such manner as to give two air spaces. In order to get the full value of such materials they should be so installed that each air space is approximately ¾ in. or more in thickness and the air spaces should be sealed at the top and bottom to prevent the circulation of air from one space to the other.

The engineer must carefully evaluate the economic considerations involved in the selection of an insulating material as adapted to various

Table 3. Conductivities (k) and Conductances (C) Used in Calculating Heat Loss Coefficients (U) in Tables 4 to 16

These constants are expressed in Btu per hour per square foot per degree Fahrenheit temperature difference. Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness

MATERIAL	DESCRIPTION	Conductivity or Conductance		Resistance	
		(k)	(C)	Per Inch Thickness $\left(\frac{1}{k}\right)$	For Thickness Listed $\left(\frac{1}{C}\right)$
AIR SPACES					
Bounded by ordinary materials	Vertical[a], ¾ in. or more in width		1.10		0.91
Bounded by aluminum foil	Vertical[a], ¾ in. or more in width		0.46		2.17
EXTERIOR FINISHES (Frame Walls)					
Brick Veneer	4 in. thick (nominal)		2.27		0.44
Stucco (1 in.)		12.50		0.08	
Wood Shingles			1.28		0.78
Yellow Pine Lap Siding			1.28		0.78
INSULATING MATERIALS					
Aluminum Foil	See Air Spaces				
Bats	Enclosed both sides	0.27		3.70	
Blankets	Made from mineral or vegetable fibers or animal hair	0.27		3.70	
Corkboard	Pure, no added binder	0.30		3.33	
Insulating Board		0.33		3.03	
Mineral Wool		0.27		3.70	
Vermiculite		0.48		2.08	
INTERIOR FINISHES					
Composition Wallboard	3/16 in. to ⅜ in. thick	0.50		2.00	
Gypsum Plaster		3.30		0.30	
Gypsum Board (⅜ in.)	Plain or decorated		3.70		0.27
Gypsum Lath (⅜ in.) and Plaster	Plaster thickness assumed ½ in.		2.4		0.42
Insulating Board (½ in.)	Plain or decorated		0.66		1.52
Insulating Board Lath (½ in.) and Plaster	Plaster thickness assumed ½ in.		0.60		1.67
Insulating Board Lath (1 in.) and Plaster	Plaster thickness assumed ½ in.		0.31		3.18
Metal Lath and Plaster	Plaster thickness assumed ¾ in.		4.40		0.23
Plywood (⅜ in.)	Plain or decorated		2.12		0.47
Wood Lath and Plaster			2.50		0.40
MASONRY MATERIALS					
Brick	Adobe	3.56		0.28	
Brick	Common	5.00		0.20	
Brick	Face	9.20		0.11	
Cement Mortar		12.00		0.08	
3 in. Clay tile (hollow)			1.28		0.78
4 in. Clay tile (hollow)			1.00		1.00
6 in. Clay tile (hollow)			0.64		1.57
8 in. Clay tile (hollow)			0.60		1.67
10 in. Clay tile (hollow)			0.58		1.72
12 in. Clay tile (hollow)			0.40		2.50
16 in. Clay tile (hollow)			0.31		3.23
Concrete	Light weight aggregate[b]	2.50		0.40	
Concrete	Sand and gravel aggregate	12.00		0.08	
3 in. Concrete blocks	Hollow, cinder aggregate		1.28		0.78
4 in. Concrete blocks	Hollow, cinder aggregate		1.00		1.00
8 in. Concrete blocks	Hollow, gravel aggregate		1.00		1.00
12 in. Concrete blocks	Hollow, gravel aggregate		0.80		1.25
8 in. Concrete blocks	Hollow, cinder aggregate		0.60		1.66
12 in. Concrete blocks	Hollow, cinder aggregate		0.53		1.88
8 in. Concrete blocks	Hollow, light weight aggregate[b]		0.50		2.00
12 in. Concrete blocks	Hollow, light weight aggregate[b]		0.47		2.13
Gypsum fiber concrete	87½ per cent gypsum and 12½ per cent wood chips	1.66		0.60	
3 in. Gypsum tile	Hollow		0.61		1.64
4 in. Gypsum tile	Hollow		0.46		2.18
Stucco		12.50		0.08	
Tile and Terrazzo	For flooring	12.00		0.08	

[a]Conductance values for horizontal air spaces depend on whether the heat flow is upward or downward, but in most cases it is sufficiently accurate to use the same values for horizontal as for vertical air spaces.

[b]Expanded slag, burned clay or pumice

TABLE 3. CONDUCTIVITIES (k) AND CONDUCTANCES (C) USED IN CALCULATING HEAT LOSS COEFFICIENTS (U) IN TABLES 4 TO 16—Concluded

These constants are expressed in Btu per hour per square foot per degree Fahrenheit temperature difference Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness.

MATERIAL	DESCRIPTION	CONDUCTIVITY OR CONDUCTANCE		RESISTANCE	
				Per Inch Thickness	For Thickness Listed
		(k)	(C)	$\left(\frac{1}{k}\right)$	$\left(\frac{1}{C}\right)$
ROOFING MATERIALS					
ASBESTOS SHINGLES			6.00		0.17
ASPHALT SHINGLES			6.50		0.15
BUILT-UP ROOFING	Assumed thickness ⅜ in		3.53		0.28
HEAVY ROLL ROOFING			6.50		0.15
SLATE		10.00		0.10	
WOOD SHINGLES			1.28		0.78
SHEATHING					
GYPSUM (½ IN.)			2.82		0.35
INSULATING BOARD (25/32 IN.)			0.42		2.37
PLYWOOD (5/16 IN.)			2.56		0.39
FIR OR YELLOW PINE (1 IN.)	Actual thickness 25/32 in.		1.02		0.98
FIR, PLUS BUILDING PAPER	Actual thickness 25/32 in		0.86		1.16
SURFACES					
STILL AIR	Ordinary non-reflective materials, vertical		1.65		0.61
15 MPH WIND VELOCITY	Ordinary non-reflective materials, vertical		6.00		0.17
WOODS					
FIR SHEATHING (1 IN.) BUILDING PAPER AND YELLOW PINE LAP SIDING			0.50		2.00
MAPLE OR OAK		1.15		0.87	
YELLOW PINE OR FIR		0.80		1.25	

building constructions Lack of good judgment in the intelligent choice of an insulating material, or its improper installation, frequently represents the difference between good or unsatisfactory results.

Computed Transmission Coefficients

Computed heat transmission coefficients of many common types of building construction are given in Tables 4 to 16, inclusive, each construction being identified by a serial number. For example, the coefficient of transmission (U) of an 8-in. brick wall and ½ in. of plaster is 0.46, and the number assigned to a wall of this construction is *67-B*, Table 6.

Example 1 Calculate the coefficient of transmission (U) of an 8-in. brick wall with ½ in of plaster applied directly to the interior surface, based on an outside wind exposure of 15 mph It is assumed that the outside course is of hard (high density) brick having a conductivity of 9.20 and that the inside course is of common (low density) brick having a conductivity of 5.0, the thicknesses each being 4 in The conductivity of the plaster is assumed to be 3.3, and the inside and outside surface coefficients are assumed to average 1.65 and 6.00, respectively, for still air and a 15 mph wind velocity.

Solution. k (hard high density brick) = 9.20, x = 4.0 in ; k (common low density brick) = 5.0, x = 4.0 in , k (plaster) = 3.3, x = ½ in.; f_i = 1.65, f_o = 6.0 Therefore,

$$U = \frac{1}{\frac{1}{6.0} + \frac{4.0}{9.20} + \frac{4.0}{5.0} + \frac{0.5}{3.3} + \frac{1}{1.65}} = \frac{1}{0.167 + 0.435 + 0.80 + 0.152 + 0.606}$$

= 0.46 Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides.

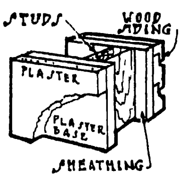
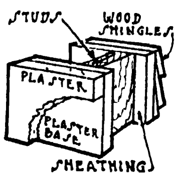
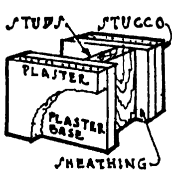
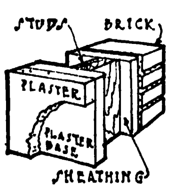
The coefficients in the tables were determined by calculations similar to those shown in *Example 1*, using fundamental Formulas 1, 2 and 3 and the values of k (or C), f_i, f_o and a in Table 3.

Actual thicknesses of lumber are used in the computations rather than

TABLE 4. COEFFICIENTS OF TRANSMISSION (U) OF FRAME WALLS

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

NO INSULATION BETWEEN STUDS[a] (SEE TABLE 5)

EXTERIOR FINISH	INTERIOR FINISH	TYPE OF SHEATHING				WALL NUMBER
		GYPSUM (½ IN THICK)	PLYWOOD (5/16 IN. THICK)	WOOD[f] (25/32 IN. THICK) BLDG. PAPER	INSULATING BOARD (25/32 IN THICK)	
		A	B	C	D	
WOOD SIDING (Clapboard) 	Metal Lath and Plaster[b]	0.33	0.32	0.26	0.20	1
	Gypsum Board (⅜ in.) Decorated	0.32	0.32	0.25	0.20	2
	Wood Lath and Plaster	0.31	0.31	0.25	0.19	3
	Gypsum Lath (⅜ in.) Plastered[c]	0.31	0.31	0.25	0.19	4
	Plywood (⅜ in.) Plain or Decorated	0.30	0.30	0.24	0.19	5
	Insulating Board (½ in.) Plain or Decorated	0.23	0.23	0.19	0.16	6
	Insulating Board Lath (½ in.) Plastered[c]	0.22	0.22	0.19	0.15	7
	Insulating Board Lath (1 in.) Plastered[c]	0.17	0.17	0.15	0.12	8
WOOD[d] SHINGLES 	Metal Lath and Plaster[b]	0.25	0.25	0.26	0.17	9
	Gypsum Board (⅜ in.) Decorated	0.25	0.25	0.25	0.17	10
	Wood Lath and Plaster	0.24	0.24	0.25	0.16	11
	Gypsum Lath (⅜ in.) Plastered[c]	0.24	0.24	0.25	0.16	12
	Plywood (⅜ in.) Plain or Decorated	0.24	0.24	0.24	0.16	13
	Insulating Board (½ in.) Plain or Decorated	0.19	0.19	0.19	0.14	14
	Insulating Board Lath (½ in.) Plastered[c]	0.19	0.18	0.19	0.13	15
	Insulating Board Lath (1 in.) Plastered[c]	0.14	0.14	0.15	0.11	16
STUCCO 	Metal Lath and Plaster[b]	0.43	0.42	0.32	0.23	17
	Gypsum Board (⅜ in.) Decorated	0.42	0.41	0.31	0.23	18
	Wood Lath and Plaster	0.40	0.39	0.30	0.22	19
	Gypsum Lath (⅜ in.) Plastered[c]	0.39	0.39	0.30	0.22	20
	Plywood (⅜ in.) Plain or Decorated	0.39	0.38	0.29	0.22	21
	Insulating Board (½ in.) Plain or Decorated	0.27	0.27	0.22	0.18	22
	Insulating Board Lath (½ in.) Plastered[c]	0.26	0.26	0.22	0.17	23
	Insulating Board Lath (1 in.) Plastered[c]	0.19	0.19	0.16	0.14	24
BRICK VENEER[e] 	Metal Lath and Plaster[b]	0.37	0.36	0.28	0.21	25
	Gypsum Board (⅜ in.) Decorated	0.36	0.36	0.28	0.21	26
	Wood Lath and Plaster	0.35	0.34	0.27	0.20	27
	Gypsum Lath (⅜ in.) Plastered[c]	0.34	0.34	0.27	0.20	28
	Plywood (⅜ in.) Plain or Decorated	0.34	0.33	0.27	0.20	29
	Insulating Board (½ in.) Plain or Decorated	0.25	0.25	0.21	0.17	30
	Insulating Board Lath (½ in.) Plastered[c]	0.24	0.24	0.20	0.16	31
	Insulating Board Lath (1 in.) Plastered[c]	0.18	0.18	0.15	0.13	32

[a]Coefficients not weighted; effect of studding neglected.
[b]Plaster assumed ¾ in. thick.
[c]Plaster assumed ½ in. thick.
[d]Furring strips between wood shingles and all sheathings except wood
[e]Small air space and mortar between building paper and brick veneer neglected
[f]Nominal thickness, 1 in.

Table 5. Coefficients of Transmission (*U*) of Frame Walls with Insulation Between Framing[a,b]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

Coefficient with *no* Insulation between Framing	Coefficient with Insulation between Framing: Blanket or Bat Insulation between Framing[c] (Thickness below): 1 in	2 in	3 in	3⅝ in. Loose Mineral Wool between Framing	Number
	A	B	C	D	
0.11	0.078	0.064	0.055	0.051	33
0.12	0.083	0.067	0.057	0.054	34
0.13	0.088	0.070	0.059	0.056	35
0.14	0.092	0.073	0.061	0.058	36
0.15	0.097	0.075	0.062	0.059	37
0.16	0.10	0.077	0.065	0.060	38
0.17	0.10	0.080	0.066	0.062	39
0.18	0.11	0.082	0.068	0.063	40
0.19	0.11	0.084	0.069	0.064	41
0.20	0.12	0.087	0.070	0.066	42
0.21	0.12	0.088	0.072	0.067	43
0.22	0.12	0.090	0.073	0.069	44
0.23	0.12	0.093	0.074	0.069	45
0.24	0.12	0.094	0.076	0.070	46
0.25	0.13	0.095	0.076	0.072	47
0.26	0.13	0.096	0.077	0.072	48
0.27	0.14	0.097	0.078	0.073	49
0.28	0.14	0.098	0.078	0.073	50
0.29	0.14	0.10	0.080	0.075	51
0.30	0.14	0.10	0.080	0.075	52
0.31	0.14	0.10	0.082	0.076	53
0.32	0.15	0.10	0.082	0.076	54
0.33	0.15	0.11	0.083	0.077	55
0.34	0.15	0.11	0.083	0.078	56
0.35	0.15	0.11	0.085	0.078	57
0.36	0.16	0.11	0.085	0.079	58
0.37	0.16	0.11	0.087	0.080	59
0.38	0.16	0.11	0.087	0.080	60
0.39	0.16	0.11	0.087	0.081	61
0.40	0.16	0.11	0.088	0.082	62
0.41	0.16	0.11	0.088	0.082	63
0.42	0.16	0.11	0.088	0.082	64
0.43	0.17	0.11	0.090	0.083	65
0.44	0.17	0.12	0.090	0.083	66

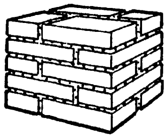
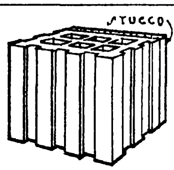
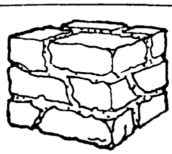
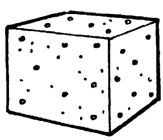
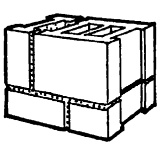
[a]This table may be used for determining the coefficients of transmission of frame constructions with the types and thicknesses of insulation indicated in Columns A to D inclusive between framing. Columns A, B and C may be used for walls, ceilings or roofs with only one air space between framing but are not applicable to ceilings with *no* flooring above. (See Table 10.) Column D is applicable to walls only. *Example:* Find the coefficient of transmission of a frame wall consisting of wood siding, ²⁵⁄₃₂ in insulating board sheathing, studs, gypsum lath and plaster, with 2 in blanket insulation between studs. According to Table 4, a wall of this construction with *no insulation* between studs has a coefficient of 0.19 (Wall No. 4D). Referring to Column B above, it will be found that a wall of this value with 2 in blanket insulation between the studs has a coefficient of 0.084.

[b]Coefficients corrected for 2 x 4 framing, 16 in. o. c.—15 per cent of surface area.

[c]Based on one air space between framing.

TABLE 6. COEFFICIENTS OF TRANSMISSION (*U*) OF MASONRY WALLS

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph

TYPE OF MASONRY		THICKNESS OF MASONRY INCHES	INTERIOR FINISH (PLUS INSULATION WHERE INDICATED) Plain Walls—No Interior Finish	Plaster (½ in.) on Walls	Metal Lath and Plaster[f] Furred	Gypsum Board (⅜ in.) Decorated—Furred	Gypsum Lath (⅜ in.) Plastered[g]—Furred	Insulating Board (½ in.) Plain or Decorated—Furred	Insulating Board Lath (½ in.) Plastered[g]—Furred	Insulating Board Lath (1 in.) Plastered[g]—Furred	Gypsum Lath[g] Plastered Plus 1 in. Blanket Insulation—Furred[h]	WALL NUMBER
			A	B	C	D	E	F	G	H	I	
SOLID[a] BRICK		8	0.50	0.46	0.32	0.31	0.30	0.22	0.22	0.16	0.14	67
		12	0.35	0.34	0.25	0.25	0.24	0.19	0.19	0.14	0.13	68
		16	0.28	0.27	0.21	0.21	0.20	0.17	0.16	0.13	0.12	69
HOLLOW[b] TILE (Stucco Exterior Finish)		8	0.40	0.37	0.27	0.27	0.26	0.20	0.20	0.15	0.13	70
		10	0.39	0.37	0.27	0.27	0.26	0.20	0.19	0.15	0.13	71
		12	0.30	0.29	0.22	0.22	0.21	0.17	0.17	0.13	0.12	72
		16	0.25	0.24	0.19	0.19	0.19	0.16	0.15	0.12	0.11	73
STONE[c]		8	0.70	0.64	0.39	0.38	0.36	0.26	0.25	0.18	0.16	74
		12	0.57	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	75
		16	0.49	0.45	0.32	0.31	0.30	0.22	0.22	0.16	0.14	76
		24	0.37	0.35	0.26	0.26	0.25	0.20	0.19	0.15	0.13	77
POURED CONCRETE[d]		6	0.79	0.71	0.42	0.41	0.39	0.27	0.26	0.19	0.16	78
		8	0.70	0.64	0.39	0.38	0.36	0.26	0.25	0.18	0.16	79
		10	0.63	0.58	0.37	0.36	0.34	0.25	0.24	0.18	0.15	80
		12	0.58	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	81
HOLLOW CONCRETE BLOCKS		Gravel Aggregate										
		8	0.56	0.52	0.34	0.34	0.32	0.24	0.19	0.17	0.15	82
		12	0.50	0.46	0.32	0.31	0.30	0.22	0.22	0.16	0.14	83
		Cinder Aggregate										
		8	0.41	0.39	0.28	0.28	0.27	0.21	0.20	0.15	0.13	84
		12	0.38	0.36	0.26	0.26	0.25	0.20	0.19	0.15	0.13	85
		Light[e] Weight Aggregate										
		8	0.36	0.34	0.26	0.25	0.24	0.19	0.19	0.15	0.13	86
		12	0.34	0.33	0.25	0.24	0.24	0.19	0.18	0.14	0.13	87

[a] Based on 4 in. hard brick and remainder common brick

[b] The 8 in. and 10 in. tile figures are based on two cells in the direction of heat flow. The 12 in. tile is based on three cells in the direction of heat flow. The 16 in. tile consists of one 10 in. and one 6 in. tile each having two cells in the direction of heat flow.

[c] Limestone or sandstone

[d] These figures may be used with sufficient accuracy for concrete walls with stucco exterior finish

[e] Expanded slag, burned clay or pumice

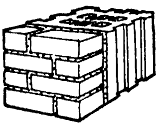
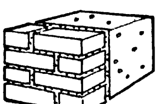
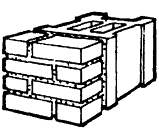
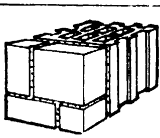
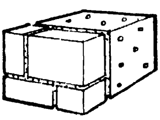
[f] Thickness of plaster assumed ¾ in.

[g] Thickness of plaster assumed ½ in.

[h] Based on 2 in. furring strips; one air space. These figures may also be used for wood or metal lath.

Table 7. Coefficients of Transmission (U) of Brick and Stone Veneer Masonry Walls

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph

Typical Construction	Facing	Backing	Plain Walls—no Interior Finish	Plaster (½ in) on Walls	Metal Lath and Plaster[e]—Furred	Gypsum Board (⅜ in) Decorated—Furred	Gypsum Lath (⅜ in) Plastered[f]—Furred	Insulating Board (½ in) Plain or Decorated—Furred	Insulating Board Lath (½ in) Plastered[f]—Furred	Insulating Board Lath (1 in) Plastered[f]—Furred	Gypsum Lath Plastered[f] Plus 1 in Blanket Insulation—Furred[g]	Wall Number
			A	**B**	**C**	**D**	**E**	**F**	**G**	**H**	**I**	
	4 in. Brick Veneer[a]	6 in. Hollow Tile[b]	0.35	0.34	0.25	0.25	0.24	0.19	0.18	0.14	0.13	88
		8 in. Hollow Tile[b]	0.34	0.32	0.25	0.24	0.23	0.19	0.18	0.14	0.13	89
		6 in. Concrete	0.59	0.54	0.35	0.35	0.33	0.24	0.23	0.17	0.15	90
		8 in. Concrete	0.54	0.50	0.33	0.33	0.31	0.23	0.23	0.17	0.15	91
		8 in. Concrete Blocks[c] (Gravel Aggregate)	0.44	0.41	0.29	0.29	0.28	0.21	0.21	0.16	0.14	92
		8 in. Concrete Blocks[c] (Cinder Aggregate)	0.34	0.33	0.25	0.24	0.24	0.19	0.18	0.14	0.13	93
		8 in. Concrete Blocks[c] (Light Weight Aggregate)[d]	0.31	0.29	0.23	0.23	0.22	0.18	0.17	0.14	0.12	94
	4 in. Cut Stone Veneer[a]	6 in. Hollow Tile[b]	0.37	0.35	0.26	0.26	0.25	0.19	0.19	0.15	0.13	95
		8 in. Hollow Tile[b]	0.36	0.34	0.25	0.25	0.24	0.19	0.19	0.15	0.13	96
		6 in. Concrete	0.63	0.58	0.37	0.36	0.34	0.25	0.24	0.18	0.15	97
		8 in. Concrete	0.57	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	98
		8 in. Concrete Blocks[c] (Gravel Aggregate)	0.47	0.44	0.30	0.30	0.29	0.22	0.21	0.16	0.14	99
		8 in. Concrete Blocks[c] (Cinder Aggregate)	0.36	0.34	0.25	0.25	0.24	0.19	0.19	0.15	0.13	100
		8 in. Concrete Blocks[c] (Light Weight Aggregate)[d]	0.32	0.30	0.23	0.23	0.22	0.18	0.18	0.14	0.12	101

[a] Calculations based on ½ in cement mortar between backing and facing except in the case of the concrete backing which is assumed to be poured in place.

[b] The hollow tile figures are based on two air cells in the direction of heat flow.

[c] Hollow concrete blocks

[d] Expanded slag, burned clay or pumice.

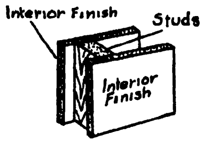
[e] Thickness of plaster assumed ¾ in.

[f] Thickness of plaster assumed ½ in

[g] Based on 2 in furring strips; one air space The figures in this column may also be used for wood or metal lath

TABLE 8. COEFFICIENTS OF TRANSMISSION (U) OF FRAME PARTITIONS OR INTERIOR WALLS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.



INTERIOR FINISH	SINGLE PARTITION (Finish on one side only of studs)	DOUBLE PARTITION (Finish on both sides of studs)		PARTITION NUMBER
		NO INSULATION BETWEEN STUDS	1 IN. BLANKET BETWEEN STUDS. ONE AIR SPACE.	
	A	B	C	
Metal Lath and Plaster[b]	0 69	0 39	0.16	1
Gypsum Board (⅜ in) Decorated	0 67	0 37	0 16	2
Wood Lath and Plaster	0 62	0 34	0.15	3
Gypsum Lath (⅜ in) Plastered[c]	0.61	0.34	0.15	4
Plywood (⅜ in) Plain or Decorated	0.59	0 33	0 15	5
Insulating Board (½ in.) Plain or Decorated	0 36	0 19	0 11	6
Insulating Board Lath (½ in.) Plastered[c]	0 35	0 18	0 11	7
Insulating Board Lath (1 in.) Plastered[c]	0 19	0 12	0 082	8

[a]Coefficients not weighted; effect of studding neglected
[b]Plaster assumed ¾ in. thick
[c]Plaster assumed ½ in. thick
[d]For partitions with other insulations between studs refer to Table 5, using values in Column B of above table in left hand column of Table 5 *Example:* What is the coefficient of transmission (U) of a partition consisting of gypsum lath and plaster on both sides of studs with 2 in blanket between studs? *Solution.* According to above table, this partition with no insulation between studs (No 4B) has a coefficient of 0 34. Referring to Table 5, it will be found that a wall having a coefficient of 0.34 with no insulation between studs, will have a coefficient of 0.11 with 2 in. of blanket insulation between studs (No 56B).

nominal thicknesses. The computations for wood shingle roofs applied over wood stripping are based on 1 by 4 in. wood strips, spaced 2 in. apart. Since no reliable figures are available concerning the conductivity of Spanish and French clay roofing tile, of which there are many varieties, the figures for such types of roofs were taken the same as for slate roofs, as it is probable that the values of U for these two types of roofs will compare favorably.

The thicknesses upon which the coefficients in Tables 4 to 17 inclusive, are based are:

Brick veneer	4 in.	1-in. lumber (S-2-S)	25/32 in.
Plaster and metal lath	¾ in.	1½-in. lumber (S-2-S)	1 5/16 in.
Plaster (on wood lath, gypsum lath or insulating board)	½ in.	2-in. lumber (S-2-S)	1⅝ in.
Slate (roofing)	½ in.	2½-in. lumber (S-2-S)	2⅛ in.
Stucco on wire mesh reinforcing	1 in.	3-in. lumber (S-2-S)	2⅝ in.
Tar and gravel or slag-surfaced built-up roofing	⅜ in.	4-in. lumber (S-2-S)	3⅝ in.
		Finish flooring (maple or oak)	13/16 in.

Solid brick walls are based on 4 in. hard brick (high density) and the remainder common brick (low density). Stucco is assumed to be 1 in. thick on masonry walls. Where metal lath and plaster are specified, the metal lath is neglected.

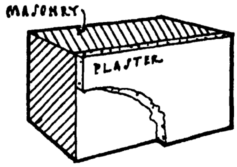
The coefficients of transmission of the pitched roofs in Table 15 apply where the roof is over a heated attic or top floor so the heat passes directly through the roof structure including whatever finish is applied to the underside of the roof rafters.

Coefficients for frame construction were corrected for the effect of the framing where such correction would *increase* the coefficients, but not where the correction would *decrease* the coefficient[8]. In the latter case,

[8]Effect of Studs and Joists on Heat Flow Through Frame Walls and Ceilings, by Paul D. Close (*Heating, Piping and Air Conditioning*, October, 1943, p 529).

TABLE 9. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY PARTITIONS

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.



TYPE OF PARTITION		THICKNESS OF MASONRY (INCHES)	TYPE OF FINISH: NO FINISH (Plain walls) A	PLASTER ONE SIDE B	PLASTER BOTH SIDES[a] C	PARTITION NUMBER
HOLLOW CLAY TILE		3	0.50	0 47	0.43	9
		4	0 45	0 42	0 40	10
HOLLOW GYPSUM TILE		3	0 35	0 33	0.32	11
		4	0.29	0 28	0 27	12
HOLLOW CONCRETE TILE OR BLOCKS	Cinder Aggregate	3	0 50	0 47	0.43	13
		4	0.45	0 42	0 40	14
	Light Weight Aggregate	3	0.41	0.39	0 37	15
		4	0.36	0 34	0 32	16
COMMON BRICK		4	0.50	0 46	0.43	17

[a]2 in. solid plaster partition. $U = 0\ 53$
[b]Expanded slag, burned clay or pumice

the correction is generally small and the uncorrected coefficient is on the side of safety. Although theoretical coefficients below 0.10 are included in the tables, a minimum coefficient of 0.10 is generally recommended to allow for possible defects in workmanship, poor construction and other factors which would increase the heat loss. The lower the theoretical wall or roof coefficient the greater will be the percentage of error due to construction defects or failure of the insulation to perform as rated.

Combined Coefficients of Transmission

If the attic is unheated, the roof structure and ceiling of the top floor must both be taken into consideration, and the combined coefficient of transmission determined. The formula for calculating the combined coefficient of transmission of a top floor ceiling, unheated attic space, and pitched roof per square foot of ceiling area is:

$$U = \frac{U_r \times U_{ce}}{U_r + \frac{U_{ce}}{n}} \tag{4}$$

where

U = combined coefficient to be used with ceiling area.
U_r = coefficient of transmission of the roof.
U_{ce} = coefficient of transmission of the ceiling.
n = the ratio of the area of the roof to the area of the ceiling.

In selecting the values to be used for U_r and U_{ce} it should be noted that the under surface of the roof and the upper surface of the ceiling are more nearly equivalent to the boundary surfaces of an internal air space than they are to the external surfaces of a wall. It would be more nearly correct to use a value of 2.2 rather than the usual value of 1.65 as coefficients for these surfaces. In most cases this would make only a minor change in U. It should be noted that the over-all coefficient should be multiplied by the ceiling and not the roof area.

If the unheated attic space between the roof and ceiling has no dormers, windows or vertical wall spaces the combined coefficient may be used for

determining the heat loss through the roof construction, attic and top floor ceiling. If the unheated attic contains windows and vertical wall spaces these must be taken into consideration in calculating the roof area and also its coefficient U_r. In this case an approximate value of U_r may be obtained as the summation of the coefficient of each individual section such as the roof, vertical walls or windows times its percentage of total area. This coefficient may then be used with reasonable accuracy in Equation 4. If there are large vertical wall areas, the most accurate procedure is to estimate the attic temperature by means of Equation 1, Chapter 6 and to calculate the heat loss by using the ceiling coefficient only, and the attic temperature instead of the outside temperature. Information concerning basement floor and wall coefficients will be found on Page 100.

The values in Table 16 apply principally to hip and double hip roofs with unheated attics. In the case of gable and gambrel (Dutch Colonial) roofs having end wall surfaces, the values in Table 16 can be used with reasonable accuracy if the end wall area is considered as part of the roof area. Thus in arriving at the combined roof and ceiling coefficient for a pitched roof with insulating board applied to the under side of the rafters, the board must also be applied to the end wall surfaces in order for the values in Table 16 to be approximately applicable.

The coefficients in Table 16 are based on ⅓ pitch roofs for which the value of n, or the ratio of the roof area to the ceiling area, is 1.2. Since these combined roof and ceiling coefficients are based on the ceiling area, small variations in the pitch do not greatly alter the coefficient. An increase in pitch above the ⅓ used for calculating the values would increase the combined coefficient, and vice versa

Basement Floor and Wall Coefficients

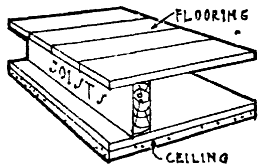
The dirt in contact with basement floors and walls below grade has an appreciable insulating value. The exact thickness or depth of the dirt which may contribute to the heat resistance of floors and walls in contact therewith is a variable and indeterminate quantity. Tests at the A.S.H.V.E. Research Laboratory[9] indicate that the rate of heat flow through basement floors, even without insulation, does not exceed 0.10 Btu per square foot per degree Fahrenheit temperature difference between the air above the floor and the soil under the floor at the point of maximum diffusion of the heat from the basement. As indicated in the footnote of Table 12, this coefficient is recommended for use whenever it is desirable to make allowance for the small basement floor heat loss, as may be the case with heated basements. Using this coefficient (0.10) the total heat loss per square foot of floor area will amount to only 2.0 Btu per square foot per hour, based on a temperature difference of 20 F between the air in the basement above the floor and the minimum soil temperature below the basement.

The same coefficient (0.10) may be used for calculating the heat loss through basement walls below grade but a somewhat higher temperature difference should be used for reasons explained in Chapter 6. Thus the total heat loss through basement walls will be greater than that through floors. According to present available data the unit wall heat loss will be

[9]A S H V.E RESEARCH REPORT No 1213—Heat Loss Through Basement Walls and Floors, by F C Houghten, S. I. Taimuty, Carl Gutberlet and C J Brown (A S H V E TRANSACTIONS, Vol 48, 1942, p. 369).

TABLE 10. COEFFICIENTS OF TRANSMISSION (U) OF FRAME CONSTRUCTION CEILINGS AND FLOORS

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides and are based on still air (no wind) conditions on both sides



TYPE OF CEILING	INSULATION BETWEEN, OR ON TOP OF, JOISTS (NO FLOORING ABOVE)												WITH FLOORING[g] (ON TOP OF CEILING JOISTS)		NUMBER
	None	Insulating Board on Top of Joists		Blanket or Bat Insulation[f] Between Joists[a]			Vermiculite Insulation Between Joists			Mineral Wool Insulation Between Joists			Single Wood Floor[b]	Double Wood Floor[c]	
		½ In	1 In	1 In	2 In	3 In.	2 In.	3 In	4 In	2 In	3 In	4 In			
	A	B	C	D	E	F	G	H	I	J	K	L	M	N	
No Ceiling		0.37	0.24										0.45	0.34	1
Metal Lath and Plaster[d]	0.69	0.26	0.19	0.19	0.12	0.092	0.18	0.14	0.11	0.12	0.092	0.066	0.30	0.25	2
Gypsum Board (⅜ in.) Plain or Decorated	0.67	0.26	0.18	0.19	0.12	0.092	0.18	0.14	0.10	0.12	0.092	0.066	0.30	0.24	3
Wood Lath and Plaster	0.62	0.25	0.18	0.19	0.12	0.092	0.17	0.14	0.10	0.12	0.092	0.065	0.29	0.24	4
Gypsum Lath (⅜ in) Plastered[e]	0.61	0.25	0.18	0.19	0.12	0.092	0.17	0.14	0.10	0.12	0.092	0.065	0.28	0.24	5
Plywood (⅜ in) Plain or Decorated	0.59	0.24	0.18	0.19	0.12	0.09	0.17	0.14	0.10	0.12	0.090	0.065	0.28	0.23	6
Insulating Board (½ in) Plain or Decorated	0.37	0.19	0.15	0.16	0.11	0.082	0.15	0.11	0.092	0.11	0.082	0.060	0.22[h]	0.19[h]	7
Insulating Board Lath (½ in.) Plastered[e]	0.35	0.19	0.15	0.15	0.10	0.082	0.14	0.11	0.091	0.10	0.082	0.060	0.21	0.18	8
Insulating Board Lath (1 in) Plastered[e]	0.23	0.15	0.12	0.12	0.089	0.072	0.12	0.097	0.080	0.089	0.072	0.055	0.16	0.14	9

[a]Coefficients corrected for framing

[b]$^{25}/_{32}$ in. yellow pine or fir.

[c]$^{25}/_{32}$ in. pine or fir sub-flooring plus $^{13}/_{16}$ in hardwood finish flooring

[d]Plaster assumed ¾ in thick

[e]Plaster assumed ½ in thick

[f]Based on insulation in contact with ceiling and consequently no air space between

[g]For coefficients for constructions in Columns M and N (except No 1) with insulation between joists, refer to Table 5 *Example*: The coefficient for No. 3-N of Table 10 is 0.24. With 2 in blanket insulation between joists, the coefficient will be 0 094 (See Table 5) (Column D of Table 5 applicable only for 3⅝ in joists)

[h]For $^{25}/_{32}$ in insulating board sheathing applied to the under side of the joists, the coefficient for *single* wood floor (Column M) is 0 18 and for *double* wood floor (Column N) is 0 16 For coefficients with insulation between joists, see Table 5

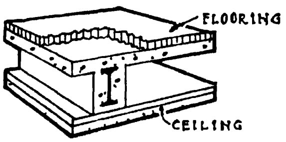
approximately twice the floor heat loss under average conditions in northern latitudes or about 4.0 Btu per hour per square foot (total, not per degree temperature difference) between the basement air and the ground at a level corresponding to the mid-height of the basement wall. Until further data are available, this value may be used with reasonable accuracy for calculating heat losses through basement walls of heated basements.

CONDENSATION IN BUILDINGS

The water vapor in buildings will be transmitted through many types of building construction if there is a difference in the vapor pressures on

TABLE 11. COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE CONSTRUCTION FLOORS AND CEILINGS

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.



TYPE OF CEILING	THICKNESS OF CONCRETE[g] (INCHES)	TYPE OF FLOORING: No Flooring (Concrete Bare)	Tile[a] or Terrazzo Flooring on Concrete	¼ In Battleship[b] Linoleum Directly on Concrete	Parquet[c] Flooring In Mastic on Concrete	Double Wood Floor on Sleepers[d]	NUMBER
		A	B	C	D	E	
No Ceiling	3	0.69	0.65	0.45	0.45	0.25	1
	6	0.59	0.56	0.41	0.41	0.23	2
	10	0.50	0.48	0.36	0.36	0.22	3
½ in. Plaster Applied to Underside of Concrete.	3	0.62	0.59	0.43	0.43	0.24	4
	6	0.54	0.52	0.39	0.39	0.22	5
	10	0.46	0.44	0.34	0.34	0.21	6
Metal Lath and Plaster[e]—Suspended or Furred	3	0.38	0.37	0.30	0.30	0.19	7
	6	0.35	0.34	0.28	0.28	0.18	8
	10	0.32	0.31	0.26	0.26	0.17	9
Gypsum Board (⅜ in) and Plaster[f]—Suspended or Furred	3	0.36	0.35	0.28	0.28	0.19	10
	6	0.33	0.32	0.27	0.27	0.18	11
	10	0.30	0.29	0.24	0.24	0.17	12
Insulating Board Lath (½ in) and Plaster[f] Suspended or Furred	3	0.25	0.24	0.21	0.21	0.15	13
	6	0.23	0.23	0.20	0.20	0.15	14
	10	0.22	0.21	0.19	0.19	0.14	15

[a]Thickness of tile assumed to be 1 in

[b]The figures in Column C may be used with sufficient accuracy for concrete floors covered with carpet

[c]Thickness of wood assumed to be ¹³⁄₁₆ in ; thickness of mastic, ⅛ in ($k = 4.5$)

[d]Based on ²⁵⁄₃₂ in. yellow pine or fir sub-flooring and ¹³⁄₁₆ in. hardwood finish flooring with an air space between sub-floor and concrete.

[e]Thickness of plaster assumed to be ¾ in.

[f]Thickness of plaster assumed to be ½ in

[g]For other thicknesses of concrete, interpolate.

TABLE 12. COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE FLOORS ON GROUND WITH VARIOUS TYPES OF FINISH FLOORING

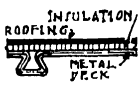
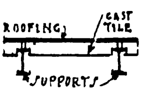
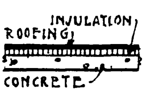
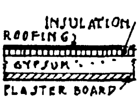
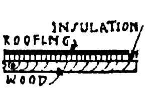
$U = 0.10^a$ Btu per hour per square foot per degree Fahrenheit temperature difference between the ground and the air over the floor.

[a]Until more complete data are available, based on tests now in progress, it is recommended that a coefficient of 0.10 be used for all types of concrete floors on the ground, with or without insulation For basement wall below grade, use the same average coefficient (0.10). A lower ground temperature should, however, be used for walls than floors as explained in Chapter 6 For further data see A.S H.V.E. RESEARCH REPORT No 1213—Heat Loss Through Basement Walls and Floors, by F. C. Houghten, S I. Taimuty, Carl Gutberlet and C J Brown (A S H V E TRANSACTIONS, Vol 48, 1942, p 369).

TABLE 13. COEFFICIENTS OF TRANSMISSION (U) OF FLAT ROOFS COVERED WITH BUILT-UP ROOFING. NO CEILING—UNDER SIDE OF ROOF EXPOSED

(See Table 14 for Flat Roofs with Ceilings)

These coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph.

TYPE OF ROOF DECK	THICKNESS OF ROOF DECK (INCHES)	NO INSULATION	INSULATION ON TOP OF DECK (COVERED WITH BUILT-UP ROOFING)							NUMBER
			INSULATING BOARD (Thickness Below)				CORKBOARD (Thickness Below)			
			½ In.	1 In.	1½ In.	2 In.	1 In.	1½ In.	2 In.	
		A	B	C	D	E	F	G	H	
Flat Metal Roof Deck[a] 		0.94	0.39	0.24	0.18	0.14	0.23	0.17	0.13	1
Precast Cement Tile 	1⅝ in	0.84	0.37	0.24	0.18	0.14	0.22	0.16	0.13	2
Concrete 	2 in	0.82	0.37	0.24	0.17	0.14	0.22	0.16	0.13	3
	4 in	0.72	0.34	0.23	0.17	0.13	0.21	0.16	0.12	4
	6 in	0.65	0.33	0.22	0.16	0.13	0.21	0.15	0.12	5
Gypsum and Wood Fiber[b] on ⅜ in. Gypsum Board 	2⅝ in.	0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11	6
	3⅜ in.	0.32	0.22	0.16	0.13	0.11	0.15	0.12	0.10	7
Wood[c] 	1 in.	0.49	0.28	0.20	0.15	0.12	0.19	0.14	0.12	8
	1½ in.	0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11	9
	2 in.	0.32	0.22	0.16	0.13	0.11	0.16	0.12	0.10	10
	3 in	0.23	0.17	0.14	0.11	0.096	0.13	0.11	0.091	11

[a]Coefficient of transmission of bare corrugated iron (no roofing) is 1.50 Btu per hour per square foot of projected area per degree Fahrenheit difference in temperature, based on an outside wind velocity of 15 mph.

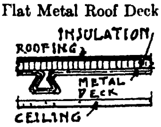
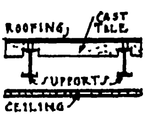
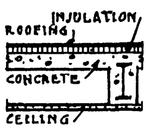
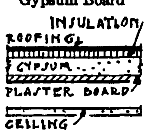
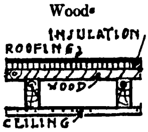
[b]87½ per cent gypsum, 12½ per cent wood fiber. Thickness indicated includes ⅜ in. gypsum board.

[c]Nominal thicknesses specified—actual thicknesses used in calculations.

Table 14 Coefficients of Transmission (*U*) of Flat Roofs Covered with Built-up Roofing. With Lath and Plaster Ceilings[a]

(See Table 13 for Flat Roofs with No Ceilings)

These coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph

Type of Roof Deck	Thickness of Roof Deck (Inches)	No Insulation	Insulation on Top of Deck (Covered with Built-Up Roofing)							Number
			Insulating Board (Thickness Below)				Corkboard (Thickness Below)			
			½ In	1 In	1½ In	2 In	1 In	1½ In	2 In	
		A	B	C	D	E	F	G	H	
Flat Metal Roof Deck 		0.46	0.27	0.19	0.15	0.12	0.18	0.14	0.11	12
Precast Cement Tile 	1⅝ in.	0.43	0.26	0.19	0.15	0.12	0.18	0.14	0.11	13
Concrete 	2 in.	0.42	0.26	0.19	0.15	0.12	0.18	0.14	0.11	14
	4 in.	0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11	15
	6 in.	0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11	16
Gypsum and Wood Fiber[b] on ⅜ in. Gypsum Board 	2⅜ in.	0.27	0.19	0.15	0.12	0.10	0.14	0.12	0.097	17
	3⅜ in.	0.23	0.17	0.14	0.11	0.097	0.13	0.11	0.091	18
Wood[c] 	1 in.	0.32	0.21	0.16	0.13	0.11	0.15	0.12	0.10	19
	1½ in.	0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.095	20
	2 in.	0.24	0.17	0.14	0.11	0.097	0.13	0.11	0.092	21
	3 in.	0.18	0.14	0.12	0.10	0.087	0.11	0.096	0.082	22

[a]Calculations based on metal lath and plaster ceilings, but coefficients may be used with sufficient accuracy for gypsum lath or wood lath and plaster ceilings It is assumed that there is an air space between the under side of the roof deck and the upper side of the ceiling

[b]87½ per cent gypsum, 12½ per cent wood fiber. Thickness indicated includes ⅜ in gypsum board

[c]Nominal thicknesses specified—actual thicknesses used in calculations

Table 15. Coefficients of Transmission (U) of Pitched Roofs

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides and are based on a wind velocity of 15 mph

Type of Ceiling (Applied Directly to Roof Rafters)	Wood Shingles (On 1 x 4 Wood Strips[c] Spaced 2 In Apart)				Asphalt Shingles or Roll Roofing (On Solid Wood Sheathing)[c]				Slate or Tile[b] (On Solid Wood Sheathing)[c]				Number
	Insulation Between Rafters				Insulation Between Rafters				Insulation Between Rafters				
	None	Blanket or Bat (Thickness Below)			None	Blanket or Bat (Thickness Below)			None	Blanket or Bat (Thickness Below)			
		1 In	2 In	3 In		1 In	2 In	3 In		1 In	2 In.	3 In	
	A	B	C[a]	D[a]	E	F	G[a]	H[a]	I	J	K[a]	L[a]	
No Ceiling Applied to Rafters	0.48[f]	0.15	0.11	0.083	0.53[f]	0.15	0.11	0.085	0.55[f]	0.16	0.11	0.085	1
Metal Lath and Plaster[d]	0.31	0.14	0.10	0.082	0.33	0.15	0.10	0.083	0.34	0.15	0.11	0.083	2
Gypsum Board (3/8 in.) Decorated	0.31	0.14	0.10	0.082	0.32	0.15	0.10	0.082	0.34	0.15	0.11	0.083	3
Wood Lath and Plaster	0.30	0.14	0.10	0.080	0.31	0.14	0.10	0.082	0.32	[a]0.15	0.10	0.082	4
Gypsum Lath (3/8 in.) Plastered[e]	0.29	0.14	0.10	0.080	0.31	0.14	0.10	0.082	0.32	0.15	0.10	0.082	5
Plywood (3/8 in.) Plain or Decorated	0.29	0.14	0.10	0.080	0.30	0.14	0.10	0.080	0.31	0.14	0.10	0.082	6
Insulating Board (1/2 in.) Plain or Decorated	0.22	0.12	0.090	0.073	0.23	0.12	0.093	0.074	0.24	0.13	0.094	0.076	7
Insulating Board Lath (1/2 in.) Plastered[e]	0.22	0.12	0.090	0.073	0.22	0.12	0.090	0.073	0.23	0.12	0.093	0.074	8
Insulating Board Lath (1 in.) Plastered[e]	0.16	0.10	0.077	0.065	0.17	0.10	0.077	0.065	0.17	0.10	0.080	0.066	9

[a]Coefficients corrected for framing.
[b]Figures in Columns I, J, K and L may be used with sufficient accuracy for rigid asbestos shingles on wood sheathing. Layer of slater's felt neglected.
[c]Sheathing and wood strips assumed 25/32 in. thick.
[d]Plaster assumed 3/4 in. thick.
[e]Plaster assumed 1/2 in. thick.
[f]No air space included in 1-A, 1-E or 1-I; all other coefficients based on one air space.

Table 16. Combined Coefficients of Transmission (U) of Pitched Roofs[a] and Horizontal Ceilings—Based on Ceiling Area[b]

Coefficients are expressed in Btu per hour per square foot of ceiling area per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

Ceiling Coefficient[f] (From Table 10)	Type of Roofing and Roof Sheathing: Wood Shingles on Wood Strips[d]			Asphalt Shingles[c] or Roll Roofing on Wood Sheathing[e]			Number
	No Roof Insulation (Rafters Exposed) (U_r = 0.48)	½ In. Insulating Board on Under Side of Rafters (U_r = 0.22)	1 In. Insulating Board on Under Side of Rafters (U_r = 0.16)	No Roof Insulation (Rafters Exposed) (U_r = 0.53)	½ In. Insulating Board on Under Side of Rafters (U_r = 0.23)	1 In. Insulating Board on Under Side of Rafters (U_r = 0.17)	
	A	B	C	D	E	F	
0.10	0.085	0.073	0.066	0.087	0.074	0.067	19
0.11	0.093	0.078	0.07	0.094	0.079	0.071	20
0.12	0.099	0.082	0.074	0.10	0.083	0.075	21
0.13	0.10	0.087	0.077	0.11	0.088	0.079	22
0.14	0.11	0.092	0.080	0.11	0.093	0.083	23
0.15	0.12	0.096	0.084	0.12	0.097	0.086	24
0.16	0.12	0.10	0.087	0.13	0.10	0.089	25
0.17	0.13	0.10	0.090	0.13	0.10	0.092	26
0.18	0.13	0.11	0.093	0.14	0.11	0.095	27
0.19	0.14	0.11	0.096	0.15	0.11	0.098	28
0.20	0.15	0.11	0.098	0.15	0.12	0.10	29
0.21	0.15	0.12	0.10	0.16	0.12	0.10	30
0.22	0.16	0.12	0.10	0.17	0.12	0.11	31
0.23	0.17	0.12	0.10	0.17	0.12	0.11	32
0.24	0.17	0.13	0.11	0.18	0.12	0.11	33
0.25	0.17	0.13	0.11	0.18	0.13	0.11	34
0.26	0.18	0.13	0.11	0.19	0.13	0.11	35
0.27	0.18	0.13	0.11	0.19	0.13	0.12	36
0.28	0.19	0.14	0.11	0.19	0.14	0.12	37
0.29	0.19	0.14	0.12	0.20	0.14	0.12	38
0.30	0.20	0.14	0.12	0.20	0.14	0.12	39
0.34	0.21	0.15	0.12	0.22	0.15	0.13	40
0.35	0.21	0.15	0.12	0.22	0.15	0.13	41
0.36	0.22	0.15	0.12	0.23	0.15	0.13	42
0.37	0.22	0.15	0.13	0.23	0.16	0.13	43
0.45	0.25	0.17	0.13	0.26	0.17	0.14	44
0.59	0.29	0.18	0.14	0.30	0.19	0.15	45
0.61	0.29	0.18	0.14	0.31	0.19	0.15	46
0.62	0.30	0.18	0.15	0.31	0.19	0.15	47
0.67	0.31	0.19	0.15	0.32	0.20	0.16	48
0.69	0.31	0.19	0.15	0.33	0.20	0.16	49

[a]Calculations based on ⅙ pitch roof (n = 1.2) using the following formula

$$U = \frac{U_r \times U_{ce}}{U_r + \frac{U_{ce}}{n}}$$

U = combined coefficient to be used with ceiling area.
U_r = coefficient of transmission of the roof
U_{ce} = coefficient of transmission of the ceiling
n = the ratio of the area of the roof to the area of the ceiling

[b]Use ceiling area (not roof area) with these coefficients

[c]Coefficients in Columns D, E and F may be used with sufficient accuracy for tile, slate and rigid asbestos shingles on wood sheathing

[d]Based on 1 x 4 in strips spaced 2 in apart.

[e]Sheathing assumed 25/32 in. thick.

[f]Values of U_{ce} to be used in this column may be selected from Table 10

the two sides of the structure[10] (see Table 11, Chapter 7). Such water vapor will also condense whenever it comes in contact with surfaces or objects at or below the dew-point temperature. Thus two types of condensation problems are encountered in building practice, namely (1) *Surface condensation* or condensation on the interior building surfaces including the walls, ceiling (or roof) and glass, and (2) *Interstitial condensation* or the transmittance of the vapor through the building materials and condensation of the moisture on surfaces or voids within the materials of construction.

Condensation within the construction as well as condensation on the interior surfaces does not necessarily occur in all buildings but only in

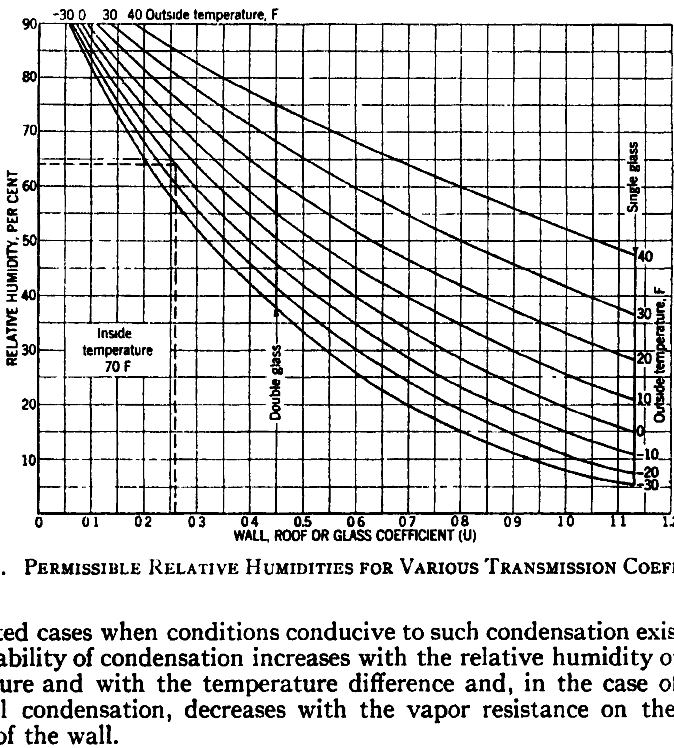


FIG. 2. PERMISSIBLE RELATIVE HUMIDITIES FOR VARIOUS TRANSMISSION COEFFICIENTS

isolated cases when conditions conducive to such condensation exist. The probability of condensation increases with the relative humidity or vapor pressure and with the temperature difference and, in the case of interstitial condensation, decreases with the vapor resistance on the warm side of the wall.

Condensation on interior building surfaces[11] (surface condensation) may be eliminated by either reducing the relative humidity or by maintaining the interior surfaces at or above the dew-point temperature. Permissible relative humidities for various wall, roof or glass coefficients and temperature differences may be determined from Fig. 2. The permissible relative humidity for any specific type of construction may be determined by first ascertaining the coefficient of transmission (U) of the construction and then locating this coefficient on the horizontal scale of Fig. 2. A vertical line drawn to the proper outside temperature curve and then to

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[10]Methods of Moisture Control and Their Application to Building Construction, by F. B. Rowley, A. B. Algren and C. E Lund (University of Minnesota *Engineering Experiment Station Bulletin* No 17).

[11]Permissible Relative Humidities in Humidified Buildings, by Paul D. Close (A.S.H V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1939, p. 766)

the left hand scale will indicate the permissible relative humidity for the conditions involved. The dotted line shown in Fig. 2 indicates the permissible relative humidity (64 per cent) if surface condensation is to be avoided, for a frame wall having a coefficient of 0.26 and for an outside temperature of −10 F.

Condensation within the construction may likewise be prevented by eliminating the moisture at the source or by providing a barrier on the warm side of the insulation construction, which may be a vapor-proof

TABLE 17. COEFFICIENTS OF TRANSMISSION (*U*) OF DOORS, WINDOWS, SKYLIGHTS AND GLASS BLOCK WALLS

Coefficients are based on a wind velocity of 15 mph, and are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air inside and outside of the door, window, skylight or wall

		Single	Double	Triple
Section A. Windows and Skylights	*U*	1.13[ac]	0.45[ae]	0.281[ae]

	Nominal Thickness Inches	Actual Thickness Inches	*U* Exposed Door	*U*[d] With Glass Storm Door
Section B. Solid Wood Doors[bc]	1	25/32	0.69	0.42
	1¼	1 1/16	0.59	0.38
	1½	1 5/16	0.52	0.35
	1¾	1⅜	0.51	0.35
	2	1⅝	0.46	0 32
	2½	2⅛	0 38	0 28
	3	2⅝	0.33	0 25

	Description	*U* Still Air Both Sides	*U* Still Air Inside 15 mph Outside
Section C. Hollow Glass Block Walls	Smooth surface glass blocks 7¾ x 7¾ x 3⅞ in. thick..........	0 40	0 49
	Ribbed surface glass blocks 7¾ x 7¾ x 3⅞ in. thick..........	0.38	0 46

[a]See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932
[b]Computed using $C = 1\ 15$ for wood, $f_i = 1.65$ and $f_o = 6\ 0$
[c]It is sufficiently accurate to use the same coefficient of transmission for doors containing thin wood panels as that of single panes of glass, namely, 1 13 Btu per hour per square foot per degree difference between inside and outside air temperatures
[d]These values may also be used with sufficient accuracy for wood storm doors. Neglect storm doors if loose and use values for exposed doors
[e]Air spaces assumed to be ¾ in. or more in width

paper properly applied under the plaster or a vapor-proof finish on the interior surface of the wall[12]. Insulating laths with vapor barriers applied to the back surface are also available. In the case of attics, the greater the heat resistance in the top floor ceiling, the lower the attic temperature and consequently the greater the tendency for condensation to take place on the underside of the roof boards which moisture will drop on to the ceiling. Thus where thick insulations are installed between ceiling joists, it is desirable to allow openings for outside air circulation through attic space as a precaution against condensation on the underside of the roof even though barriers are used in the ceiling below.

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CHAPTER 5

Air Leakage

Causes of Infiltration, Infiltration Due to Wind Pressure, Infiltration Through Walls, Window and Door Leakage, Crack Method, Air Change Method, Infiltration Due to Temperature Difference, Sealing of Vertical Openings

THE air leakage which takes place through various apertures in buildings must be considered in heating and cooling calculations, and properly evaluated. This *infiltration* as it is sometimes designated takes place through cracks around doors and windows, through solid walls and through fireplaces and chimneys. Although the latter sources of leakage may be considerable, they are often neglected on the assumption that dampers would be closed during periods of extreme cold weather or else that the fireplace will be in use at such times and will therefore contribute to the heat supplied and therefore lessen the heating load.

CAUSES OF INFILTRATION

The displacement of heated air in buildings by unheated outside air is due to two causes, namely, (1) the pressure exerted by the wind and (2) the difference in density of outside and inside air because of differences in temperature. The former is generally referred to as *infiltration* and the latter as *stack* or *chimney effect.*

In either case an exact estimate of the amount of infiltration under design conditions is difficult to make. The complicating factors include (1) variations in building construction particularly as to width of crack or size of openings through which air leakage takes place, (2) the variations in wind velocity and direction, (3) the exposure of the building with respect to air leakage openings and with respect to adjoining buildings, (4) the variations in outside temperatures as influencing the chimney effect, (5) the relative area and resistance of openings on the windward and leeward sides and on the lower floors and on the upper floors, and (6) the influence of a planned air supply and the related outlet vents. Tight construction is essential for preventing large heat loss due to infiltration.

INFILTRATION DUE TO WIND PRESSURE

The wind causes a pressure to be exerted on one or two sides of a building. As a result, air comes into the building on the windward side through cracks or porous construction, and a similar quantity of air leaves on the leeward side through like openings. In general the resistance to air movement is similar on the windward to that on the leeward side. This causes a building up of pressure within the building and a lesser air leakage than that experienced in single wall tests as determined in the laboratory. It is assumed that actual building leakages owing to this building up of pressure will be 80 per cent of laboratory test values. While there are cases where this is not true, tests in actual buildings substantiate the factor for the general case. Mechanical ventilating systems are frequently designed to produce positive or negative pressures in an enclosure which are greater or lower than prevalent wind pressures. In such designs, if the rate at which air is specified to be introduced to or removed from the enclosure by positive means exceeds the infiltration rate, it is common practice to use the greater value in determining the heating capacity to warm the outside air.

Infiltration Through Walls

Data on infiltration through brick and frame walls are given in Table 1[1]. The brick walls listed in this table are walls which show poor workmanship and which are constructed of porous brick and lime mortar. For good workmanship, the leakage through hard brick walls with cement-lime mortar does not exceed one-third the values given. These tests indicate that plastering reduces the leakage by about 96 per cent; a heavy coat of cold water paint, 50 per cent; and 3 coats of oil paint carefully applied, 28 per cent. The infiltration through walls ranges from 6 to 25 per cent of that through windows and doors in a 10-story office building, with imperfect sealing of plaster at the baseboards of the rooms. With perfect sealing the range is from 0.5 to 2.7 per cent or a practically negligible quantity, which indicates the importance of good workmanship in proper sealing at the baseboard. It will be noted from Table 1, that the infiltration through properly plastered walls can be neglected.

The value of building paper when applied between sheathing and

Table 1. Infiltration Through Walls[a]
Expressed in cubic feet per square foot per hour

Type of Wall		Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
8½ in Brick Wall	Plain	2	4	8	12	19	23
	Plastered	0 02	0 04	0 07	0 11	0 16	0 24
13 in. Brick Wall	Plain	1	4	7	12	16	21
	Plastered	0 01	0 01	0 03	0 04	0.07	0 10
Frame Wall, with lath and plaster[b]		0 03	0.07	0.13	0.18	0.23	0.26

[a]The values given in this table are 20 per cent less than test values to allow for building up of pressure in rooms and are based on test data reported in the papers listed in chapter footnotes

[b]Wall construction Bevel siding painted or cedar shingles, sheathing, building paper, wood lath and 3 coats gypsum plaster

shingles is indicated by Fig. 1, which represents the effect on outside construction only, without lath and plaster. The effectiveness of plaster properly applied is no justification for the use of low grade building paper or of the poor construction of the wall containing it. Not only is it difficult to secure and maintain the full effectiveness of the plaster but also it is highly desirable to have two points of high resistance to air flow with an air space between them. The infiltration indicated in Fig. 1 is that determined in the laboratory and should be multiplied by the factor 0.80 to give proper working values.

Window and Door Leakage

There are two methods of estimating air leakage through window and door cracks, namely, (1) the crack method and (2) the air change method.

[1]A S H.V.E. Research Reports No 786—Infiltration Through Plastered and Unplastered Brick Walls, by F. C. Houghten and Margaret Ingels (A.S H V E Transactions, Vol. 33, 1927, p. 377). No. 826—Air Infiltration Through Various Types of Brick Wall Construction, by G L. Larson, D. W. Nelson and C. Braatz (A S.H.V.E. Transactions, Vol 35, 1929, p 183). No. 851—Air Infiltration Through Various Types of Wood Frame Construction, by G L Larson, D. W. Nelson and C Braatz (A S H V.E. Transactions, Vol 36, 1930, p 99)

The crack method is generally regarded as being more accurate than the purely arbitrary air change method, provided the variables entering into the crack method, such as crack width and clearance, can be properly evaluated.

Crack Method

The crack method is based on known air leakage factors for various types of windows and widths of crack and clearance. The wind velocity and length of crack are also considered when the crack method is employed. The amount of infiltration for various types of windows is given in Table 2[2]. The fit of double-hung wood windows is determined by crack and clearance. Crack thickness is equivalent to one-half the difference between the inside window frame dimension and the outside sash width. The difference between the width of the window frame guide and the sash thickness is considered as the clearance. The length of the perimeter

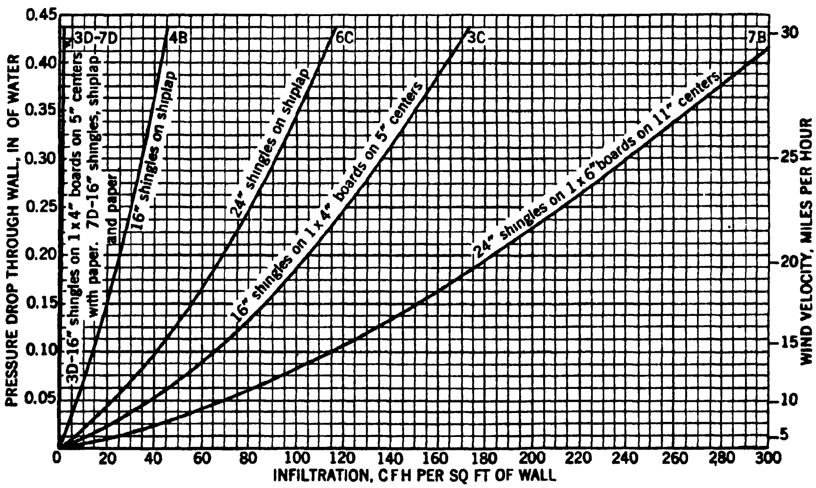

FIG. 1. INFILTRATION THROUGH VARIOUS TYPES OF SHINGLE CONSTRUCTION

opening or crack for a double-hung window is equal to three times the width plus two times the height, or in other words, it is the outer sash perimeter length plus the meeting rail length. Not all the window crack in any given room is necessarily used in estimating the infiltration heat loss by the crack method. The length of crack to be selected in any given case depends on the number of exposed sides as explained in Chapter 6.

Values of leakage shown in Table 2 for the average double-hung wood window were determined by setting the average measured crack and

[2]A.S.H.V.E. RESEARCH REPORTS No 686—Air Leakage, by F. C Houghten and C C Schrader (A S H - V.E. TRANSACTIONS, Vol. 30, 1924, p. 105). No 704—Air Leakage Around Window Openings, by C C Schrader (A.S.H V.E. TRANSACTIONS, Vol. 30, 1924, p 313). No 803—Air Leakage on Metal Windows in a Modern Office Building, by F. C. Houghten and M E O'Connell (A S H V.E TRANSACTIONS, Vol. 34, 1928, p. 321). No. 815—Air Leakage Through a Pivoted Metal Window, by F C Houghten and M. E. O'Connell (A S H.V.E. TRANSACTIONS, Vol 34, 1928, p 519) No. 817—Effect of Frame Calking and Storm Sash on Infiltration Around and Through Windows, by W M Richtmann and C Braatz (A.S.H.V.E. TRANSACTIONS, Vol 34, 1928, p. 547). No 909—Air Infiltration Through Double-Hung Wood Windows, by G. L. Larson, D. W. Nelson and R W Kubasta (A S H V.E. TRANSACTIONS, Vol. 37, 1931, p. 571). The Weathertightness of Rolled Section Steel Windows, by J E Emswiler and W C. Randall (A.S.H.V.E. TRANSACTIONS, Vol 34, 1928, p 527) Pressure Differences Across Windows in Relation to Wind Velocity, by J. E. Emswiler and W. C. Randall (A S H V.E. TRANSACTIONS, Vol 36, 1930, p. 83). Air Infiltration Through Steel Framed Windows, by D. O Rusk, V H Cherry and L. Boelter (A S.H V.E. TRANSACTIONS, Vol. 39, 1933, p. 169)

Table 2. Infiltration Through Windows

Expressed in Cubic Feet per Foot of Crack per Hour[a]

Type of Window	Remarks	Wind Velocity, Miles per Hour: 5	10	15	20	25	30
Double-Hung Wood Sash Windows (Unlocked)	Around frame in masonry wall—not calked[b]	3	8	14	20	27	35
	Around frame in masonry wall—calked[b]	1	2	3	4	5	6
	Around frame in wood frame construction[b]	2	6	11	17	23	30
	Total for average window, non-weatherstripped, 1/16-in crack and 3/64-in clearance[c] Includes wood frame leakage[d]	7	21	39	59	80	104
	Ditto, weatherstripped[d]	4	13	24	36	49	63
	Total for poorly fitted window, non-weatherstripped, 3/32-in crack and 3/32-in clearance[e] Includes wood frame leakage[d].	27	69	111	154	199	249
	Ditto, weatherstripped[d]	6	19	34	51	71	92
Double-Hung Metal Windows[f]	Non-weatherstripped, locked	20	45	70	96	125	154
	Non-weatherstripped, unlocked	20	47	74	104	137	170
	Weatherstripped, unlocked	6	19	32	46	60	76
Rolled Section Steel Sash Windows[k]	Industrial pivoted, 1/16-in crack[g]	52	108	176	244	304	372
	Architectural projected, 1/32-in crack[h]	15	36	62	86	112	139
	Architectural projected, 3/64-in crack[h]	20	52	88	116	152	182
	Residential casement, 1/64-in crack[i]	6	18	33	47	60	74
	Residential casement, 1/32-in crack[i]	14	32	52	76	100	128
	Heavy casement section, projected, 1/64-in crack[j]	3	10	18	26	36	48
	Heavy casement section, projected 1/32-in crack[j]	8	24	38	54	72	92
Hollow Metal, vertically pivoted window[f]		30	88	145	186	221	242

[a]The values given in this table, with the exception of those for double-hung and hollow metal windows, are 20 per cent less than test values to allow for building up of pressure in rooms, and are based on test data reported in the papers listed in chapter footnotes

[b]The values given for frame leakage are per foot of sash perimeter as determined for double-hung wood windows Some of the frame leakage in masonry walls originates in the brick wall itself and cannot be prevented by calking For the additional reason that calking is not done perfectly and deteriorates with time, it is considered advisable to choose the masonry frame leakage values for calked frames as the average determined by the calked and non-calked tests

[c]The fit of the average double-hung wood window was determined as 1/16-in crack and 3/64-in clearance by measurements on approximately 600 windows under heating season conditions.

[d]The values given are the totals for the window opening per foot of sash perimeter and include frame leakage and so-called *elsewhere* leakage The frame leakage values included are for wood frame construction but apply as well to masonry construction assuming a 50 per cent efficiency of frame calking.

[e]A 3/32-in. crack and clearance represent a poorly fitted window, much poorer than average.

[f]Windows tested in place in building.

[g]Industrial pivoted window generally used in industrial buildings Ventilators horizontally pivoted at center or slightly above, lower part swinging out

[h]Architecturally projected made of same sections as industrial pivoted except that outside framing member is heavier and it has refinements in weathering and hardware Used in semi-monumental buildings such as schools Ventilators swing in or out and are balanced on side arms 1/32-in crack is obtainable in the best practice of manufacture and installation, 3/64-in. crack considered to represent average practice

[i]Of same design and section shapes as so-called *heavy section casement* but of lighter weight. 1/64-in crack is obtainable in the best practice of manufacture and installation, 1/32-in crack considered to represent average practice

[j]Made of heavy sections Ventilators swing in or out and stay set at any degree of opening. 1/64-in crack is obtainable in the best practice of manufacture and installation, 1/32-in. crack considered to represent average practice

[k]With reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions is negligible With 3/64-in crack, representing poor installation, leakage at contact with steel framework is about one-third, and at mullions about one-sixth of that given for industrial pivoted windows in the table

clearance of a large number of windows found in a field survey on nine windows tested in the laboratory. In addition, the table gives figures for a poorly fitted window. All of the figures for double-hung wood windows are for the *unlocked* condition. Just how a window is closed, or fits when it is closed, has considerable influence on the leakage. The leakage will be high if the sash are short, if the meeting rail members are warped, or if the frame and sash are not fitted squarely to each other. It is possible to have a window with approximately the average crack and clearance that will have a leakage at least double that of the figures shown. Values for the average double-hung wood window in Table 2 are considered to be easily obtainable figures provided the workmanship on the window is good. Should it be known that the windows under consideration are poorly fitted, the larger leakage values should be used. Locking a window generally decreases its leakage, but in some cases may push the meeting rail members apart and increase the leakage. On windows with large clearances, locking will usually reduce the leakage.

Wood casement windows may be assumed to have the same unit leakage as for the average double-hung wood window when properly fitted. Locking, a normal operation in the closing of this type of window, maintains the crack at a low value.

For metal pivoted sash, the length of crack is the total perimeter of the movable or ventilating sections. Frame leakage on steel windows may be neglected when they are properly grouted with cement mortar into brick work or concrete. When they are not properly sealed, the linear feet of sash section in contact with steel work at mullions should be figured at 25 per cent of the values for industrial pivoted windows as given in Table 2.

When storm sash are applied to well fitted windows, very little reduction in infiltration is secured, but the application of the sash does give an air space which reduces the heat transmission and helps prevent the frosting of the windows[3]. By applying storm sash to poorly fitted windows, a reduction in leakage of 50 per cent may be obtained, the effect so far as air leakage is concerned being roughly equivalent to that obtained by the installation of weatherstrips.

Door Leakage

Doors vary greatly in fit because of their large size and tendency to warp. For a well fitted door, the leakage values for a poorly fitted double-hung wood window may be used. If poorly fitted, twice this figure should be used. If weatherstripped, the values may be reduced one-half. A single door which is frequently opened, such as might be found in a store, should have a value applied which is three times that for a well fitted door. This extra allowance is for opening and closing losses and is kept from being greater by the fact that doors are not used as much in the coldest and windiest weather.

The infiltration rate through swinging and revolving doors is generally a matter of judgment by the engineer making cooling load determinations and in the absence of adequate research data the values given in Table 3 represent current engineering practice[4]. These values are based on the average number of persons in a room at a specified time, which may also

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[3]Fuel Saving Resulting from the Use of Storm Windows and Doors, by A. P. Kratz and S Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 87).

[4]The Infiltration Problem of Multiple Entrances, by A. M. Simpson and K. B. Atkinson (A.S.H.V E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June, 1936, p. 345). Infiltration Characteristics of Entrance Doors, by A. M. Simpson (*Refrigerating Engineering*, June, 1936).

TABLE 3. INFILTRATION THROUGH OUTSIDE DOORS FOR COOLING LOADS[a]
Expressed in Cubic Feet per Minute per Person Entering Room

Application	Pair 36 in. Swinging Doors, Single Entrance[b]	Application	Pair 36 in. Swinging Doors, Single Entrance[b]
Bank	7.5	Hospital Room	3.5
Barber Shop	4.5	Lunch Room	5.0
Broker's Office	7.0	Men's Shop	3.5
Candy and Soda	6.0	Office	3.0
Department Store	8.0	Office Building	2.0
Dress Shop	2.5	Public Building	2.5
Drug Store	7.0	Restaurant	2.5
Furrier	2.5	Shoe Store	3.5

[a]For doors located in only one wall or where doors in other walls are of revolving type.
[b]Vestibules with double pair swinging doors, infiltration may be assumed 75 per cent of swinging door values.
Infiltration for 72 in revolving doors may be assumed 60 per cent of swinging door values.

be the same occupancy assumed for determining the outside ventilation requirements outlined in Chapters 2 and 7.

Air Change Method

The amount of air leakage is sometimes roughly estimated by assuming a certain number of air changes per hour for each room, the number of changes assumed being dependent upon the type, use and location of the room, as indicated in Table 4. This method may be used to advantage as a check on the calculations made in the more exact manner. On the other hand, where it is not possible to determine or pre-determine with accuracy the width of crack or clearance of windows, or where other sources of air leakage cannot readily be evaluated, as is often the case, the use of the air change method may be justified.

The values in Table 4 may be used with reasonable accuracy for residences and are the requirements for each room. The total infiltration allowance for the entire building should be one-half the sum of the infiltration allowances of the individual rooms, since whatever air enters on the windward side generally leaves the building on the leeward side and the infiltration requirements therefore do not exist simultaneously on all sides or in all rooms. An allowance of one air change per hour for all sources of air leakage for the entire volume may be considered average for a well constructed residence.

The air leakage for vestibules due to opening and closing of doors is sometimes based on the air change method, even though the air leakage estimates for other rooms are based on the crack method. Except for vestibules and reception halls, it is not advisable to attempt to apply the

TABLE 4. AIR CHANGES TAKING PLACE UNDER AVERAGE CONDITIONS EXCLUSIVE OF AIR PROVIDED FOR VENTILATION[a]

Kind of Room or Building	Number of Air Changes taking Place per Hour	Kind of Room or Building	Number of Air Changes taking Place per Hour
Rooms, 1 side exposed	1	Rooms with no windows or outside doors	½ to ¾
Rooms, 2 sides exposed	1½	Entrance Halls	2 to 3
Rooms, 3 sides exposed	2	Reception Halls	2
Rooms, 4 sides exposed	2	Bath Rooms	2
		Stores	1 to 3

[a]For rooms with weatherstripped windows or storm sash, use ½ these values, where applicable.

air change method to factories and industrial and commercial buildings because of wide variations in the type and percentage of fenestration which is the principal source of air leakage in such buildings.

INFILTRATION DUE TO TEMPERATURE DIFFERENCE

The air exchange due to temperature difference, inside to outside, is a chimney effect, causing air to enter through openings at lower levels and to leave at higher levels[5]. Although it is not appreciable in low buildings, this loss should be considered in tall, single story buildings with openings near the ground level and near the ceiling. Also in tall, multi-story buildings it may be a considerable item unless the sealing between various floors and rooms is quite perfect.

In tall buildings, temperature difference or chimney effect will produce a head that will add to the effect of the wind at lower levels and subtract from it at higher levels. On the other hand, the wind velocity at lower levels may be somewhat abated by surrounding obstructions. Furthermore, the chimney effect is reduced in multi-story buildings by the partial isolation of floors, thereby preventing free upward movement, so that wind and temperature difference may seldom cooperate to the fullest extent. Making the rough assumption that the *neutral zone*[6] is located at mid-height of a building, and that the temperature difference is 70 F, Equations 1 and 2 may be used to determine an equivalent wind velocity to be used in connection with Tables 1 and 2 that will allow for both wind velocity and temperature difference:

$$V_e = \sqrt{V^2 - 1.75\,a} \tag{1}$$

$$V_e = \sqrt{V^2 + 1.75\,b} \tag{2}$$

where

V_e = equivalent wind velocity to be used in conjunction with Tables 1 and 2.
V = wind velocity upon which infiltration would be determined if temperature difference were disregarded.
a = distance of windows under consideration from mid-height of building if *above* mid-height, feet.
b = distance if *below* mid-height, feet.

The coefficient 1.75 allows for about one-half the temperature difference head.

For buildings of unusual height, Equation 1 would indicate negative infiltration at the highest stories, which condition may, at times, actually exist.

Sealing of Vertical Openings

In tall, multi-story buildings, every effort should be made to seal off vertical openings such as stair-wells and elevator shafts from the remainder of the building. Stair-wells should be equipped with self-closing doors, and, in exceptionally high buildings, should be closed off into sections of not over 10 floors each. Plaster cracks should be filled. Elevator enclosures should be tight and solid doors should be used.

If the sealing of the vertical openings is made effective, no allowance

[5]A.S.H.V.E RESEARCH REPORTS No 994—Wind Velocities Near a Building and Their Effect on Heat Loss, by F C. Houghten, J. L. Blackshaw and Carl Gutberlet (A S H V E TRANSACTIONS, Vol 40, 1934, p. 387). No. 1069—Heating Requirements of an Office Building as Influenced by the Stack Effect, by F. C. Houghten and Carl Gutberlet (A.S.H V.E. TRANSACTIONS, Vol. 43, 1937, p 437) Flue Action in High Buildings, by H. L. Alt (A S H.V E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, May, 1932, p. 376). Influence of Stack Effect on the Heat Loss in Tall Buildings, by Axel Marin (A.S.H.V.E TRANSACTIONS, Vol. 40, 1934, p. 377).

[6]Neutral Zone in Ventilation, by J. E. Emswiler (A.S H.V.E. TRANSACTIONS, Vol. 32, 1926, p 59)

need be made for the chimney effect. Instead, the greater wind movement at the greater heights makes it advisable to install additional heating surface on the upper floors above the level of neighboring buildings, this additional surface being increased as the height is increased. One arbitrary rule is to increase the heating surface on floors above neighboring buildings by an amount ranging from 5 per cent to 20 per cent. This extra heating surface is required only on the windward side and on windy days, and hence automatic temperature control is especially desirable with such installations.

In stair-wells that are open through many floor levels although closed off from the remainder of each floor by doors and partitions, the stratification of air makes it advisable to increase the amount of heating surface at the lower levels and to decrease the amount at higher levels. One rule is to calculate the heating surface of the entire stair-well in the usual way and to place 50 per cent of this in the bottom third, the normal amount in the middle third and the balance in the top third.

Infiltration and Air for Combustion

Infiltration in residences normally supplies the air required for combustion by fuel burning appliances, but in some residences weather-stripping, sealing and caulking may reduce infiltration to the point that special openings must be provided to supply adequate air to the heating appliances.

CHAPTER 6

Heating Load

General Procedure, Outside Temperatures, Inside Temperatures, Attic Temperatures, Temperatures in Unheated Spaces, Ground Temperatures, Basement Temperatures and Heat Loss, Transmission Heat Loss, Heat Loss Through Ceilings and Roofs, Infiltration Loss, Selection of Wind Velocities, Auxiliary Heat Sources, Intermittently Heated Buildings, Residence Heat Loss Problems

IN the design of a heating system, an estimate must be made of the maximum probable heat loss of each room or space to be heated, based on maintaining a specified inside air temperature during periods of minimum selected design weather conditions. The heat losses may be divided into two groups, namely (1) the transmission losses or heat losses through the confining walls, floor, ceiling, glass or other surfaces and (2) the infiltration losses or heat losses due to air leakage through cracks and crevices, around doors and windows, opening of doors and other sources of interchange of air between the inside and outside.

GENERAL PROCEDURE

The general procedure for calculating heat losses of a structure is as follows:

1. Select the outside design temperature. The data on climatic conditions given in Table 1 and the Design Temperature Zone Map, Fig. 1, will be useful but should be applied with judgment as suggested in the section Outside Design Temperatures

2. Select the inside air temperature, at the 60-in. breathing line or the 30-in. line, which is to be maintained in the building during the coldest weather. (See Table 2).

3. Estimate temperatures in adjacent unheated spaces and the attic. The attic temperature need not be estimated if the combined roof and ceiling coefficient is used

4. Select or compute the heat transmission coefficients for outside walls and glass; also for inside walls, floors, or top-floor ceilings, if these are next to unheated space; include roof if next to heated space. (See Chapter 4).

5. Measure amount of net outside wall, glass and roof next to heated spaces, as well as any cold walls, floors or ceilings next to unheated space. Such measurements are made from building plans, or from the actual building, using inside dimensions.

6. Compute the heat transmission losses for each kind of wall, glass, floor, ceiling and roof in the building by multiplying the heat transmission coefficient in each case by the area of the surface in square feet and the temperature difference between the inside and outside air. (See Items 1, 2, and 3).

7. Select unit values and compute the heat equivalent of the infiltration of cold air taking place around outside doors and windows. These unit values depend on the kind or width of crack and wind velocity, and when multiplied by the length of crack and the temperature difference between the inside and outside air, the result expresses the heat required to warm up the cold air leaking into the building per hour. (See Chapter 5).

8. The sum of the heat losses by transmission (Item 6) through the outside wall and glass, as well as through any cold floors, ceilings or roof, plus the heat equivalent (Item 7) of the cold air entering by infiltration represents the total heat loss equivalent for any building.

OUTSIDE DESIGN TEMPERATURES

There are no hard and fast rules for selecting the outside design temperature to be used for a given locality or type of building or heating system, and the problem is to some extent a matter of judgment and experience. The outside design temperature is seldom taken as the lowest temperature, or even the lowest daily mean temperature ever recorded in a given locality. Such temperatures are rarely repeated in successive years. A temperature somewhat higher than the minimum or the lowest

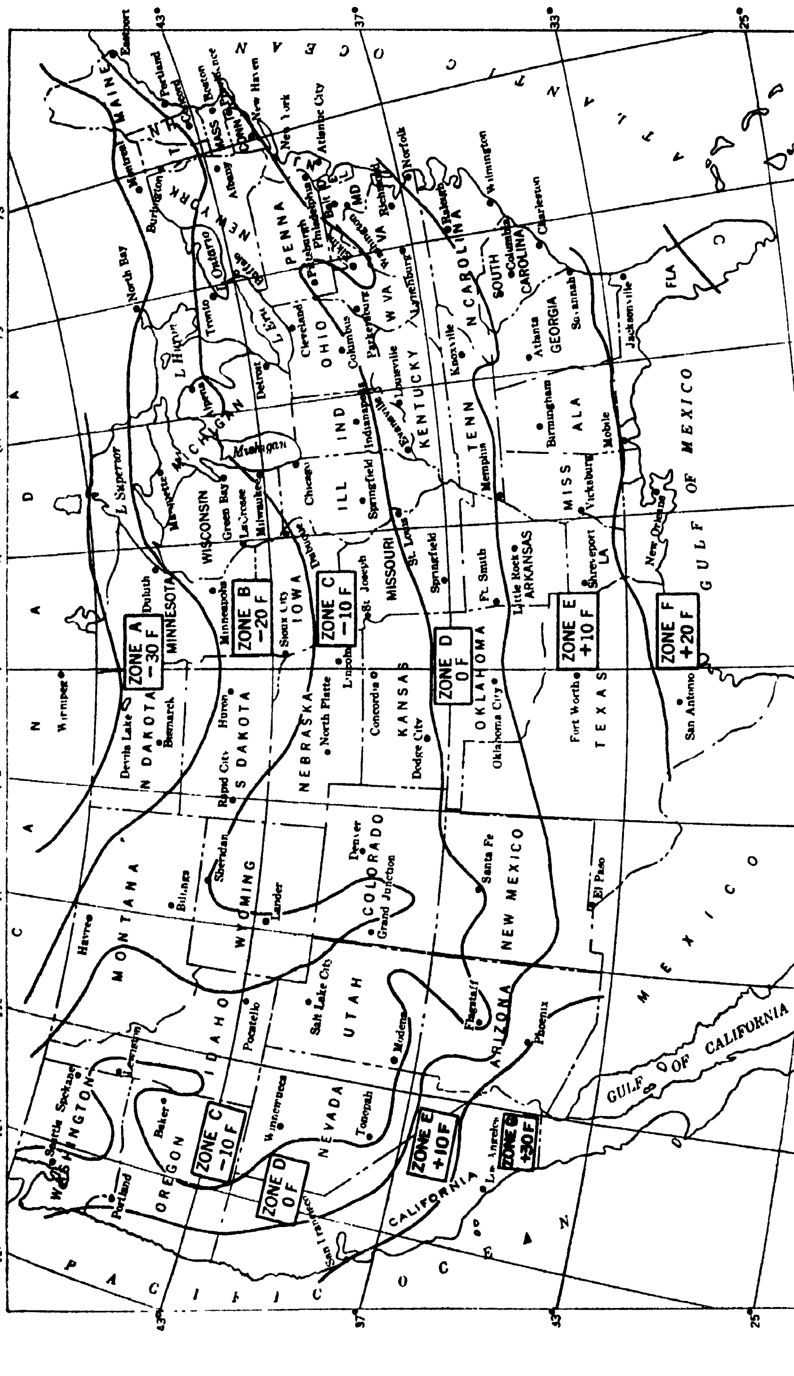


FIG. 1. DESIGN TEMPERATURE ZONE MAP

Table 1. Climatic Conditions Compiled from Weather Bureau Records[a]

Col A State	Col B City	Col C Normal Temperature, Oct 1-May 1	Col D Lowest Temperature Ever Reported	Col E Average Annual Minimum Temperature[b]	Col F Average Wind Velocity Dec., Jan., Feb., Miles per Hour	Col G Direction of Prevailing Wind, Dec., Jan., Feb.
Alabama	Birmingham	53.8	−10	12	8.0	N
	Mobile	58.9	−1	22	10.2	N
Arizona	Flagstaff	35.8	−30	−15	7.8	SW
	Phoenix	59.5	16	26	5.3	E
Arkansas	Fort Smith	50.4	−15	6	8.1	E
	Little Rock	51.6	−12	10	8.3	NW
California	Los Angeles	58.5	28	37	6.3	NE
	San Francisco	54.2	27	37	7.5	N
Colorado	Denver	38.9	−29	−11	7.5	S
	Grand Junction	38.9	−21	−2	4.3	NW
Connecticut	New Haven	38.4	−15	−1	9.5	N
District of Col.	Washington	43.4	−15	−1	7.9	NW
Florida	Jacksonville	62.0	10	29	9.1	NE
Georgia	Atlanta	51.5	−8	12	11.4	NW
	Savannah	58.5	8	22	9.5	NW
Idaho	Lewiston	42.3	−23	1	5.3	E
	Pocatello	35.7	−28	−12	9.3	SE
Illinois	Chicago	36.4	−23	−8	12.5	W
	Springfield	39.8	−21	−7	12.1	NW
Indiana	Evansville	45.1	−16	1	9.8	S
	Indianapolis	40.3	−25	−6	11.5	S
Iowa	Dubuque	33.9	−32	−17	7.1	NW
	Sioux City	32.6	−35	−20	11.6	NW
Kansas	Concordia	39.8	−25	−13	7.7	N
	Dodge City	41.4	−26	−10	10.3	NW
Kentucky	Louisville	45.3	−20	−5	9.9	SW
Louisiana	New Orleans	61.6	7	26	8.7	NE
	Shreveport	56.2	−5	16	8.1	SE
Maine	Eastport	31.5	−23	−15	12.6	NW
	Portland	33.8	−21	−6	9.2	NW
Maryland	Baltimore	43.8	−7	8	8.1	SW
Massachusetts	Boston	38.1	−18	−3	11.2	W
Michigan	Alpena	29.6	−28	−12	11.0	W
	Detroit	35.8	−24	−11	12.7	SW
	Marquette	28.3	−27	−13	10.8	NW
Minnesota	Duluth	24.3	−41	−28	13.5	SW
	Minneapolis	29.4	−34	−23	11.3	NW
Mississippi	Vicksburg	56.8	−1	18	8.3	SE
Missouri	St. Joseph	39.5	−24	−12	9.3	W
	St. Louis	43.6	−22	−2	11.7	S
	Springfield	44.3	−29	−5	10.9	SE
Montana	Billings	34.3	−49	−30	11.9	SW
	Havre	27.6	−57	−36	9.5	SW
Nebraska	Lincoln	37.0	−29	−13	10.7	S
	North Platte	35.4	−35	−17	8.3	W
Nevada	Tonopah	39.7	−15	−2	10.0	SE
	Winnemucca	37.9	−36	−10	8.2	NE
New Hampshire	Concord	33.3	−35	−15	6.6	NW
New Jersey	Atlantic City	41.6	−9	6	15.6	NW
New York	Albany	35.2	−24	−11	8.4	S
	Buffalo	34.8	−20	−4	17.2	W
	New York	40.7	−14	−3	16.7	NW
New Mexico	Santa Fe	38.3	−13	0	7.8	NE
North Carolina	Raleigh	50.0	−2	13	7.9	SW
	Wilmington	54.3	5	18	8.1	W

[a] United States data from U. S. Weather Bureau, and Canadian data from Meteorological Service of Canada, corrected to 1943.

[b] Obtained by averaging the lowest temperatures (one for each year) recorded for the period of years the local Weather Bureau has been in operation.

TABLE 1. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS[a]—(Concluded)

COL A	COL B	COL C	COL D	COL E	COL F	COL G
State or Province	City	Normal Temperature, Oct 1-May 1	Lowest Temperature Ever Reported	Average Annual Minimum Temperature[b]	Average Wind Velocity Dec, Jan, Feb, Miles per Hour	Direction of Prevailing Wind, Dec., Jan, Feb
North Dakota	Bismarck	24.6	−45	−31	9.0	NW
	Devils Lake	20.3	−46	−33	10.4	W
Ohio	Cleveland	37.2	−17	−2	15.0	SW
	Columbus	39.9	−20	−3	11.6	SW
Oklahoma	Oklahoma City	47.9	−17	2	11.8	NW
Oregon	Baker	35.2	−25	−17	6.8	SE
	Portland	46.1	−2	18	7.4	S
Pennsylvania	Philadelphia	42.7	−11	6	11.0	NW
	Pittsburgh	41.0	−20	−2	11.7	W
Rhode Island	Providence	37.2	−17	1	12.6	NW
South Carolina	Charleston	57.4	7	22	10.6	SW
	Columbia	54.0	−2	19	8.1	NE
South Dakota	Huron	28.2	−43	−26	10.5	NW
	Rapid City	33.4	−34	−21	7.9	W
Tennessee	Knoxville	47.9	−16	2	7.1	SW
	Memphis	51.1	−9	9	9.4	S
Texas	El Paso	53.5	−5	16	9.0	NW
	Ft Worth	55.2	−8	12	10.3	NW
	San Antonio	60.6	4	21	8.3	NE
Utah	Modena	36.3	−32	−15	9.3	W
	Salt Lake City	40.0	−20	2	6.7	SE
Vermont	Burlington	31.5	−29	−17	11.7	S
Virginia	Lynchburg	46.8	−7	8	8.1	NW
	Norfolk	49.3	2	15	12.4	N
	Richmond	47.0	−3	10	8.0	SW
Washington	Seattle	46.3	3	20	10.1	SE
	Spokane	37.7	−30	−5	6.3	SW
West Virginia	Elkins	39.4	−28	−8	5.8	W
	Parkersburg	42.6	−27	−1	7.3	SW
Wisconsin	Green Bay	30.0	−36	−18	10.6	SW
	LaCrosse	31.7	−43	−21	5.6	S
	Milwaukee	33.4	−25	−12	12.0	W
Wyoming	Lander	30.0	−40	−12	3.9	SW
	Sheridan	31.0	−41	−26	5.1	NW
Alta	Edmonton	22.8	−57	−41.3	7.5	SW
B C	Vancouver	42.6	2	13.1	4.5	E
	Victoria	44.0	−2	19.4	12.6	N
Man	Winnipeg	17.2	−54	−37.7	10.1	NW
N B	Fredericton	27.5	−35	−25.0	9.1	NW
N S	Yarmouth	34.8	−12	−0.1	14.3	NW
Ont	London	32.6	−27	−13.7	10.3	W
	Ottawa	26.4	−35	−24.1	8.4	W
	Port Arthur	22.0	−40	−29.5	8.0	NW
	Toronto	32.6	−26	−11.2	13.6	SW
P E I	Charlottetown	29.4	−27	−13.2	9.8	NW
Que	Montreal	28.1	−29	−18.4	11.3	SW
	Quebec	24.5	−34	−23.2	13.3	SW
Sask.	Prince Albert	16.0	−70	−47.3	5.1	W
Yukon	Dawson	1.9	−68	−54.3	3.7	S

[a]United States data from U S Weather Bureau, and Canadian data from Meteorological Service of Canada, corrected to 1943

[b]Obtained by averaging the lowest temperatures (one for each year) recorded for the period of years the local Weather Bureau has been in operation

daily mean on record may properly be assumed in making the heat loss computations. Column D Table 1 lists the lowest dry-bulb temperatures ever reported in the places listed. Column E, Table 1 shows the average of the lowest annual dry-bulb temperatures for the same localities.

Temperatures for other cities may be obtained from local weather bureau records[1]. The design temperatures shown in Fig. 1 are generally representative of the practice in various sections of the United States, although in some instances, due to local conditions of altitude or exposure, the design temperatures may vary somewhat from those indicated.

The A.S.H.V.E. Technical Advisory Committee on Weather Design Conditions has recommended the adoption for heating of an outside design temperature which is equalled or exceeded during 97½ per cent of the hours in December, January, February, and March, but the work of compiling the temperatures for the various localities is not yet completed.

INSIDE TEMPERATURES

The inside air temperature which must be maintained within a building is understood to be the dry-bulb temperature at the breathing line, 5 ft

TABLE 2. WINTER INSIDE DRY-BULB TEMPERATURES USUALLY SPECIFIED[a]

Type of Building	Deg F	Type of Building	Deg F
Schools—		Theaters—	
Class rooms	70–72	Seating space	68–72
Assembly rooms	68–72	Lounge rooms	68–72
Gymnasiums	55–65	Toilets	68
Toilets and baths	70		
Wardrobe and locker rooms	65–68	Hotels—	
Kitchens	66	Bedrooms and baths	70
Dining and lunch rooms	65–70	Dining rooms	70
Playrooms	60–65	Kitchens and laundries	66
Natatoriums	75	Ballrooms	65–68
		Toilets and service rooms	68
Hospitals—			
Private rooms	70–72	Homes	70–72
Private rooms (surgical)	70–80	Stores	65–68
Operating rooms	70–95	Public buildings	68–72
Wards	68	Warm air baths	120
Kitchens and laundries	66	Steam baths	110
Toilets	68	Factories and machine shops	60–65
Bathrooms	70–80	Foundries and boiler shops	50–60
		Paint shops	80

[a]The most comfortable dry-bulb temperature to be maintained depends on the relative humidity and air motion. These three factors considered together constitute what is termed the *effective temperature*. (See Chapter 2.)

above the floor, or the 30-in. line, and not less than 3 ft from the outside walls. Inside air temperatures, usually specified, vary in accordance with the use to which the building is to be put and Table 2 presents values which conform to good practice.

The proper dry-bulb temperature to be maintained depends upon the relative humidity and air motion, as explained in Chapter 2. In other words, a person may feel warm or cool at the same dry-bulb temperature, depending on the relative humidity and air motion. The optimum winter *effective temperature* for sedentary persons, as determined at the A.S.H.V.E. Research Laboratory, is 66 deg.

As explained in Chapter 2 for so-called still air conditions, a relative humidity of approximately 50 per cent is required to produce an effective

[1]Carnegie Institute of Technology, Bulletin, An Analysis of Winter Temperatures for One Hundred and Twenty Cities, by Clark M. Humphreys.

temperature of 66 deg when the dry-bulb temperature is 70 F. However, even where provision is made for artificial humidification, the relative humidity is seldom maintained higher than 40 per cent during the extremely cold weather, and where no provision is made for humidification, the relative humidity may be 20 per cent or less. Consequently, in using the figures listed in Table 2, consideration should be given to whether provision is to be made for humidification, and if so, the actual relative humidity to be maintained.

Temperature at Proper Level: In making the actual heat loss computations, however, for the various rooms in a building it is often necessary to modify the temperatures given in Table 2 so that the air temperature at the proper level will be used. By *air temperature at the proper level* is meant, in the case of walls, the air temperature at the mean height between floor and ceiling; in the case of glass, the air temperature at the mean height of the glass; in the case of roof or ceiling, the air temperature at the mean height of the roof or ceiling above the floor of the heated room; and in the case of floors, the air temperature at the floor level.

Temperature at Ceiling: The air temperature at the ceiling is generally higher than at the breathing level due to stratification of air resulting from the tendency of the warmer or less dense air to rise. An allowance for this fact should be made in calculating ceiling heat losses, particularly in the case of high ceilings. However, the exact allowance to be made may be somewhat difficult to determine as it depends on many factors, including (1) the type of heating system, (2) ceiling height, and (3) the inside-outside temperature differential. The type of heating system is particularly important as the temperature gradient from floor to breathing-level to ceiling may depend to a large extent on whether direct radiation, unit heaters or warm air is used, and in the latter case, whether the circulation is by gravity, auxiliary fan or forced air. Although with properly adjusted air flow the temperature differential with unit heaters can be reduced to a minimum, it is also possible with improper adjustment that the temperature differential will be increased over that which would normally result without mechanical circulation of the air.

It would be difficult from present available information to establish rules for determining the temperature difference to use in all cases. However, for residences and other structures having ceiling heights under 10 ft, the comparatively small temperature differential between the breathing level and ceiling may generally be neglected without serious error. For higher ceilings where specific test data are not available, an allowance of approximately 1 per cent per foot of height above the breathing level may be made for ceiling heights up to 15 ft and approximately 1/10 of 1 deg per foot of height above this level. The values in Table 3 are calculated on this basis. For direct radiation and gravity warm air systems, the allowance should be increased from 50 per cent to 100 per cent over those given in Table 3. These rules should, however, be used with considerable discretion.

Temperature at Floor Level: According to the University of Illinois Research Residence tests[2], the temperature at the floor level ranged from about 2½ to 6 deg below that at the breathing level, or somewhat greater than the difference between the breathing level and ceiling temperatures. Tests at the University of Wisconsin[3] indicated a some-

[2]*University of Illinois Engineering Experiment Station Bulletin* No. 318—Investigation of Oil-Fired Forced Air Furnace Systems in the Research Residence, by A. P. Kratz and S. Konzo.

[3]A.S.H.V.E. RESEARCH REPORT No. 1011—Tests of Three Heating Systems in an Industrial Type of Building, by G. L. Larson, D. W. Nelson, and John James (A S H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 185).

TABLE 3. APPROXIMATE TEMPERATURE DIFFERENTIALS BETWEEN BREATHING LEVEL AND CEILING, APPLICABLE TO CERTAIN TYPES OF HEATING SYSTEMS[a]

CEILING HEIGHT (FT)	BREATHING LEVEL TEMPERATURE (5 FT ABOVE FLOOR)									
	60	65	70	72	74	76	78	80	85	90
10	3.0	3.3	3.5	3.6	3.7	3.8	3.9	4.0	4.3	4.5
11	3.6	3.9	4.2	4.3	4.4	4.6	4.7	4.8	5.1	5.4
12	4.2	4.6	4.9	5.0	5.2	5.3	5.5	5.6	6.0	6.3
13	4.8	5.2	5.6	5.8	5.9	6.1	6.2	6.4	6.8	7.2
14	5.4	5.9	6.3	6.5	6.7	6.8	7.0	7.2	7.7	8.1
15	6.0	6.5	7.0	7.2	7.4	7.6	7.8	8.0	8.5	9.0
16	6.1	6.6	7.1	7.3	7.5	7.7	7.9	8.1	8.6	9.1
17	6.2	6.7	7.2	7.4	7.6	7.8	8.0	8.2	8.7	9.2
18	6.3	6.8	7.3	7.5	7.7	7.9	8.1	8.3	8.8	9.3
19	6.4	6.9	7.4	7.6	7.8	8.0	8.2	8.4	8.9	9.4
20	6.5	7.0	7.5	7.7	7.9	8.1	8.3	8.5	9.0	9.5
25	7.0	7.5	8.0	8.2	8.4	8.6	8.8	9.0	9.5	10.0
30	7.5	8.0	8.5	8.7	8.9	9.1	9.3	9.5	10.0	10.5
35	8.0	8.5	9.0	9 2	9.4	9.6	9.8	10.0	10.5	11.0
40	8.5	9.0	9.5	9.7	9.9	10.1	10.3	10.5	11.0	11.5
45	9 0	9.5	10.0	10.2	10.4	10.6	10.8	11.0	11.5	12.0
50	9.5	10.0	10.5	10.7	10.9	11.1	11.3	11.5	12.0	12.5

[a]The figures in this table are based on an increase of 1 per cent per foot of height above the breathing level (5 ft) up to 15 ft and 1/10 of one degree for each foot above 15 ft This table is generally applicable to forced air types of heating systems. For direct radiation or gravity warm air, increase values 50 per cent to 100 per cent

what smaller differential between the floor and breathing level temperatures. As a general rule, if the breathing level to ceiling temperature differential is neglected (as with ceiling heights under 10 ft), the breathing level-floor differential may also be neglected as the two are somewhat compensating, especially where both floor and ceiling heat losses are calculated for the same space. In other cases, the 10 ft temperature differentials in Table 3 may be used in arriving at the floor heat loss, these differentials to be subtracted from the breathing level temperature.

ATTIC TEMPERATURES

Frequently it is necessary to estimate the attic temperature, and in such cases Equation 1 can be used for this purpose.

$$t_a = \frac{A_c U_c t_i + t_o (A_r U_r + A_w U_w + A_g U_g)}{A_r U_r + A_w U_w + A_g U_g + A_c U_c} \qquad (1)$$

where

t_a = attic temperature, degrees Fahrenheit.

t_i = inside temperature near top floor ceiling, degrees Fahrenheit.

t_o = outside temperature, degrees Fahrenheit.

A_c = area of ceiling, square feet.

A_r = area of roof, square feet.

A_w = area of net vertical wall surface, square feet.

A_g = area of glass, square feet.

U_c = coefficient of transmission of ceiling, based on surface coefficient of 2.20 (upper surface, see Chapter 4).

U_r = coefficient of transmission of roof, based on surface coefficient of 2.20 (lower surface, see Chapter 4).

U_w = coefficient of transmission of vertical wall surface.

U_g = coefficient of transmission of glass.

Example 1. Calculate the temperature in an unheated attic, assuming the following conditions: $t_i = 70$; $t_o = 10$; $A_c = 1000$; $A_r = 1200$; $A_w = 100$; $A_g = 10$; $U_r = 0.50$; $U_c = 0.40$; $U_w = 0.30$; $U_g = 1.13$.

Solution: Substituting these values in Equation 1:

$$t_a = \frac{(1000 \times 0.40 \times 70) + 10\,[(1200 \times 0.50) + (100 \times 0.30) + (10 \times 1.13)]}{(1200 \times 0.50) + (100 \times 0.30) + (10 \times 1.13) + (1000 \times 0.40)}$$

$$t_a = \frac{34{,}413}{1041} = 33.1 \text{ F.}$$

Equation 1 neglects the effect of any interchange of air such as would take place through attic vents or louvers intended to preclude attic condensation. However, according to tests[4], such venting of attics by means of louvers or other small openings does not appreciably reduce the attic temperature and may be neglected without serious error. The attic temperature may be calculated in the usual manner by means of Equation 1, allowing the full value of the roof. The error resulting from this assumption will generally be considerably less than if the roof were neglected (as is sometimes the practice) and the attic temperature assumed to be the same as the outside temperature.

TEMPERATURES IN UNHEATED SPACES

The heat loss from heated rooms into unheated rooms or spaces must be based on the estimated or assumed temperature in such unheated spaces. This temperature will generally range between the inside and outside temperatures, depending on the relative areas of the surfaces adjacent to the heated room and exposed to the outside. If the respective surface areas adjacent to the heated room and exposed to the outside are approximately the same, and if the coefficients of transmission are approximately equal, the temperature in the unheated space may be assumed to be the mean of the inside and outside design temperatures. If, however, the surface areas and coefficients are unequal, the temperature in the unheated space should be estimated by means of Equation 2.

$$t_u = \frac{t(A_1U_1 + A_2U_2 + A_3U_3 + \text{etc.}) + t_o\,(A_aU_a + A_bU_b + A_cU_c + \text{etc.})}{A_1U_1 + A_2U_2 + A_3U_3 + \text{etc.} + A_aU_a + A_bU_b + A_cU_c + \text{etc}} \quad (2)$$

where

t_u = temperature in unheated space, degrees Fahrenheit.
t = inside design temperature of heated room, degrees Fahrenheit.
t_o = outside design temperature, degrees Fahrenheit.
A_1, A_2, A_3, etc. = areas of surface of unheated space adjacent to heated space, square feet.
A_a, A_b, A_c, etc. = areas of surface of unheated space exposed to outside, square feet.
U_1, U_2, U_3, etc. = coefficients of transmission of surfaces of A_1, A_2, A_3, etc.
U_a, U_b, U_c, etc. = coefficients of transmission of surfaces A_a, A_b, A_c, etc.

Example 2. Calculate the temperature in an unheated space adjacent to a heated room having surface areas (A_1, A_2, and A_3) in contact therewith of 100, 120 and 140 sq ft and coefficients (U_1, U_2, and U_3) of 0 15, 0.20 and 0.25 respectively. The surface areas of the unheated space exposed to the outside (A_a and A_b) are respectively 100 and 140 sq ft and the corresponding coefficients are 0.10 and 0.30. The sixth surface is on the ground and is neglected in this example. Assume $t = 70$ and $t_o = -10$.

[4]Methods of Moisture Control and Their Application to Building Construction, by F. B. Rowley, A B. Algren and C. E. Lund (University of Minnesota, *Engineering Experiment Station Bulletin*, No. 17)

Solution. Substituting in Equation 2:

$$t_u = \frac{70[(100\times0.15)+(120\times0.20)+(140\times0.25)]+-10[(100\times0.10)+(140\times0.30)]}{(100\times0.15)+(120\times0.20)+(140\times0.25)+(100\times0.10)+(140\times0\ 30)}$$

$$t_u = \frac{4660}{126} = 37\text{ F.}$$

The temperature in unheated spaces having large glass areas and with two or more surfaces exposed to the outside (such as sleeping porches and sun parlors), are generally assumed to be the same as the outside temperature.

GROUND TEMPERATURES

Ground temperatures to be assumed for estimating basement heat losses will usually differ in the case of basement walls and floors, the temperatures under the floors being generally higher than those adjacent to walls.

Temperatures Adjacent to Basement Walls

Ground temperatures near the surface and under open spaces vary with the climate, the season of the year and the depth below the surface. The nearer the surface (during the cold weather) the lower the temperature. Frost will penetrate to a depth of over 4 ft in some localities if not protected by snow. A thick blanket of snow will result in a higher ground temperature near the surface. Consequently ground temperatures near the surface may be higher in cold climates where the snow remains on the ground for a greater length of time than in more moderate climates where the snow melts away periodically during the winter.

Complete data for various localities are not as yet available but in estimating heat losses through vertical walls below grade, it is advisable not to assume average ground temperatures above 32 F in northern climates when estimating heat losses from heated basements. This is for the mean height of the basement wall. Since the recommended wall coefficient for basement walls in contact with the soil is only 0.10, any small variation in the assumed ground temperature will not materially affect the calculated heat loss.

Temperatures Under Basement Floors

The temperature under basement floors[5] is influenced by the heat from the basement or protected from the influence of atmospheric conditions by the basement. In computing losses through basement floors the ground temperatures may be assumed the same as the approximate water temperature at depths of 30 to 60 ft given in Fig. 3, Chapter 26.

BASEMENT TEMPERATURES AND HEAT LOSS

The allowance to be made for basement heat loss depends on whether the basement is to be heated or not.

If a basement is completely below grade and is *not heated*, the temperature in the basement will normally range between that in the rooms above and the ground temperature. Basement windows will of course

[5]A.S.H.V.E. RESEARCH REPORT No. 1213—Heat Loss Through Basement Walls and Floors, by F. C Houghten, S. I. Taimuty, Carl Gutberlet and C. J. Brown (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 369).

lower the basement temperature when it is colder outside and any heat given off by the heating plant will increase the basement temperature. In any case, the exact basement temperature is likely to be a somewhat indeterminate quantity, if the basement is not heated. Since the basement temperature will generally be lower than that of the rooms above, an allowance should theoretically be made for the loss from the rooms above through the floor over the basement.

If the basement is *heated* and a specified temperature is to be maintained, the heat loss should be estimated in the usual manner, based on the proper wall and floor coefficients (see Chapter 4) and the outside air and the ground temperatures. Heat loss through windows and walls above grade should be based on outside temperatures and the proper air-to-air coefficients. Heat loss through basement walls below grade should be based on the floor and wall coefficients for surfaces in contact with the soil and on the proper ground temperature.

TRANSMISSION HEAT LOSS

The basic formula for the loss of heat by transmission through any surface is given in Equation 3.

$$H_t = A\,U\,(t - t_o) \tag{3}$$

where

H_t = heat loss transmitted through the wall, roof, ceiling, floor or glass, Btu per hour.

A = area of wall, glass, roof, ceiling, floor or other exposed surfaces, square feet.

U = coefficient of transmission, air to air, Btu per hour per square foot per degree Fahrenheit temperature difference (Chapter 4).

t = inside temperature near surface involved which may not necessarily be the so-called breathing line temperature, degrees Fahrenheit.

t_o = outside temperature, or temperature of adjacent unheated space or of the ground, degrees Fahrenheit.

Example 3 Calculate the transmission loss through an 8 in. brick wall having an area of 150 sq ft if the inside temperature (t) is 70 F and the outside temperature (t_o) is −10 F.

Solution. The coefficient of transmission (U) of a plain 8 in. brick wall is 0.50 (Chapter 4, Table 6). The area (A) is 150 sq ft. Substituting in Equation 3:

$$H_t = 150 \times 0.50 \times [70 - (-10)] = 6000 \text{ Btu per hour.}$$

Transmission Loss Through Ceilings and Roofs

The transmission heat loss through top floor ceilings, attics and roofs may be estimated by either of two methods:

1. By substituting in Equation 3 the ceiling area (A), the inside-outside temperature difference ($t - t_o$) and the proper value of (U):

   a. *Flat roofs.* Select the coefficient of transmission of the ceiling and roof from Table 14, Chapter 4.

   b. *Pitched roofs.* Select the combined roof and ceiling coefficient from Table 16, Chapter 4 or calculate the combined roof and ceiling coefficient by means of Equation 4, Chapter 4, where this formula is applicable as explained in Chapter 4.

2. By estimating the attic temperature (based on the inside and outside design temperature) by means of Equation 1, and substituting for t_o in Equation 3, the value of t_a thus obtained, together with the ceiling area (A) and the ceiling coefficient (U). This applies to *pitched roofs*. In the case of *flat roofs* it is not necessary to calculate the attic temperatures as the ceiling-roof heat loss can be determined as per paragraph 1a.

INFILTRATION HEAT LOSS

The infiltration heat loss includes (1) the *sensible heat loss* or the heat required to warm the outside air entering by infiltration and (2) the *latent heat loss* or the heat equivalent of any moisture which must be added.

Sensible Heat Loss

The formula for the heat required to warm the outside air which enters a room by infiltration, to the temperature of the room, is given in Equation 4.

$$H_s = 0.24\ Qd\ (t - t_o) \tag{4}$$

where

H_s = heat required to raise temperature of air leaking into building from t_o to t, Btu per hour.

0.24 = specific heat of air.

Q = volume of outside air entering building, cubic feet per hour (see Chapter 5).

d = density of air at temperature t_o, pounds per cubic foot.

It is sufficiently accurate to use $d = 0.075$ in which case Equation 4 reduces to

$$H_s = 0.018\ Q\ (t - t_o) \tag{4a}$$

The volume of outside air entering per hour (Q) depends on the wind velocity and direction, the width of crack or size of openings, the type of openings and other factors, as explained in Chapter 5. Where the crack method is used for estimating the amount of air leakage, it is more convenient to express the heat loss due to air leakage in terms of the crack length, as follows:

$$H_s = 0.018\ Q\ L\ (t - t_o) = B\ L\ (t - t_o) \tag{4b}$$

where

B = air leakage per foot of crack (Chapter 5) for the wind velocity and type of windows or door crack involved multiplied by 0.018.

L = length of window or door crack to be taken into consideration, feet.

Example 4. What is the infiltration heat loss per hour through the crack of a 3 x 5 ft double-hung wood window, based on an average non-weatherstripped window and a wind velocity of 15 mph? Assume inside and outside temperatures to be 70 F and zero respectively.

Solution. According to Table 2, Chapter 5, the air leakage through a window of this type (based on 1/16 in. crack and 3/64 in clearance) is 39 cu ft per foot of crack per hour. Therefore, $B = 39 \times 0.018 = 0.70$ The length of crack (L) is $(2 \times 5) + (3 \times 3)$, or 19 ft; $t = 70$ and $t_o = 0$ Substituting in Equation 4b,

$$H_s = 0.70 \times 19 \times (70 - 0) = 931 \text{ Btu per hour.}$$

Crack Length to be Used for Computations

The amount of crack used for computing the infiltration heat loss should not be less than half of the total crack in the outside walls of the room. For a building having no partitions, whatever wind enters through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, take one-half the total crack for computing each side and end of the building. In a room with one exposed wall, take all the crack; with two exposed walls, take the wall having the most crack; and with three or four exposed walls, take the wall having the most crack; but in no case take less than half the total crack.

The total infiltration loss of a building having partitions will not be

equal to the sum of the infiltration losses of the various rooms since at any given time infiltration will take place only on the windward side or sides and not on the leeward side. Therefore, if a building has more than one room which is divided by interior walls or partitions, it is sufficiently accurate to use half of the total infiltration losses for determining the total heat requirements.

Latent Heat Loss

When it is intended to add moisture to air leaking into a room for the maintenance of proper winter comfort conditions, it is necessary to determine the heat equivalent to evaporate the required amount of water vapor, which may be calculated by the equation:

$$H_l = Q\, d \left(\frac{m_i - m_o}{7000}\right) h_{fg} \tag{5}$$

where

H_l = heat required to increase moisture content of air leaking into building from m_o to m_i, Btu per hour

Q = volume of outside air entering building, cubic feet per hour.

d = density of air at temperature t_i, pounds per cubic foot.

m_i = vapor density of inside air, grains per pound of dry air.

m_o = vapor density of outside air, grains per pound of dry air.

h_{fg} = latent heat of vapor at m_i, Btu per pound.

If the latent heat of vapor (h_{fg}) is assumed to be 1060 Btu per pound, Equation 5 reduces to

$$H_l = 0.0114\, Q\, (m_i - m_o) \tag{5a}$$

Equations 4a, 4b and 5a may also be used for determining the sensible and latent heat gains due to infiltration in cooling load computations.

SELECTION OF WIND VELOCITIES

The effect of wind on the heating requirements of any building should be given consideration under two heads:

1. Wind movement increases the heat transmission of walls, glass, and roof, affecting poor walls to a much greater extent than good walls.
2. Wind movement materially increases the infiltration (inleakage) of cold air through the cracks around doors and windows, and even through the building materials themselves, if such materials are at all porous.

Theoretically as a basis for design, the most unfavorable combination of temperature and wind velocity should be chosen. It is entirely possible that a building might require more heat on a windy day with a moderately low outside temperature than on a quiet day with a much lower outside temperature. However, the combination of wind and temperature which is the worst would differ with different buildings, because wind velocity has a greater effect on buildings which have relatively high infiltration losses. It would be possible to work out the heating load for a building for several different combinations of temperature and wind velocity which records show to have occurred and to select the worst combination; but designers generally do not feel that such a degree of refinement is justified.

It has been the practice for many years in estimating air leakage by the crack method to use the *average* wind velocity during the months of December, January and February. This *average* wind velocity may not necessarily correspond with that occurring during periods when the

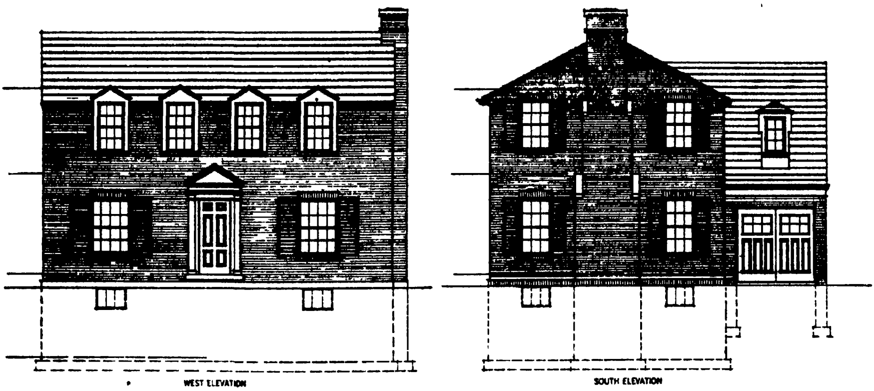
outside design temperature prevails, the latter being not an *average* but rather a near extreme, that is, a specified number of degrees above the lowest temperature recorded in the locality involved. Therefore instead of using the aforementioned *average* wind velocity, it is the practice of some designers to use in all cases a wind velocity of 15 mph together with the proper design temperature. Although a 15 mph wind velocity is higher than the general average wind velocity during December, January and February in various United States cities, this and higher wind velocities frequently occur during periods of outside temperature corresponding to the design temperature. It should be added that this wind velocity also corresponds with that on which the heat loss coefficients in Chapter 4 are based, although the effect of variations in wind velocity on the infiltration losses is generally much greater than the effect of wind velocity on the heat loss by transmission through walls. Therefore, pending further investigation of this subject, either the average during December, January and February or a 15 mph wind velocity may be used at the discretion of the designer. Where the air change method is used for estimating infiltration losses, the wind velocity is not considered.

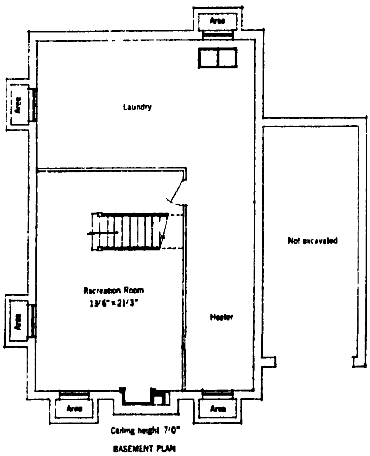
Exposure Factors: Many designers use empirical exposure factors to increase the calculated heat loss of rooms or spaces on the side or sides of the building exposed to the prevailing winds. However, according to a survey made in 1943, many Guide users have found that the use of exposure factors is not necessary as the Guide method of calculating heat losses provides an ample heat loss allowance. Therefore exposure factors may be regarded as "factors of safety" for the rooms or spaces exposed to the prevailing winds, to allow for additional capacity for these rooms or spaces, or to "balance the radiation," particularly in the case of multistory buildings. Although the exposure allowance is frequently assumed to be 15 per cent, the actual allowance to be made, if any, must to a large extent be a matter of experience and judgment of the designer, since there are at present no authentic test data available from which rules could be developed for the many conditions encountered in practice.

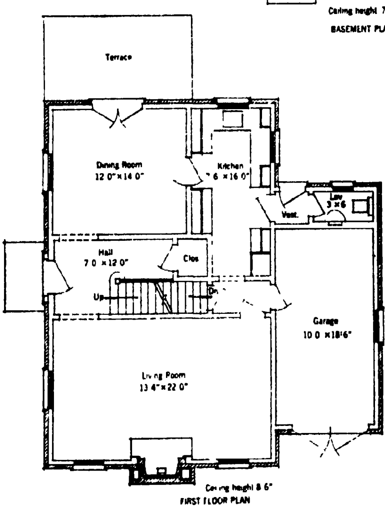
As stated previously, the value of U in the tables of Chapter 4 is based on a wind velocity of 15 mph and the surface resistance for this wind velocity (0.17) is sufficiently low that higher wind velocities will decrease the surface resistance to a negligible degree and therefore have only a slight effect on the average over-all coefficient. On the other hand, infiltration losses vary almost directly as the wind velocity as will be apparent from the factors in Table 2 of Chapter 5. The more exact method therefore would be to differentiate among the various exposures more accurately by calculating the infiltration and transmission losses separately for the different sides of the building, using different assumed wind velocities for the infiltration losses on the various sides of the building.

AUXILIARY HEAT SOURCES

The heat supplied by persons, lights, motors and machinery should always be ascertained in the case of theaters, assembly halls, and industrial plants, but allowances for such heat sources must be made only after careful consideration of all local conditions. In many cases, these heat sources should not be allowed to affect the size of the installation at all, although they may have a marked effect on the operation and control of the system. In general, it is safe to say that where audiences are involved, the heating installation must have sufficient capacity to bring the building up to the stipulated inside temperature before the audience







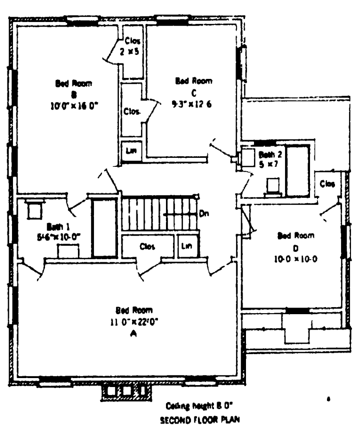


FIG. 2. ELEVATIONS AND FLOOR PLANS OF RESIDENCE

TABLE 4. HEAT LOSS CALCULATION SHEET FOR UNINSULATED RESIDENCE (FIG. 2)

A	B	C	D	E	F	G
ROOM OR SPACE	PART OF STRUCTURE	NET AREA OR CRACK LENGTH	COEFFICIENT	TEMP DIFF [a]	HEAT LOSS (Btu per hour)	TOTALS (Btu per hour)
Bedroom A	Walls	238 sq ft	0.28	80	5330	
	Glass	40 sq ft	0.45	80	1440	
	Infiltration	36 lin ft[b]	0.35[c]	80	1010	
	Ceiling[d]	242 sq ft	0.69	39.8	6660	14,440
Bedroom B and Closet	Walls	156 sq ft	0.28	80	3490	
	Glass	40 sq ft	0.45	80	1440	
	Infiltration	36 lin ft[e]	0.35	80	1010	
	Ceiling[d]	160 sq ft	0.69	39 8	4400	10,340
Bedroom C	Walls	114 sq ft	0.28	80	2560	
	Glass	27 sq ft	0.45	80	970	
	Infiltration	18 lin ft[f]	0.35	80	500	
	Ceiling[d]	120 sq ft	0.69	39.8	3300	7,330
Bedroom D and Closet	Walls	118 sq ft	0.28	80	2650	
	Glass	20 sq ft	0 45	80	720	
	Infiltration	18 lin ft	0.35	80	500	
	Ceiling[d]	120 sq ft	0.69	39.8	3300	
	Floor over Garage	110 sq ft	0.25	35[g]	960[m]	8,130
Bathroom 1	Walls	30 sq ft	0.28	80	670	
	Glass	14 sq ft	0 45	80	500	
	Infiltration	18 lin ft	0.35	80	500	
	Ceiling[d]	55 sq ft	0.69	39.8	1510	3,180
Bathroom 2	Walls	79 sq ft	0.26	80	1770	
	Glass	9 sq ft	0.45	80	320	
	Infiltration	15 lin ft	0.35	80	420	
	Ceiling[d]	35 sq ft	0.69	39.8	960	
	Floor over Garage	35 sq ft	0.25	35	310[m]	3,780
Living Room	Walls	267 sq ft	0.28	80	5980	
	Walls (adjoining garage)	94 sq ft	0.39[h]	35	1280[m]	
	Glass	50 sq ft	0.45	80	1800	
	Infiltration	40 lin ft	0.35	80	1120	10,180
Dining Room	Walls	166 sq ft	0.28	80	3720	
	Glass (doors)	35 sq ft	1.13	80	3160	
	Glass (window)	20 sq ft	0.45	80	720	
	Infiltration[i]	31 lin ft	0.35	80	870	8,470
Kitchen and Entrance to Garage	Walls (outside)	96 sq ft	0.28	80	2150	
	Walls (adjoining garage)	51 sq ft	0.39[h]	35	700[m]	
	Infiltration	27 lin ft	0.35	80	760	
	Glass	18 sq ft	0.45	80	650	
	Door to garage	17 sq ft	0.51	35	300[m]	4,560

TABLE 4. HEAT LOSS CALCULATION SHEET FOR UNINSULATED RESIDENCE (FIG. 2) (Concluded)

A	B	C	D	E	F	G
ROOM OR SPACE	PART OF STRUCTURE	NET AREA OR CRACK LENGTH	COEFFICIENT	TEMP DIFF [a]	HEAT LOSS (Btu per hour)	TOTALS (Btu per hour)
Lavette and Vestibule	Walls (outside)	82 sq ft	0.28	80	1840	
	Walls (adjoining garage)	85 sq ft	0.39[h]	35	1160[m]	
	Door	19 sq ft	0.51	80	780	
	Glass	9 sq ft	0.45	80	320	
	Infiltration	19 lin ft	0.35	80	530	4,630
Entrance Hall	Walls	39 sq ft	0.28	80	870	
	Door	21 sq ft	0.38	80	640	
	Infiltration	20 lin ft	0 35	80	560	
	Ceiling[d, p]	87 sq ft	0.69	39.8	2490	4,560
Garage	Walls	167 sq ft	0.28	45	2110	
	Glass	53 sq ft	1.13	45	2700	
	Doors	44 sq ft	0.51	45	1010	
	Infiltration	37 lin ft	1 62[j]	45	2700	
	Floor (heat gain)	185 sq ft	0.10[k]	−15[k]	−280	
	Heat gain				−4710[m]	3,530
Recreation Room[n]	Floor	287 sq ft	0 10	20	570	
	Walls	220 sq ft	0.10	38	840	
	Glass	8 sq ft	1.13	80	720	
	Infiltration	8 lin ft	0.76	80	490	2,620
Total						85,750

[a]The inside-outside temperature difference is 70 − (−10) or 80 F, except where otherwise noted

[b]Only the south windows are used for arriving at the window crack for this room, on the assumption that whatever air enters through the south window cracks will leave through the west window cracks or elsewhere

[c]Double-hung wood windows with storm sash are assumed to have the same leakage per foot of crack as weatherstripped windows The air leakage per foot of crack is about 19 5 cu ft per foot of crack for a wind velocity of 12 5 mph (See Table 2, Chapter 5) The heat equivalent of the air leakage per hour per degree temperature difference per foot of crack is obtained by multiplying this value by 0 018 or 19.5 × 0 018 = 0 35.

[d]In this problem the ceiling heat losses are calculated by estimating the attic temperature and then calculating the loss through the ceiling using the proper temperature difference This unheated attic is not ventilated during the winter months The attic temperature is estimated from Equation 4 to be 30.2 F when the outside temperature is −10 F and the room temperature is 70 F The temperature difference is therefore 70 − 30 2 or 39 8 F

[e]The window crack in the west wall having two windows is used

[f]One-half the total crack is used in these rooms

[g]Temperature in garage assumed to be 35 F

[h]Coefficient for wall adjoining garage calculated on basis of metal lath and plaster on both sides of studs ($U = 0\ 39$)

[i]The door crack is used for estimating the infiltration in this room and as the French doors are weatherstripped the infiltration coefficient is assumed to be the same as in Note b.

[j]The leakage for the garage doors is assumed to be twice that for poorly-fitted double-hung wood windows or about 90 cu ft per foot of crack for a wind velocity of 12 5 mph. The infiltration coefficient is therefore 0.018 × 90 or 1 62

[k]The ground temperature is assumed to be 50 F and, as the garage temperature is 35 F, the heat transfer will be from the ground to the garage, and this heat gain should therefore be subtracted from the heat loss

[m]The heat losses from various rooms into the garage are heat gains for the garage

[n]Heat is to be provided for the recreation room and this space is therefore figured on the basis of a 70 F temperature. Heat loss into the basement from recreation room is neglected, the calculations being based only on losses through the outside walls, glass and floor.

[p]The upstairs hall ceiling is included with the downstairs entrance hall because these are connected by means of the stairway. The heat should be provided downstairs.

arrives. In industrial plants, quite a different condition exists, and heat sources, if they are always available during the period of human occupancy, may be substituted for a portion of the heating installation. In no case should the actual heating installation (exclusive of heat sources) be reduced below that required to maintain at least 40 F in the building.

Electric Motors and Machinery

Motors and the machinery which they drive, if both are located in the room, convert all of the electrical energy supplied into heat, which is retained in the room if the product being manufactured is not removed until its temperature is the same as the room temperature.

If power is transmitted to the machinery from the outside, then only the heat equivalent of the brake horsepower supplied is used. In the first case the Btu supplied per hour $= \frac{\text{Motor horsepower}}{\text{Efficiency of motor}} \times 2546$, and in the second case Btu per hour $=$ bhp $\times$ 2546, in which 2546 is the Btu equivalent of 1 hp-hr. In some mills this is the chief source of heating and it is frequently sufficient to overheat the building even in zero weather, thus requiring cooling by ventilation the year round.

The heat (in Btu per hour) from electric lamps is obtained by multiplying the watts per lamp by the number of lamps and by 3.413. One cubic foot of producer gas gives off about 150 Btu per hour; one cubic foot of manufactured gas about 535 Btu per hour; and one cubic foot of natural gas about 1000 Btu per hour. A Welsbach burner averages 3 cu ft of gas per hour and a fish-tail burner, 5 cu ft per hour. For information concerning the heat supplied by persons, refer to data given in Chapter 2.

INTERMITTENTLY HEATED BUILDINGS

In the case of intermittently heated buildings additional heat is required for raising the temperature of the air, the building materials and the material contents of the building to the specified inside temperature. The rate at which this additional heat must be supplied depends upon the heat capacity of the structure and its material contents and upon the time in which these are to be heated[6].

This additional heat may be figured and allowed for as conditions require, but inasmuch as the heating system proportioned for taking care of the heat losses will usually have a capacity about 100 per cent greater than that required for average winter weather, and inasmuch as most buildings may either be continuously heated or have more time allowed for heating-up during the few minimum temperature days, no allowance is usually made except in the size of boilers or furnaces. For churches, auditoriums and other intermittently heated buildings, additional capacity should be provided.

RESIDENCE HEAT LOSS PROBLEMS

Example 5. Calculate the heat loss of residence shown in Fig. 2 located in the vicinity of Chicago. Assume inside and outside design temperatures to be 70 F and −10 F respectively. The attic is unheated. Assume ground temperature to be 50 F under basement and garage floors and 32 F adjoining basement walls. Estimate in-

[6]Heat Requirement Tables for Intermittently Heated Buildings, (*Engineering Experiment Station Bulletin*, No. 60, A. and M College of Texas, College Station, Texas), contains a set of tables applicable to either intermittent heating or cooling Further information may be found in a paper, A Method of Compiling Tables for Intermittent Heating, by Elmer G Smith (A S H V E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June, 1942, p 386).

TABLE 5. SUMMARY OF HEAT LOSSES OF UNINSULATED RESIDENCE

Heat losses given in Btu per hour

Room or Space	Walls	Ceiling and Roof	Floor	Glass and Door	Infiltration	Totals
Bedroom A	5330	6660	....	1440	1010	14,440
Bedroom B	3490	4400	....	1440	1010	10,340
Bedroom C	2560	3300	....	970	500	7,330
Bedroom D	2650	3300	960	720	500	8,130
Bathroom 1	670	1510	....	500	500	3,180
Bathroom 2	1770	960	310	320	420	3,780
Living Room	7260	....	....	1800	1120	10,180
Dining Room	3720	....	....	3880	870	8,470
Kitchen	2850	....	....	950	760	4,560
Lavette	3000	....	....	1100	530	4,630
Entrance Hall	870	2490	....	640	560	4,560
Garage	−1030[a]	......	−1550[b]	3410	2700	3,530
Recreation	840	....	570	720	490	2,620
Totals	33,980	22,620	290	17,890	10,970	85,750
Percentages	39.6	26 4	0 3	20.9	12.8	100 0

Wall heat loss of 2110 Btu minus wall heat gain of 3140 Btu

[b] Heat gains; 960, 310 and 280 Btu

filtration by crack method, assuming average wind velocity to be 12.5 mph during December, January and February. No wall, ceiling or roof insulation is to be figured in this problem, but all first and second floor windows are to have storm sash. The building is constructed as follows (transmission coefficients (U) in parentheses):

Walls: Brick veneer, building paper, wood sheathing, studding, metal lath and plaster (0 28). Walls of dormer over garage, same except wood siding in place of brick veneer (0 26)

Attic Walls Brick veneer, building paper, wood sheathing on studding (0 42).

Basement Walls: 10 in. concrete (0 10).

Roof. Asphalt shingles on wood sheathing on rafters (0 53).

Ceiling (Second floor): Metal lath and plaster (0 69)

Windows: Double-hung wood windows with storm sash (0 45) Steel casement sash in basement (1 13).

Floor (Bedroom D): Maple finish flooring on yellow pine sub-flooring; metal lath and plaster ceiling below (0.25).

Floor (Basement and Garage): 4 in. stone concrete on 3 in cinder concrete (0 10).

Solution: The calculations for this problem are given in Table 4, and a summary of the results in Table 5. The values in column F of Table 4 were obtained by multiplying together the figures in columns C, D and E. The heat losses are calculated to the nearest 10 Btu. See reference notes for Table 4 for further explanation of data.

Attention is called to the summary of heat losses (Table 5) of the uninsulated residence (Fig 2) As storm windows are used in this instance the *glass* and *door* transmission heat losses of 20 9 per cent are relatively small. The infiltration losses (12.8 per cent) are also comparatively small in this case because the storm windows serve substantially the same purpose as weatherstripping In this problem, the wall, ceiling and floor transmission losses comprise 66 3 per cent of the total. If the building is insulated, the relative heat loss percentages will materially change. (See Example 6 and Table 6)

Example 6. Calculate the heat loss of residence shown in Fig. 2 based on the same conditions as in Example 5 but insulated throughout as shown on the following page (coefficients in parentheses)

Walls: Brick veneer, 25⁄32 in insulation board sheathing, studding, 1 in. insulation board lath and plaster (0.13). Walls of dormer over garage same except wood siding in place of brick veneer (0.12).

Attic Walls: Brick veneer, 25⁄32 in. insulation board sheathing on studding (0.28).

Walls Adjoining Garage: Plaster on 1 in. insulation board, studding, metal lath and plaster (0.18).

Basement Walls (Recreation Room): 10 in. concrete (0.10).

Roof: Asphalt shingles on wood sheathing on rafters (0.53).

Ceiling (Second floor): 1 in. insulation board and plaster; ½ in. insulation board on top of ceiling joists (0.15).

Windows: Same as Example 5.

Floor (Bedroom D): Maple finish flooring on yellow pine sub-flooring; ½ in. insulation board and plaster ceiling below (0.18).

Solution: The procedure for calculating the heat losses is similar to that for Example 5. A summary of the results is given in Table 6.

TABLE 6. SUMMARY OF HEAT LOSSES OF INSULATED RESIDENCE

Heat losses given in Btu per hour

ROOM OR SPACE	WALLS	CEILING AND ROOF	FLOOR	GLASS AND DOOR	INFILTRATION	TOTALS
Bedroom A	2670	2370	....	1440	1010	7,490
Bedroom B	1750	1570	....	1440	1010	5,770
Bedroom C	1280	1170	....	970	500	3,920
Bedroom D	1320	1170	690	720	500	4,400
Bathroom 1	340	540	....	500	500	1,880
Bathroom 2	820	340	220	320	420	2,120
Living Room	3580	....	....	1800	1120	6,500
Dining Room	1860	....	....	3880	870	6,610
Kitchen	1400	....	....	950	760	3,110
Lavette	1460	....	....	1100	530	3,090
Entrance Hall	440	850	....	640	560	2,490
Garage	−400[a]	....	−1190[b]	3410	2700	4,520
Recreation	840	....	570	720	490	2,620
Totals	17,360	8,010	290	17,890	10,970	54,520
Percentages	31.9	14.7	0 5	32 8	20 1	100.0

[a]Wall heat loss of 1050 Btu minus wall heat gains of 590, 320 and 540 Btu

[b]Heat gains; 690, 220 and 280 Btu.

CHAPTER 7

Cooling Load

Design Outside Temperatures, Components of Heat Gain, Normal Heat Transmission, Solar Heat Transmission, Solar Radiation Through Glass, Heat Introduced by Outside Air, Heat Emission of Appliances, Moisture Through Walls

LOAD calculations for summer air conditioning are more complicated than heating load calculations because there are more factors to be considered. Due to the variable nature of some of the contributing load components and the fact that they do not necessarily impose their maximum effect simultaneously, considerable care must be used in determining their phase relationship so that equipment of proper capacity may be selected to maintain specified indoor conditions.

The conditions to be maintained in an enclosure are variable and depend upon several factors, especially the outside design conditions, duration of occupancy and relationship between air motion, dry-bulb and wet-bulb temperatures. Information concerning the proper indoor effective temperature to be maintained is given in Chapter 2, for different geographical locations and for various age groups of individuals.

Summer dry-bulb and wet-bulb temperatures of various cities are given in Table 1. The temperatures are not the maximums but the design temperatures which should be used in air conditioning calculations. The maximum outside wet-bulb temperatures as given in Weather Bureau reports usually occur only from 1 to 4 per cent of the time, and they are therefore of such short duration that it is not practical to design a cooling system for them. The temperatures shown in Table 1 are based on available design conditions known to be successfully applied.

COMPONENTS OF HEAT GAIN

A cooling load determination is composed of five components which are classified in the following manner:

1. Normal heat transfer through windows, walls, partitions, doors, floors, ceilings, etc.
2. Transfer of solar radiation through windows, walls, doors, skylights, and roof.
3. Heat emission of occupants within enclosures.
4. Heat introduced by infiltration of outside air and controlled ventilation.
5. Heat emission of mechanical, chemical, gas, steam, hot water and electrical appliances located within enclosures.

The components of heat gain, classified by source, are further classified as sensible and latent heat gain.

The first two components fall into the classification of sensible heat gain, that is, they tend to raise the temperature of the air within the structure. The last three components not only produce sensible heat gain but they may also tend to increase the moisture content of the air within the structure.

Normal Heat Transmission

By normal heat transmission, as distinguished from solar heat transmission, is meant the transmission of heat through windows, walls, partitions, etc. from without to interior of enclosure by virtue of difference

TABLE 1. DESIGN DRY- AND WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER

State	City	Design Dry-Bulb	Design Wet-Bulb	Summer Wind Velocity MPH	Prevailing Summer Wind Direction
Ala.	Birmingham	95	78	5.2	S
	Mobile	95	80	8.6	SW
Ariz.	Phoenix	105	76	6.0	W
Ark.	Little Rock	95	78	7.0	NE
Calif.	Los Angeles	90	70	6.0	SW
	San Francisco	90	65	11.0	SW
Colo.	Denver	95	64	6.8	S
Conn.	New Haven	95	75	7.3	S
Del.	Wilmington	95	78	9.7	SW
D. C.	Washington	95	78	6.2	S
Fla.	Jacksonville	95	78	8.7	SW
	Tampa	94	79	7.0	E
Ga.	Atlanta	95	76	7.3	NW
	Savannah	95	78	7.8	SW
Idaho	Boise	95	65	5.8	NW
Ill.	Chicago	95	75	10.2	NE
	Peoria	95	76	8.2	S
Ind.	Indianapolis	95	76	9.0	SW
Iowa	Des Moines	95	77	6.6	SW
Kansas	Wichita	100	75	11.0	S
Ky.	Louisville	95	76	8.0	SW
La.	New Orleans	95	79	7.0	SW
	Shreveport	100	78	6.2	S
Maine	Portland	90	73	7.3	S
Md.	Baltimore	95	78	6.9	SW
Mass.	Boston	92	75	9.2	SW
Mich.	Detroit	95	75	10.3	SW
Minn.	Minneapolis	95	75	8.4	SE
Miss.	Vicksburg	95	78	6.2	SW
Mo.	Kansas City	100	76	9.5	S
	St. Louis	95	78	9.4	SW
Mont.	Helena	95	67	7.3	SW
Nebr.	Lincoln	95	75	9.3	S
Nev.	Reno	95	65	7.4	W
N. H.	Manchester	90	73	5.6	NW
N. J.	Trenton	95	78	10.0	SW
N. Y.	Albany	92	75	7.1	S
	Buffalo	93	75	12.2	SW
	New York	95	75	12.9	SW
N. M.	Santa Fe	90	65	6.5	SE
N. C.	Asheville	90	75	5.6	SE
	Wilmington	95	79	7.8	SW
N. Dak.	Bismarck	95	73	8.8	NW
Ohio	Cincinnati	95	78	6.6	SW
	Cleveland	95	75	9.9	S
Okla.	Oklahoma City	101	76	10.1	S
Ore.	Portland	90	65	6.6	NW
Pa.	Philadelphia	95	78	9.7	SW
	Pittsburgh	95	75	9.0	NW
R. I.	Providence	93	75	10.0	NW
S. C.	Charleston	95	80	9.9	SW
	Greenville	95	76	6.8	NE
S. Dak.	Sioux Falls	95	75	7.6	S
Tenn.	Chattanooga	95	77	6.5	SW
	Memphis	95	78	7.5	SW
Texas	Dallas	100	78	9.4	S
	El Paso	100	69	6.9	E
	Galveston	95	80	9.7	S
	Houston	95	78	7.7	S
	San Antonio	100	78	7.4	SE

TABLE 1. DESIGN DRY- AND WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER (Concluded)

State	City	Design Dry-Bulb	Design Wet-Bulb	Summer Wind Velocity MPH	Prevailing Summer Wind Direction
Utah	Salt Lake City	92	63	8.2	SE
Vt.	Burlington	90	73	8.9	S
Va.	Norfolk	95	78	10.9	S
	Richmond	95	78	6.2	SW
Wash.	Seattle	85	65	7.9	S
	Spokane	90	65	6.5	SW
W. Va.	Parkersburg	95	75	5.3	SE
Wis.	Madison	95	75	8.1	SW
	Milwaukee	95	75	10 4	S
Wyo.	Cheyenne	95	65	9.2	S

between outside and inside air temperatures. This load is calculated in a manner similar to that described in Chapter 6 (except that flow of heat is reversed) by means of the formula:

$$H_t = A\,U\,(t_o - t) \tag{1}$$

where

H_t = heat transmitted through the material of wall, glass, floor, etc., Btu per hour.

A = net inside area of wall, glass, floor, etc., square feet.

t = inside temperature, degrees Fahrenheit.

t_o = outside temperature, degrees Fahrenheit.

U = coefficient of transmission of wall, glass, floor, etc., Btu per hour per square foot per degree Fahrenheit difference in temperature (Tables 3 to 17, Chapter 4).

Solar Heat Transmission

Calculations of the solar heat transmitted through walls and roofs are difficult to determine because of periodic character of heat flow and time lag due to heat capacity of construction.

The variation in radiation intensity on differently oriented surfaces is given in Fig. 1, and in Tables 2, 3, 4 and 5. The greater part of the radiation intensity is always direct radiation from the sun. However, during the time when the sun is shining, any surface receives radiation of a lower intensity coming from all parts of the sky due to reflection and refraction. This scattered radiation intensity was found to vary from a very low value to values as high as 20 per cent of the total radiation observed on certain days in Pittsburgh. The curves and tables are for combined direct solar and scattered sky radiation, and are given to represent expected design radiation intensity for August 1. They were prepared by the A.S.H.V.E. Laboratory from data[1] obtained by pyrheliometer observations.

A study of these curves discloses the periodic relationship and wide variation in solar intensity on various surfaces. It will be observed that both the roof (horizontal surface) and south wall radiation curves are in exact phase relationship with each other, while those for the east and

[1]A.S.H.V.E. RESEARCH REPORT NO. 1147—Heat Gain Through Glass Blocks by Solar Radiation and Transmittance, by F. C. Houghten, David Shore, H. T. Olson and Burt Gunst (A S.H V.E. TRANSACTIONS, Vol. 46, 1940, p. 83).

west walls overlap each other due to scattered sky radiation on the west wall during the forenoon and on the east wall during the afternoon. This phase relationship has an important bearing on the cooling load. Failure to consider the periodic character of heat flow resulting from diurnal movement of the sun and the lag due to heat capacity of the structure, which determine the timing and magnitude of the heat wave flowing through the wall, may result in a large error in load calculations.

The values of solar intensity appearing in Fig. 1 must not be confused with the actual heat transmission through the wall for much of the solar radiation impinging against the outer surface fails to pass through the wall. Instead it is delivered to the outside air by reflection, radiation, convection and conduction. A mathematical solution for the determination of solar heat transmission has been developed but the equations

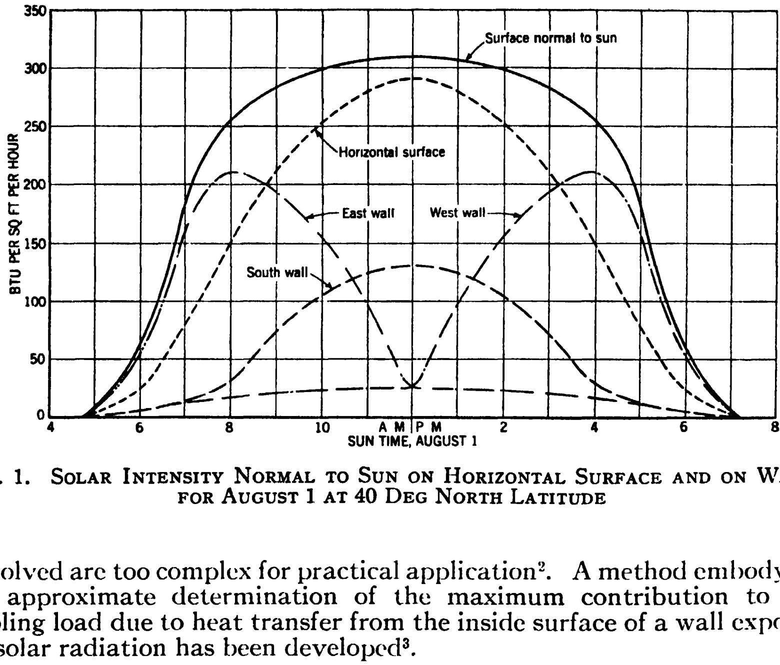


FIG. 1. SOLAR INTENSITY NORMAL TO SUN ON HORIZONTAL SURFACE AND ON WALLS FOR AUGUST 1 AT 40 DEG NORTH LATITUDE

involved are too complex for practical application[2]. A method embodying an approximate determination of the maximum contribution to the cooling load due to heat transfer from the inside surface of a wall exposed to solar radiation has been developed[3].

The heat flow in summer through various types of roofs and walls has been measured by the A.S.H.V.E. Laboratory. The curves in Fig. 2, give the heat flow through the inside surface of roofs[4] with details of the construction of the roofs tested. The conditions for which these results are given are: solar radiation for 40 deg north latitude on August 1

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[2]A S H V E RESEARCH REPORT No 923—Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F C. Houghten, J L Blackshaw, E M Pugh and Paul McDermott (A S H V E TRANSACTIONS, Vol 38, 1932, p 231). Effect of Heat Storage and Variation in Outdoor Temperature and Solar Intensity on Heat Transfer Through Walls, by J S Alford, J E Ryan and F O Urban (A S H V E TRANSACTIONS, Vol 45, 1939, p. 369) Periodic Heat Flow in Building Walls Determined by Electrical Analogy Method, by Victor Paschkis (A S H V E TRANSACTIONS, Vol. 48, 1942, p 75) Summer Comfort Factors as Influenced by the Thermal Properties of Building Materials, by C O. Mackey and L T. Wright, Jr (A S.H V.E TRANSACTIONS, Vol 49, 1943, p 148).

[3]Periodic Heat Flow—Homogeneous Walls or Roofs, by C O Mackey and L T Wright, Jr (A S H V E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September, 1944, P 546)

[4]A.S H.V E RESEARCH REPORT No 1157—Summer Cooling Load as Affected by Heat Gain Through Dry, Sprinkled and Water Covered Roofs, by F C. Houghten, H T Olson and Carl Gutberlet (A S H V E TRANSACTIONS, Vol. 46, 1940, p. 231).

TABLE 2. SOLAR RADIATION (DIRECT PLUS SKY) IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS AND A HORIZONTAL SURFACE

For 30 Deg North Latitude on August 1

Sun Time	Intensity of Solar Radiation, Btu per Sq Ft per Hour							
	Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal Surface
5:20	0	0	0	0	0	0	0	0
6:00	37	47	23	4.5	4.5	4.5	4.5	11
7:00	119	145	91	11	11	11	11	64
8:00	153	207	149	17	17	17	17	147
9:00	130	194	158	35	21	21	21	213
10:00	86	152	143	63	23.5	23.5	23.5	262
11:00	35	94	85	80	25.5	25.5	25.5	290
12:00	26	26	65	85	65	26	26	300
1:00	25.5	25.5	25 5	80	85	94	35	290
2:00	23.5	23.5	23.5	63	143	152	86	262
3:00	21	21	21	35	158	194	130	213
4:00	17	17	17	17	149	207	153	147
5:00	11	11	11	11	91	145	119	64
6:00	4.5	4.5	4.5	4.5	23	47	37	11
6:40	0	0	0	0	0	0	0	0

TABLE 3. SOLAR RADIATION (DIRECT PLUS SKY) IMPINGING AGAINST WALLS HAVING SEVERAL ORIENTATIONS AND A HORIZONTAL SURFACE

For 35 Deg North Latitude on August 1

Sun Time	Intensity of Solar Radiation, Btu per Sq Ft per Hour							
	Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal Surface
5:07	0	0	0	0	0	0	0	0
6:00	43	49	27	4.5	4.5	4.5	4.5	13
7:00	121	151	97	11	11	11	11	72
8:00	147	207	155	25	17	17	17	151
9:00	120	194	169	49	21	21	21	213
10:00	71	152	156	83	23.5	23.5	23.5	245
11:00	28	94	129	103	25.5	25.5	25.5	288
12:00	26	26	84	109	84	26	26	298
1:00	25.5	25.5	25.5	103	129	94	28	288
2:00	23.5	23.5	23.5	83	156	152	71	245
3:00	21	21	21	49	169	194	120	213
4:00	17	17	17	25	155	207	147	151
5:00	11	11	11	11	97	151	121	72
6:00	4.5	4 5	4.5	4.5	27	49	43	13
6:53	0	0	0	0	0	0	0	0

Table 4. Solar Radiation (Direct plus Sky) Impinging Against Walls Having Several Orientations and a Horizontal Surface

For 40 Deg North Latitude on August 1

Sun Time	Intensity of Solar Radiation, Btu per Sq Ft per Hour							
	Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal Surface
4:50	0	0	0	0	0	0	0	0
5:00	5	6	4	2.5	2.5	2.5	2.5	5
6:00	49	56	32	4.5	4.5	4.5	4.5	20
7:00	123	162	109	11	11	11	11	85
8:00	137	211	166	29	17	17	17	160
9:00	102	195	181	74	21	21	21	212
10:00	54	152	171	103	23.5	23.5	23.5	244
11:00	28	94	144	124	41	25.5	25.5	281
12:00	26	26	98	128	98	26	26	290
1:00	25.5	25.5	41	124	144	94	28	281
2:00	23.5	23.5	23.5	103	171	152	54	244
3:00	21	21	21	74	181	195	102	212
4:00	17	17	17	29	166	211	137	160
5:00	11	11	11	11	109	162	123	85
6:00	4.5	4.5	4.5	4.5	32	56	49	20
7:00	2.5	2.5	2.5	2.5	4	6	5	5
7:10	0	0	0	0	0	0	0	0

Table 5. Solar Radiation (Direct plus Sky) Impinging Against Walls Having Several Orientations and a Horizontal Surface

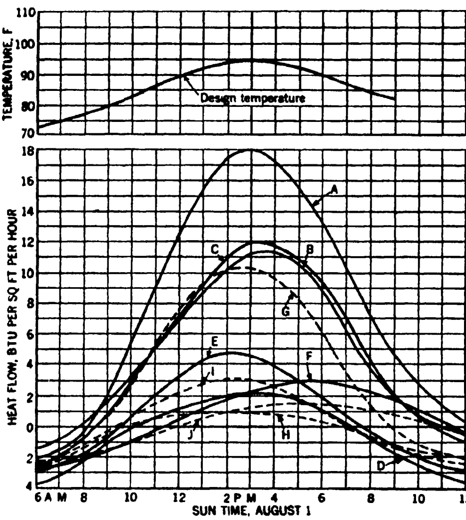
For 45 Deg North Latitude on August 1

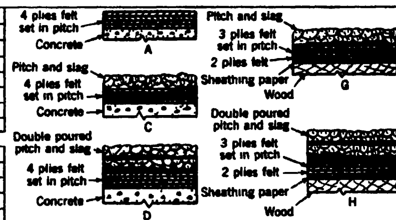
Sun Time	Intensity of Solar Radiation, Btu per Sq Ft per Hour							
	Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal Surface
4:25	0	0	0	0	0	0	0	0
5:00	22	20	17	3.5	3.5	3.5	3.5	9
6:00	87	99	56	5.5	5.5	5.5	5.5	27
7:00	151	192	134	12	12	12	12	89
8:00	144	237	188	48	17	17	17	156
9:00	100	199	197	93	21	21	21	205
10:00	46	153	184	121	23.5	23.5	23.5	243
11:00	28	94	158	146	63	25.5	25.5	259
12:00	26	26	116	156	116	26	26	281
1:00	25.5	25.5	63	146	158	94	28	259
2:00	23.5	23.5	23.5	121	184	153	46	243
3:00	21	21	21	93	197	199	100	205
4:00	17	17	17	48	188	237	144	156
5:00	12	12	12	12	134	192	151	89
6:00	5.5	5.5	5.5	5.5	56	99	87	27
7:00	3.5	3.5	3.5	3.5	17	20	22	9
7:35	0	0	0	0	0	0	0	0

as given in Fig. 1 and Table 4; outdoor design temperature reaching a maximum of 95 F as shown by the temperature curve in Fig. 2 and an indoor temperature of 75 F.

Curves in Fig. 3 were prepared by the A.S.H.V.E. Laboratory from recent tests made there and show the heat flow through the inside surface of three types of walls for various orientations[5]. The results are given for the following conditions: 90 per cent of the solar radiation given in Fig. 1 and Table 4 for 40 deg north latitude on August 1; outdoor design temperature reaching a maximum of 93 F as shown by the temperature curve in Fig. 3 and an indoor temperature of 78 F and 50 per cent relative humidity.

The heat flow shown in Figs. 2 and 3 is a combination of normal transmission and solar radiation transmission and is the total heat flow through the wall or roof. Due to the heat capacity of walls and roofs there is a





Curve	Type of roof and condition tested	U = Btu per sq ft per hr per F temp difference on two sides	Heat capacity, Btu per sq ft per F (Values for dry roof)
A	2 in concrete, smooth asphalt finish (black)	0.40	6.28
B	A plus aluminum paint		
C	2 in concrete, pitch and slag	0.31	7.37
D	2 in concrete, double poured pitch and slag (sprinkled)	0.26	8.27
E	D with 1 in water		
F	D with 6 in water		
G	2 in plank, pitch and slag	0.20	3.97
H	2 in plank, double poured pitch and slag (sprinkled)	0.18	4.95
I	H with 1 in water		
J	H with 6 in water		

Fig. 2. Relation Between Time and Heat Flow Through Inside Surface of Horizontal Roofs Corrected to Design Day of August 1

time lag[6] in the transmission of heat through them as shown by the curves. For the types of construction covered in Figs. 2 and 3 and for the conditions indicated, the heat flow through the inside surface at any given time can be read directly. For other types of construction, the curves may be used as a guide in estimating the heat flow. The time lag for other types of construction is included in Table 6 which was prepared by the A.S.H.V.E. Laboratory from data collected by it and by other authorities.

Solar Radiation Transmitted Through Glass

Windows present a problem somewhat different from that of opaque walls, because they permit a large percentage of the solar energy to pass

[5]A.S.H.V.E. Research Report No 1195—Heat Gain Through Walls and Roofs as Affected by Solar Radiation, by F. C. Houghten, E. C. Hach, S I. Taimuty and Carl Gutberlet (A.S.H V.E. Transactions, Vol. 48, 1942, p. 91).

[6]Loc. Cit. Note 2.

through. A small amount is reflected and a portion is absorbed by the glass. The amount absorbed depends upon the character and thickness of the glass and the angle between it and the sun's rays. The temperature of the glass is raised by the absorbed heat and this heat is then delivered to the air on each side in proportion to the difference between the glass and air temperatures[7].

The A.S.H.V.E. tests[8] indicate that a single pane of double strength glass 0.127 in. thick absorbs approximately 11 per cent of the solar radiation passing through it when the impingement is normal. For smaller angles of impingement, the glass retards percentages of the total radiant energy approximately in proportion to the sine of the angle.

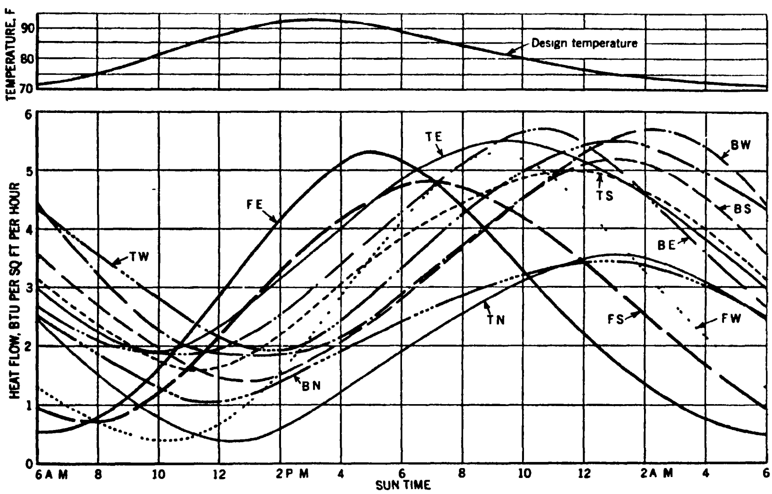


FIG. 3. RELATION BETWEEN TIME AND HEAT FLOW THROUGH THE INSIDE SURFACE OF WALLS OF DIFFERENT CONSTRUCTION AND ORIENTATION ON A 93 F DESIGN DAY WITH 90 PER CENT OF DESIGN SOLAR RADIATION

Walls BE, BS, BW and BN—12 in. solid brick and plaster facing east, south, west and north respectively
Walls TE, TS, TW and TN—4 in. brick veneer, 8 in. tile and plaster facing east, south, west and north respectively.
Walls FE, FS and FW—4 in. brick veneer, building paper, ⅞ in, matched sheathing, 2 x 4 in studs, metal lath and plaster facing east, south and west respectively

The amount of solar radiation delivered to an unshaded glass surface may be obtained from Tables 2, 3, 4 or 5. These values must be used only for the net glass area on which the sun shines and not the entire glass area. Tests at the A.S.H.V.E. Research Laboratory[9] have determined the percentage of heat from solar radiation actually delivered to a room with various types of outdoor and indoor shading. The data in Table 7 are taken from these tests.

The percentage values in this table were obtained by dividing the total amount of heat actually entering through the shaded window by the total amount of heat calculated to enter through a bare window (solar

[7]Heat Absorbing Glass Windows, by W. W. Shaver (A.S H V.E. TRANSACTIONS, Vol. 41, 1935, p. 287).

[8]A.S.H.V.E. RESEARCH REPORT No. 974—Radiation of Energy Through Glass, by J. L. Blackshaw and F. C. Houghten (A S H V.E. TRANSACTIONS, Vol. 40, 1934, p. 93). A.S.H.V.E. RESEARCH REPORT No 975—Studies of Solar Radiation Through Bare and Shaded Windows, by F. C. Houghten, Carl Gutberlet, and J. L. Blackshaw (A S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 101) A S H.V.E RESEARCH REPORT No. 1180—Heat Gain Through Western Windows With and Without Shading, by F. C. Houghten and David Shore (A.S H V E. TRANSACTIONS, Vol 47, 1941, p. 251).

[9]Loc Cit. Note 8.

TABLE 6. TIME LAG IN TRANSMISSION OF SOLAR RADIATION THROUGH WALLS AND ROOFS

TYPE AND THICKNESS OF WALL OR ROOF	TIME LAG, HOURS
1-in. yellow pine horizontal roof, water proofing, smooth black finish	1
2-in. yellow pine horizontal roof, water proofing, smooth black finish	1¾
4-in. reinforced clay tile horizontal roof, water proofing, slag finish	2¼
2-in. gypsum horizontal roof, water proofing, slag finish	2¼
Slate and slaters felt on 2¼ in. tongue and grooved yellow pine, sloped roof	2½
4-in. gypsum horizontal roof, water proofing, slag finish	4½
6-in. concrete horizontal roof, water proofing, slag finish	5
1-in. concrete, 4-in. cinders, 1½-in. concrete, water proofing, smooth black finish	8
Wood siding, 1-in. sheathing, 2 x 4 studs, lath and plaster	2
Wood siding, 1-in. sheathing, 2 x 4 studs (studding space filled with insulation) lath and plaster	5
4-in. brick, 1-in. sheathing, 2 x 4 studs, lath and plaster	7
4-in. brick, 8-in. tile and plaster	10½
13-in. brick, plastered	12
9-in. brick, 3¾-in. tile, 5¼-in. air space, 3¾-in. tile and 1¼-in. plaster	16

radiation plus glass transmission, based on observed outside glass temperature). For bare windows on which the sun shines, the transmission of heat from outside air to glass may be small or negative as the glass temperature is raised by the solar radiation absorbed.

In calculating the total heat gain through windows on the sunny side of buildings, it is sufficiently accurate to proceed as outlined herewith:

Consider the total heat gain as that resulting from solar radiation and neglect the heat transmission through the glass caused by the difference between the temperatures of the inside and outside air. This method should be used except at times when the calculated heat gain per square foot due to normal transmission exceeds the solar intensity. At such times, solar radiation may be neglected and the total heat gain considered as resulting from normal transmission.

The solar heat transmission through windows or skylights may be expressed by the formula:

$$H_G = A_G f I \qquad (2)$$

where

H_G = solar radiation transmitted through a window, Btu per hour.

A_G = net area of glass exposed to sun's rays, square feet.

f = percentage of solar radiation (expressed as a decimal) transmitted to the inside (Table 7). For bare windows, $f = 1$.

I = intensity of solar radiation striking surface, Btu per hour per square foot (Tables 2, 3, 4 and 5).

TABLE 7. SOLAR RADIATION TRANSMITTED THROUGH SHADED WINDOWS

TYPE OF APPURTENANCE	FINISH FACING SUN	PER CENT DELIVERED TO ROOM
Canvas awning	Plain	28
Canvas awning	Aluminum	22
Inside shade, fully drawn	Aluminum	45
Inside shade, one-half drawn	Buff	68
Inside Venetian blind, fully covering window, slats at 45 deg.	Aluminum	58
Outside Venetian blind, fully covering window, slats at 45 deg.	Aluminum	22

In Equation 2, $f = 1$ for bare windows because the tests from which Table 7 was obtained showed that approximately all of the solar radiation impinging on a bare window became a part of the heat load in the room. This was because almost all of the heat absorbed by the glass flowed into the room by conduction. Other tests[10] have indicated that in the case of a building having floors of high heat capacity such as concrete floors on which the solar radiation falls, some of the heat entering a bare window is absorbed by the floor and does not immediately become a part of the cooling load, but is delivered back to the air in the building at a slow rate.

The maximum solar intensity on any surface is of limited duration as shown in Fig. 1. In the case of windows the total energy impinging on the glass before and after the time of maximum intensity is further reduced

TABLE 8. HEAT GAIN THROUGH GLASS BLOCKS[a]

		Solar Radiation Heat Gain (Direct plus Sky) Btu per Sq Ft per Hour						Total Heat Gain[b] (Solar Radiation plus Normal Transmission) Btu per Sq Ft per Hour					
Side		East[c]	West[c]	South				East[c]	West[c]	South			
N. Latitude Degrees		40	40	30	35	40	45	40	40	30	35	40	45
Sun Time	Outside Temp F												
7:00	74	65.0		1.0	2.8	3.0	5.0	61.0		−4.5	−2.0	−0.5	1.0
8·00	76	63.0	0.0	3.0	4.4	6.5	11.0	77.5		0 0	2.0	4.0	5.0
9:00	79	40.0	5.0	5.5	7.1	10.2	13.4	73.5	5 0	5.0	7.0	10 0	12.0
10:00	83	24.0	6.0	8.5	11.3	14.7	17.1	57.5	6.5	11 0	15.0	18.0	20 8
11:00	87	15 5	7.0	12 0	15.2	18 7	21.8	45.0	7.5	16.5	22 0	25.5	32 0
12:00	90	10.0	10.0	14.0	17.4	21 0	24.8	36.5	10 5	21 5	28.0	33.8	40 8
1:00	93	7 0	15.5	12.0	15.2	18.7	21 8	30.0	22.0	25 0	31.8	38.5	46 0
2:00	94	6 0	24.0	8.5	11.3	14.7	17.1	24.0	35.0	26 0	32.0	39.0	47.0
3:00	95	5.0	40 0	5.5	7 1	10.2	13 4	19 5	55 0	24.0	29.8	36 5	45 0
4:00	95	4.5	65.0	3.0	4 4	6 5	11.0	15 5	77.0	20.0	25.5	31 5	40 5
5:00	93	4 0	63.0	1 0	2 8	3 0	5 0	12 5	85 5	15.0	20.0	25.2	33 5
6:00	91	2.5	23.5	0.0	0 7	0 7	3.0	10 5	55.0	9 5	13.5	18.0	25 5
7:00	89	1 5	0.0			0.0	0 7	8.0	18.5	3.5	7.0	11.0	18 0

[a]For August 1.

[b]Inside temperature, 78 F.

[c]For east and west walls these values can be applied to all latitudes between 30 and 45 deg N without excessive errors.

by increased shading of the glass from the frame, or wall. The cooling load due to solar radiation therefore does not have to be calculated as a steady load. Another point which should be noted is that the maximum solar radiation load on the east wall occurs early in the morning when the outside temperature is low.

Tests have been made which indicate that solar radiation through window glass is the most important factor to contend with in the cooling of an office building. At times it was shown to account for as much as 75 per cent of the total internal sensible cooling necessary. Because of the importance of the sun load, cooling systems should be zoned so that the side of the building on which the sun is shining can be controlled separately from the other sides of the building. If buildings are provided with awnings so that the window glass is shielded from sunshine, the

[10]A.S H.V.E. RESEARCH REPORT No. 1002—Cooling Requirements of Single Rooms in a Modern Office Building, by F. C. Houghten, Carl Gutberlet, and Albert J. Wahl (A S H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 53).

amount of cooling required will be reduced and there will also be less difference in the cooling requirements of different sides of the building. The total cooling load for a building exposed to the sun on more than one side is of course less than the sum of the maximum cooling loads in the individual rooms since the maximum solar radiation load on the different sides occurs at different times. In determining the total cooling load for a building if the time when the maximum load occurs is not obvious, the load should be calculated for various times of day to determine the times at which the sum of the loads on the different sides of the building is a maximum.

The direct solar and scattered sky radiation penetration through glass block panels is given in Table 8 for various times of the day for south, east and west exposures for different latitudes on August 1. This table also gives the total heat gain into an air conditioned space when 78 F is maintained indoors, resulting from the effect of both radiation and air to air transmission. These values result from A.S.H.V.E. Laboratory data[11] and apply for expected design radiation intensity, and for a design day having a maximum temperature of 95 F. The resulting heat gains are averages for four typical glass block designs, two having smooth exterior faces, and the other two having exterior ribbed faces.

Heat Emission of Occupants

The heat and moisture given off by human beings under different states of activity are shown in various tables and figures of Chapter 2 which covers the physical and physiological principles of air conditioning. It will be observed from these data that the rate of sensible and latent heat emission by human beings varies greatly depending upon state of activity. In many applications this component becomes a large percentage of total load. Metabolic rates are markedly variable for some extreme environmental conditions and this is another important factor which must be considered in cooling load computations.

Heat Introduced by Outside Air

An allowance must be made for the heat and moisture in the outside air introduced for ventilation purposes and entering the building through cracks, doors, and other places where infiltration might occur.

The volume of air entering due to infiltration may be estimated from data given in Chapters 5 and 6. Information on the amount of outside air required for ventilation will be found in Chapter 2.

The possible peak load caused by infiltration and ventilation requirements must be carefully considered. In general, as the ventilation increases the infiltration will tend to decrease. The external pressure on the windward side of the building is often greater than the pressure within and under this condition there will still be infiltration even with large ventilation quantities. Frequently this does not appreciably affect the refrigeration calculation as infiltration can usually be compensated for by decreasing ventilation. Infiltration, however, does affect the required apparatus dew-point, particularly in rooms of one exposure.

The heat gain resulting from outside air introduced may be determined by Equation 3:

$$H = \frac{Q}{v}(h_o - h_i) \tag{3}$$

[11]Loc. Cit. Note 1.

where

H = heat to be removed from outside air entering the building, Btu per hour.
Q = volume of outside air entering building, cubic feet per hour.
v = cubic feet of outside air per pound of dry air.
h_o = enthalpy of outside air, Btu per pound of dry air.
h_i = enthalpy of inside air, Btu per pound of dry air.

The latent heat gain resulting from outside air introduced may be determined by Equation 4:

$$H_l = \frac{Q}{v} h_{fg} (W_o - W_i) \qquad (4)$$

where

H_l = latent heat to be removed, Btu per hour.
h_{fg} = latent heat of evaporation at temperature at which water is condensed, Btu per pound.
W_o = humidity ratio of outside air, pounds water per pound dry air.
W_i = humidity ratio of inside air, pounds water per pound dry air.

Heat Emission of Appliances

Heat generating appliances which give off either sensible heat or both sensible and latent heat in an air conditioned enclosure may be divided into three general classes of equipment or devices: (1) electrical appliances, (2) gas appliances, and (3) steam heating appliances.

In the first group may be found such devices as lights[12], fans, motors, toasters, waffle irons, etc. The wattages are usually marked on the name-plates and it is only necessary to multiply the aggregate wattage by 3.413 (Btu per watthour) in order to estimate the heat added to the conditioned space by such devices in Btu per hour.

Electric motors are usually rated in units of horsepower *output*. To determine the corresponding input, which is the rate at which heat is added to the conditioned space by full-load operation of such motors, some idea of motor efficiency is necessary. The aggregate input in horsepower should then be multiplied by 2546 (Btu per horsepower hour). When motor efficiencies are not known, the data in Table 9 may be used.

Table 9. Heat Generated by Motors

Name-plate Rating Horsepower	Heat Gain in Btu per Hour per Horsepower	
	Connected Load in Same Room	Connected Load Outside of Room
⅛ to ½	4250	1700
½ to 3	3700	1150
3 to 20	2950	400

In the second group belong such appliances as coffee urns, gas ranges, steam tables, broilers, hot plates, etc. For heat generating capacities of such appliances refer to Table 10.

Judgment must be used in the application of data given in Table 10. Consideration must be given to the heat contributed by appliances which are in use at the time of peak load. The quantity of heat will

[12]Cooler Footcandles for Air Conditioning, by W G. Darley (A S H.V E. Transactions, Vol 46, 1940, p. 367). Lighting and Air Conditioning Design Factors, Report of *I. E S*—A S H.V.E. Joint Committee on Lighting in Air Conditioning (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, September, 1941, p 605). Lighting and Air Conditioning, by Howard M Sharp (*Heating and Ventilating*, November, 1942, p 35)

Table 10. Heat Gain from Various Sources[a]

Source	Per Cent Name-plate Rating	Percentage or Btu per Hour		
		Sensible	Latent	Total
Electrically Heated Equipment				
1. Electric Oven-Baking		80%	20%	100%
2. Baker's Oven	70	80%	20%	100%
3. Coffee Urn—per Gallon Capacity	.....	1025	1025	2050
4. Glass Coffee Maker—per Section	20	90%	10%	100%
5. Warming Receptacle	30	80%	20%	100%
6. Plate Warmer	50	100%	0%	100%
7 Hot Plates	80	.....	.....	.....
8. Steam Table—Water Bath Type	100	65%	35%	100%
9 Frying Griddles	75	90%	10%	100%
10 Fry Kettle	70	.....	.....	.....
11. Waffle Baker	40	75%	25%	100%
12 Sandwich Grille	60	90%	10%	100%
13 Toaster—Intermittent or Timed Control	50	90%	10%	100%
14 Toaster—Continuous	100	90%	10%	100%
15 Hair Dryer in Beauty Parlor—600 w	.....	2050	.....	2050
16. Permanent Wave Machine in Beauty Parlor—24-25 w Units	.....	2050	.....	2050
Gas Burning Equipment[b]				
17 Gas Heated Oven - Baking .. *no vent connection*	70	72%	28%	100%
gravity vent connection	35	100%	0%	100%
18 Open Top Burner—per Hole	50	45%	55%	100%
19 Hot or Closed Top—per Burner *no vent connection*	100	45%	55%	100%
gravity vent connection	70	50%	50%	100%
20 Coffee Urn—Large 18 in Diam —Single Drum	.....	5000	5000	10,000
21 Coffee Urn—Small 12 in Diam —Single Drum	.....	3000	3000	6000
22 Coffee Urn—per Gallon of Rated Capacity	.....	500	500	1000
23 Plate Warmer *no vent connection*	60	90%	10%	100%
gravity vent connection	10	100%	0%	100%
24 Steam Table—Water Bath Type *no vent connection*	60	55%	45%	100%
gravity vent connection	42	50%	50%	100%
25 Frying Griddles *no vent connection*	75	81%	19%	100%
gravity vent connection	52	90%	10%	100%
26. Fry Kettle *no vent connection*	70	72%	28%	100%
gravity vent connection	49	80%	20%	100%
27. Egg Boiler—per Egg Compartment	.....	2500	2500	5000
28 Cigar Lighter—Continuous Flame Type	.....	2250	250	2500
29 Curling Iron Heater	.....	2250	250	2500
30 Pilot Light		180	20	200
Steam Heated Equipment[c]				
31. Steam Heated Surface Not Polished—per Square Foot of Surface	.....	330	0	330
32. Steam Heated Surface Polished—per Square Foot of Surface	.....	130	0	130
33 Insulated Surface—per Square Foot	.....	80	0	80
34. Bare Pipes, Not Polished—per Square Foot of Surface	.....	400	0	400
35 Bare Pipes, Polished—per Square Foot of Surface	.....	220	0	220
36. Insulated Pipes—per Square Foot	.....	110	0	110
37. Coffee Urn—Large, 18 in Diam —Single Drum	.....	2000	2000	4000
38. Coffee Urn—Small, 12 in Diam —Single Drum	.....	1200	1200	2400
39. Egg Boiler—per Egg Compartment	.....	2500	2500	5000
40. Steam Table—per Square Foot of Top Surface	.....	300	800	1100
Miscellaneous				
41. Heat Liberated By Food per Person, as in a Restaurant	.....	30	30	60
42. Heat Liberated from Hot Water used direct and on towels per hour—Barber Shops	.....	100	200	300

[a] Heat gain from electric or gas residential ranges or cooking stoves depends on size of the family, socio-economic status of the individual, time of day for principal meal, and whether the equipment is manually or automatically controlled. Total heat gain will probably not exceed 40 per cent name-plate rating. Per cent sensible and latent heat will depend upon use of equipment; dry heat; baking or boiling.

[b] Name-plate ratings of gas burning equipment can be obtained from a Directory of Approved Gas Appliances and Listed Accessories, obtainable from *American Gas Association*

[c] Steam Requirements of Process Equipment, Report of the Commercial Relations Committee, *National District Heating Association* (*Heating, Piping and Air Conditioning*, November, 1942, p 675)

depend upon whether products of combustion are vented to a flue, whether they escape into the space to be conditioned, or whether appliances are hooded allowing part of the heat to escape through a stack. There are no generally accepted data available on the effects of venting and shielding heating appliances but it is believed that, when they are properly hooded with a positive fan exhaust system through the hood, 50 per cent of the heat will be carried away and 50 per cent dissipated in the space to be conditioned. Where latent as well as sensible heat is given off, it is usually safe to assume that all latent heat will be removed by a properly designed and operated vent or hood.

TABLE 11. PERMEABILITY OF VARIOUS MATERIALS TO WATER VAPOR

Material	Permeability Grains per Sq Ft per Hr per Inch Hg
	Group 1[a]
Plaster base and plaster, ¾ in.	14.7
Fir sheathing, ¾ in.	2.9
Waterproof paper	49.1
Pine lap siding	4.9
Paint film	3.4
Sugar cane fiberboard, ¾ in.	12.5
Brick masonry, 4 in.	1.1
	Group 2[b]
Foil-surfaced reflective insulation, double-faced	0.08 to 0.13
Roll roofing—smooth, 40 to 65 lb/roll 108 sq ft	0 13 to 0.17
Duplex or laminated papers, 30-30-30	1 37 to 2.58
Plaster, wood lath	11.00
Plaster, 3 coats of lead and oil	3.68 to 3 84
Plaster, 2 coats of Al. paint	1.15
Plaster, fiberboard or gypsum lath	19.73 to 20.57
Plywood, ½ in., 5-ply Douglas fir	2.67 to 2.74
Insulating lath and sheathing, board type	25.68 to 34.27
Insulating sheathing, surface-coated	3.03 to 4.36
Insulating cork blocks, 1 in.	6.19
Mineral wool, unprotected, 4 in.	29.07

[a]Calculating Vapor and Heat Transfer Through Walls, by L. G. Miller (*Heating and Ventilating* 35 No. 11, 56 November, 1938)

[b]How to Overcome Condensation in Building Walls and Attics, by L. V. Teesdale (*Heating and Ventilating*, Vol. 36, No. 4, April, 1939)

Moisture Through Walls

In some applications walls of the conditioned space may be in contact with other spaces which have in them a higher water vapor pressure than that in the conditioned space. It is known that water vapor will flow through the building materials in proportion to the vapor pressure difference on the two sides of the material. The total amount of water vapor transmitted is dependent on the permeability which is usually expressed in grains of moisture per square foot per hour per inch of mercury vapor pressure difference. The values for permeability in Table 11 are quoted from a publication of the *National Bureau of Standards*[13]. The water vapor entering the conditioned space must be added to the latent cooling load.

[13]Moisture Condensation in Building Walls, by Harold W. Woolley (*U. S. Department of Commerce, National Bureau of Standards*, Building Materials and Structures Report BMS63).

Vapor barriers, to be effective in reducing entrance of moisture, must seal completely the walls, ceilings, and floors that are exposed to space having excessive vapor pressure and all doors must have gaskets applied to them to make the barrier effective.

ILLUSTRATION

From the foregoing discussion it is obvious that the determination of the maximum cooling load is rather complicated by reason of the variable nature of contributing load components. An illustrative example will explain the method presented in the foregoing text. Alternative procedures for calculating cooling loads have been devised by various writers. One of the most recent, intended particularly for residential installations, is given in Bulletin No. 18 of the *American Gas Association.*

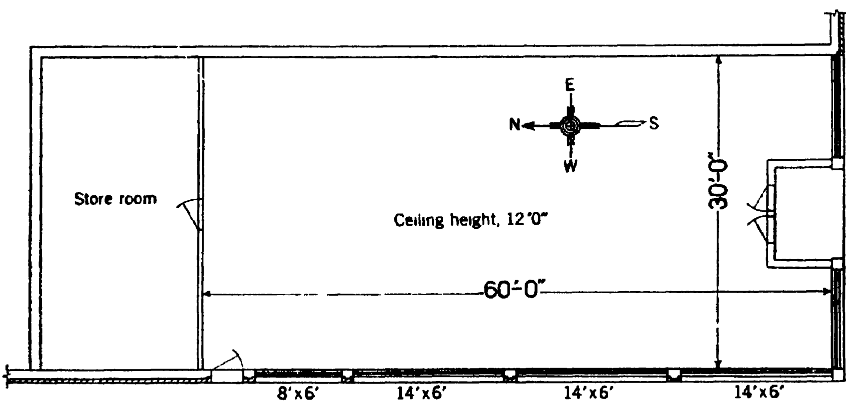


FIG. 4. PLAN DIAGRAM OF CLOTHING STORE

Example 1. Determine cooling load requirements for a clothing store illustrated in Fig. 4 and located in Pittsburgh, Pa., Latitude 40 deg. This is a one-story building located on a corner and it faces south and west. Assume building on east and north sides conditioned.

Wall construction, 8 in. concrete block, 4 in. brick veneer, plaster on walls, $U = 0.33$ (Table 7, Chapter 4, No. 93 B)

Roof construction, 2 in concrete, $1\frac{1}{2}$ in insulating board, metal lath and plaster ceiling, $U = 0.26$ (Table 14, Chapter 4, No 14 B).

Floor, maple flooring on yellow pine, no ceiling below, $U = 0.34$ (Table 10, Chapter 4, No 1-N).

Partition, wood lath and plaster on both sides of studding, $U = 0.34$ (Table 8, Chapter 4, No. 3 B)

Windows, provided with awnings.

Front doors, 2 ft 6 in x 7 ft (glass paneled)

Side door, 3 ft x 7 ft (glass paneled), $U = 1.13$ (Table 17 A, Chapter 4).

Occupancy, 10 clerks, 40 patrons

Lights, 4200 w.

Outside design conditions, dry-bulb 95 F; wet-bulb 75 F.

Inside design conditions, dry-bulb 80 F; wet-bulb 67 F.

Basement temperature, 85 F.

Store room temperature, 88 F.

Solution. It is obvious from the shape and exposure of this store and the large glass area on the west side that the maximum cooling load will occur during the afternoon when the sun is shining on the west wall. From Fig. 1, the peak load may be expected at 4:00 p.m.

The combined normal transmission and solar radiation transmission through the roof at 4:00 p. m. is obtained from Fig. 2. While none of the roofs in Fig. 2 is exactly like this one, roof *C* is similar. A heat flow of 11 Btu per square foot per hour was assumed, slightly less than for roof *C*. The combined normal transmission and solar radiation

COMBINED NORMAL AND SOLAR RADIATION TRANSMISSION:

SURFACE	DIMENSIONS	AREA SQ FT	BTU PER HOUR PER SQ FT	BTU PER HOUR
S Wall	(30 ft x 12 ft) − 155	205	3	615
W Wall	(60 ft x 12 ft) − 321	399	2.5	998
Roof	60 ft x 30 ft	1800	11	19,800
Total				21,413

NORMAL TRANSMISSION:

SURFACE	DIMENSIONS	AREA SQ FT	U	TEMP. DIFF DEG F	BTU PER HOUR
S Glass	2 (2 ft 6 in. x 7 ft) + 2 (10 ft x 6 ft)	155	1.13	15	2,627
Floor	26 ft x 54 ft	1404	0.34	5	2,387
N Partition	30 ft x 12 ft	360	0.34	8	979
Total					5,993

transmission through the south and west walls at 4:00 p.m. is obtained from curves *TS* and *TW* in Fig. 3.

The normal heat transmission through the south glass, floor and partition is determined by application of Formula 1. Solar radiation transmission through the south glass can be neglected. The solar intensity *I* for the south side at 4:00 p m. is 29. Applying a shade factor of 0.28, the calculated solar radiation transmission is $29 \times 0.28 = 8$ Btu per square foot per hour which is less than the normal transmission; therefore the total heat gain can be taken as that due to normal transmission.

Solar radiation intensity on the west glass at 4:00 p.m. from Table 4 is 211 Btu per square foot per hour. As explained in the text, normal transmission can be neglected because it is small in comparison with solar radiation transmission.

To determine the heat gain from the outside air it is necessary first to determine the volume of the outside air to be introduced. Since the windows are sealed so as not to permit infiltration and since there are only three doors in this store through which infiltration can take place, infiltration is neglected in solving this example. The dew-point within the store is, however, affected by the outside air admitted when the doors are in use.

Assuming a constant apparatus dew-point, the infiltration will affect the dew-point within the space, raising it when outside air is above the line drawn on the psychrometric chart between the apparatus dew-point and the room conditions and lowering it when it is below. The volume of the store is 21,600 cu ft. Good practice indicates that in a store of this character there should be a minimum of from 1 to 1½ outside air changes per hour. On a basis of 1½ air changes the volume of outside air to be introduced would be 32,400 cu ft per hour. The minimum ventilation requirements as given in the CODE OF MINIMUM REQUIREMENTS FOR COMFORT AIR CONDITIONING[14] are 10 cfm per

[14]Code of Minimum Requirements for Comfort Air Conditioning (A.S H V E TRANSACTIONS, Vol 44, 1938, p 27).

person On this basis the ventilation requirements would be 30,000 cu ft per hour. Since this will produce approximately 1½ outside air changes per hour, 30,000 cu ft per hour will be considered in this application.

To determine load imposed by occupants it will be found from Chapter 2 that the average person standing at rest will dissipate 431 Btu per hour and that the moisture dissipated in 0.198 lb per hour.

To determine the latent heat load, the sum of the moisture evaporated from occupants and that to be removed from outside air is multiplied by the latent heat of evaporation at the temperature at which the moisture is condensed in the conditioner. Since outside air is positively introduced, a mixture of outside and recirculated air passes through the conditioner. To remove the moisture, the air must be cooled to a temperature below the dew-point of the mixture To obtain an approximate value of the latent heat of evaporation, assume that the air is cooled to 55 F. At this temperature, h_{fg} = 1062 7 Btu per hour (steam table).

SOLAR RADIATION THROUGH GLASS.

W Glass. $A_G = 3\ (14\ \text{ft} \times 6\ \text{ft}) + (8\ \text{ft} \times 6\ \text{ft}) + (3\ \text{ft} \times 7\ \text{ft}) = 321$ sq ft.

$H_G = 321 \times 0\ 28 \times 211 = 18{,}965$ Btu per hour (Equation 2).

OUTSIDE AIR.

$H = \dfrac{Q}{v}\ (h_o - h_i)$ (Equation 3).

$v = v_a + \mu v_{as}$ (Equation 17, Chapter 1)

v_a = specific volume of dry air at 95 F = 13 97 cu ft per pound (Table 6, Chapter 1).

v_{as} = difference between volume of saturated mixture and specific volume of dry air at 95 F = 0.82 cu ft per pound (Table 6, Chapter 1).

μ = per cent saturation at 95 F dry-bulb and 75 F wet-bulb = 38 4 per cent (by calculation, Chapter 1)

$v = 13\ 97 + (0\ 384 \times 0.82) = 14\ 28$ cu ft per pound dry air.

$h_o = h_a + \mu h_{as}$ (Equation 19, Chapter 1).

h_a = specific enthalpy of dry air at 95 F = 22 80 Btu per pound (Table 6, Chapter 1).

h_{as} = difference between enthalpy of saturated mixture and specific enthalpy of dry air at 95 F = 40 25 Btu per pound (Table 6, Chapter 1).

$h_o = 22\ 80 + (0\ 384 \times 40\ 25) = 38\ 26$ Btu per pound dry air.

μ at 80 F dry-bulb and 67 F wet-bulb = 50 2 per cent (by calculation, Chapter 1).

$h_i = h_a + \mu h_{as} = 19\ 19 + (0\ 502 \times 24\ 32) = 31.40$ Btu per pound dry air (Table 6, Chapter 1)

$H = \dfrac{30{,}000}{14\ 28}\ (38\ 26 - 31.40) = 14{,}410$ Btu per hour.

W_o = humidity ratio of outside air at 95 F and 75 F = 0.384 × 0.03652 = 0.01402 lb water per pound dry air. (Equation 14, Chapter 1).

W_i = humidity ratio of inside air at 80 F and 67 F = 0 502 × 0 02221 = 0 01115 lb water per pound dry air. (Equation 14, Chapter 1).

Weight of water to be removed $= \dfrac{Q}{v}\ (W_o - W_i) = \dfrac{30{,}000}{14\ 28}\ (0\ 01402 - 0\ 01115) =$ 6.03 lb per hour.

OCCUPANTS:

50 × 431 = 21,550 Btu per hour.

50 × 0.198 = 9.90 lb water per hour evaporated

Lights:

4200 × 3.413 = 14,335 Btu per hour.

Summary:

Component of Load	Btu per Hour
Combined Normal and Solar Radiation Transmission	21,413
Normal Transmission	5,993
Solar Radiation Through Glass	18,965
Outside Air	14,410
Occupants	21,550
Lights	14,335
Total (Sensible and Latent Heat Gain)	96,666

Latent Heat:

Outside air 6.03 lb water per hour.
Occupants 9.90 lb water per hour.

15.93 lb water per hour.

$15.93 \times h_{fg} = 15.93 \times 1062.7 = 16{,}929$ Btu per hour.

Condition Line:

The condition line for this application may be determined from Equation 25 in Chapter 1 by taking the ratio of 96,666 ÷ 15.93 = 6070 Btu per pound water. In this case, it crosses the saturation curve at a temperature for which the enthalpy h_s and the humidity ratio W_s satisfy the equation,

$$\frac{31.40 - h_s}{0.01115 - W_s} = 6070.$$

A trial-by-error computation gives a quick solution of 57.4 F, which can also be determined graphically on a Mollier chart. This is the apparatus dew-point as explained in Chapters 1 and 20.

Refrigeration Load:

Assuming 100 per cent saturation efficiency for the air conditioning apparatus, operation at the apparatus dew-point is possible. The thermodynamic properties involved in calculating the refrigeration required are:

	Inside Air	*After Cooling*	*After Separating*
t	80.0	57.4	57.4
h	31.40	24.678	24.65
W	0.01115	0.01115	0.01004

The refrigeration required is 31.4 − 24.678 = 6.722 Btu per pound dry air. The total outside and recirculated dry air to be circulated through the air conditioning apparatus is 96,666 ÷ (31.40 − 24.65) = 14,321 lb per hour. Hence, the total refrigeration required or the cooling load is 14,321 × 6.722 = 96,265 Btu per hour, or 96,265 ÷ 12,000 (Btu extracted per hour per ton of refrigeration) = 8.02 tons.

The energy removed with the water eliminated is 14,321 × (24.678 − 24.65) = 401 Btu per hour. This plus the refrigeration accounts for the total removal of 96,265 + 401 = 96,666 Btu per hour as required.

CHAPTER 8

Combustion and Fuels

Principles of Combustion, Classification of Coals, Firing Methods for Coals, Firing Methods for Coke, Classification of Oils, Combustion of Oil, Classification of Gas, Combustion of Gas, Dustless Treatment of Coal

THE data given in the first part of this chapter are of general application to the various fuels used in domestic heating which are coal, coke, oil and gas. The choice of fuel is a question of dependability, cleanliness, fuel availability, economy, operating requirements and control.

FUNDAMENTAL PRINCIPLES OF COMBUSTION

Combustion may be defined as the chemical combination of a substance with oxygen with a resultant evolution of heat. The rate of combustion depends partly upon the specific rate of reaction of the combustible substance with oxygen and partly upon the rate at which oxygen is supplied and the surrounding conditions as they define the temperature.

Complete combustion is obtained when all of the combustible elements in the fuel are oxidized with all of the oxygen with which they can combine. All of the oxygen supplied may not be utilized.

Perfect combustion is defined as the result of supplying the required amount of oxygen for combination with all of the combustible elements of the fuel and utilizing all of the oxygen so supplied.

The oxygen required for the process of combustion is obtained from air which is a mechanical mixture of oxygen, nitrogen and small amounts of carbon dioxide, water vapor and inert gases. These inert gases are generally included with the nitrogen, and for engineering purposes the values given herewith may be used.

	By Volume Per Cent	By Weight Per Cent
Oxygen, O_2	20.9	23.15
Nitrogen, N_2	79 1	76.85

The combination of oxygen with the combustible elements and compounds of a fuel is in accordance with fixed laws. In the case of perfect combustion the reactions and resultant combinations are shown in Table 1.

The most important condition governing the process of combustion is temperature. It is necessary to bring a combustible substance to its ignition temperature before it will unite in chemical combination with oxygen to produce combustion. The ignition temperatures for several of the combustible constituents of fuels are presented in Table 1.

HEAT OF COMBUSTION

As previously stated, the process of combustion results in the evolution of heat. The heat generated by the complete combustion of a unit of fuel is constant for a given combination of combustible elements and compounds, and is known as the *heat of combustion*, *calorific value*, or *heating value* of the fuel.

Table 1. General Data of Combustible Elements and Compounds

Substance	Molecular Symbol	Chemical Reaction of Combustion	Ignition Temperature[a] Deg F	Calorific Value: Btu per Lb, Higher	Calorific Value: Btu per Lb, Lower	Calorific Value: Btu per Cubic Foot, Higher	Theoretical Oxygen and Air Requirements: Lb per Lb, O_2	Lb per Lb, Air	Cubic Ft per Cubic Ft, O_2	Cubic Ft per Cubic Ft, Air
Carbon (to CO)	—	$2C + O_2 = 2CO$	—	4380	—	—	1.333	5.76	—	—
Carbon (to CO_2)	—	$2C + 2O_2 = 2CO_2$	—	14540	—	—	2.667	11.52	—	—
Sulphur	S_2	—	—	—	—	—	1.000	4.32	—	—
Sulphur (to SO_2)	—	$S + O_2 = SO_2$	—	4050	—	—	—	—	—	—
Sulphur (to SO_3)	—	$2S + 3O_2 = 2SO_3$	—	5940	—	—	—	—	—	—
Carbon monoxide	CO	$2CO + O_2 = 2CO_2$	1166-1319	10160	—	342	0.572	2.46	0.5	2.391
Methane	CH_4	$CH_4 + 2O_2 = CO_2 + 2H_2O$	1202-1346	23850	21670	1073	4.000	17.28	2.0	9.564
Acetylene	C_2H_2	$2C_2H_2 + 5O_2 = 4CO_2 + 2H_2O$	763-824	21460	21020	1590	3.077	13.29	2.5	11.955
Ethylene	C_2H_4	$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$	986-1123	21450	20420	1675	3.429	14.81	3.0	14.346
Ethane	C_2H_6	$2C_2H_6 + 7O_2 = 4CO_2 + 6H_2O$	986-1123	22230	20500	1883	3.733	16.13	3.5	16.737
Hydrogen	H_2	$2H_2 + O_2 = 2H_2O$	1063-1166	62000	52920	348	8.000	34.56	0.5	2.391
Hydrogen sulphide	H_2S	$2H_2S + 3O_2 = 2H_2O + 2SO_2$	599-608	—	—	—	1.412	6.10	1.5	7.173

[a]From International Critical Tables, 1927

The heat of combustion of the several fuel elements and compounds in their *pure* state is given in Table 1.

The reaction of the carbon in the fuel with oxygen may result in the formation of carbon monoxide or carbon dioxide. In burning to carbon monoxide, the carbon is not completely oxidized and, as shown by the data, the heat produced is considerably less than if it were completely oxidized. This fact is of greatest importance in considering the efficiency of combustion.

The calorific value of a fuel is determined by direct measurement of the heat evolved during combustion in a calorimeter. Although the ash and moisture content of coal from a given mine or locality may vary widely, the heating value of the coal, on a *moisture and ash free* basis, remains relatively constant. It is therefore possible to approximate the heating value of a shipment of coal *as received* if its moisture and ash content are determined, and if the heating value of similar coal on a moisture and ash free basis is known. This may be calculated by Equation 1.

$$\text{Heating value, } \textit{as received} = \frac{\text{Heating value, } \textit{moisture and ash free} \times [100 - (\text{Moisture} + \text{Ash})]}{100} \quad (1)$$

where, moisture and ash are expressed in per cent.

Typical analyses of the coals of the United States are given in *U. S. Bureau of Mines Bulletin* 446.

As practically all fuels, including coal, oil, and gas, contain hydrogen, water vapor is one of the products of combustion. The amount of water vapor produced is proportional to the hydrogen content of the fuel.

When the heating value of a fuel is determined in a calorimeter the water vapor is condensed and the latent heat of vaporization is included in the heating value of the fuel. The heating value so determined is termed the *gross* or *higher* heating value and this is what is ordinarily meant when the heating value of a fuel is specified. In burning the fuel, however, the products of combustion are not cooled to the dew-point and the higher heating value cannot be obtained.

FLAME

The appearance of the flame or products of combustion may serve as an approximate measure of the temperatures developed in the combustion process. The luminosity of a flame is caused by the heating to incandescence of unconsumed particles of combustible matter in the gases, and the higher the temperature of these particles the whiter the flame. Table 2 gives some approximate flame temperature data.

TABLE 2. APPROXIMATE FLAME TEMPERATURE DATA

APPEARANCE OF FLAME	TEMPERATURE DEG F
Red, visible in daylight	1000
Light red	1800
Orange-red	2000
Orange-yellow	2200
Yellow-white	2400
Bright white	2600

AIR AND COMBUSTION

The weight of air required for the perfect combustion of a pound of fuel may be determined by use of the ultimate analysis of the fuel as applied to Equations 2 to 4. The various elements are expressed in percentages by weight.

Solid and Liquid Fuels:

$$\text{Pounds air required per pound fuel} = 34.56 \left[\frac{C}{3} + \left(H - \frac{O}{8}\right) + \frac{S}{8}\right] \tag{2}$$

Gaseous Fuels:

$$\text{Pounds air required per pound fuel} = 2.46\ CO + 34.56\ H_2 + 17.28\ CH_4 + 13.29\ C_2H_2 + 14.81\ C_2H_4 + 16.13\ C_2H_6 + 6.10\ H_2S - 4.32\ O_2 \tag{3}$$

When the analysis is given on a volumetric basis the equation is expressed as follows:

$$\text{Cubic feet air required per cubic foot gas} = 2.39\ (CO + H_2) + 9.56\ CH_4 + 11.98\ C_2H_2 + 14.35\ C_2H_4 + 16.74\ C_2H_6 - 4.78\ O_2 \tag{4}$$

Equations 5 and 6 may be used as approximate methods of determining the theoretical air requirement for any fuel.

$$\text{Pounds air required per pound fuel} = 0.755 \times \frac{\text{Heating value (Btu per pound)}}{1000} \tag{5}$$

$$\text{Cubic feet air required per unit fuel} = \frac{\text{Heating value (Btu per unit)}}{100} \tag{6}$$

Approximate values for the theoretical air required for different fuels are given in Table 3.

It is customary to make use of the analysis of the products of combustion to determine the amount of flue gas produced and the actual

TABLE 3. APPROXIMATE THEORETICAL AIR REQUIREMENTS

Solid Fuel	Pounds Air Per Pound Fuel
Anthracite	9.6
Semi-bituminous coal	11.2
Bituminous coal	10.3
Lignite	6.2
Coke	11.2

Fuel Oil	Pounds Air Per Gallon Fuel
Commercial Standard No. 1	102.6
Commercial Standard No. 2	104.5
Commercial Standard No. 3	106.5
Commercial Standard No. 5	112.0
Commercial Standard No. 6	114.2

Gaseous Fuels	Cubic Feet Air Per Cubic Foot Gas
Natural gas	10.0
Mixed, natural and water gas	4.4
Carbureted water gas	4.4
Water gas, coke	2.1
Coke oven gas	5.2

amount of air supplied for combustion. The analysis of flue gases has been well described in various publications of the *Bureau of Mines* and in the literature and the details of Orsat manipulation need not be considered in this discussion. (See Chapter 34.)

The weight of dry flue gas per pound of fuel burned is used in combustion loss calculations and may be determined by Equation 7.

$$\text{Pounds dry flue gas per pound fuel} = \frac{11\ CO_2 + 8\ O_2 + 7\ (CO + N_2)}{3\ (CO_2 + CO)} \times C \qquad (7)$$

Values for CO_2, O_2, CO and N_2 are percentages by volume from the flue gas analysis and C is the weight of carbon burned per pound of fuel corrected for carbon in the ash.

EXCESS AIR

Since one measure of the efficiency of combustion is the relation existing between the amount of air theoretically required for *perfect* combustion and the amount of air actually supplied, a method of determining the latter factor is of value. Equation 8 will give reasonably accurate results, for most solid and liquid fuels, for determining the amount of air supplied per pound of fuel.

$$\text{Pounds dry air supplied per pound of fuel} = \frac{3.04\ N_2}{(CO_2 + CO)} \times C \qquad (8)$$

Values for CO_2, CO and N are percentages by volume from the flue gas analysis and C is the weight of carbon burned per pound of fuel corrected for carbon in the ash.

The difference between the air actually supplied for combustion and the theoretical air required is known as excess air.

$$\text{Per cent excess air} = \frac{\text{Air supplied} - \text{Theoretical air}}{\text{Theoretical air}} \qquad (9)$$

Since the calculation is usually made from Orsat readings, Equation 10 will be found to be a convenient statement of this relationship.

$$\text{Per cent excess air} = \frac{100\left(O_2 - \frac{CO}{2}\right)}{N_2 \times 0.264 - \left(O_2 - \frac{CO}{2}\right)} \qquad (10)$$

In this formula the symbols represent volumetric percentages of the flue gas constituents as determined by analysis.

The amount of excess air in its relation to the percentage of CO_2 is shown by the curves in Fig. 1 for several fuels. These are approximate values. It should be noted that in hand-fired furnaces with long periods between firings the combustion goes through a cycle in each period and the quantity of excess air present varies.

Due to the different carbon-hydrogen ratios of the different fuels the maximum CO_2 attainable varies. Representative values for perfect combustion of several fuels are given in Table 4.

In considering the factor of excess air it should be noted that a deficiency of air supply will result in combustible products passing to the stack unburned. An excess of air absorbs heat from the products of combustion and results in a greater loss of sensible heat to the stack. An excess of air is always required, however, to eliminate combustible losses occasioned

by poor mixing of the fuel and air. It is considered good practice, under usual operating conditions, to supply from 25 to 50 per cent excess air, dependent upon the fuel utilized.

SECONDARY AIR

When bituminous coal is hand-fired in a furnace the volatile matter in the fuel distills off leaving coke on the grate. The product of combustion of the coke is CO_2 and under certain conditions some CO may arise from the bed. The combustion of the volatile matter and the CO may amount

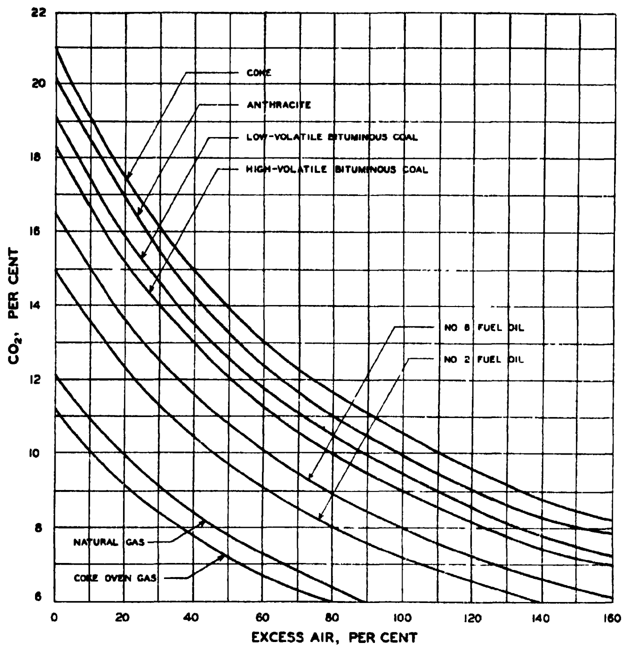


FIG. 1. RELATION BETWEEN CO_2 AND EXCESS AIR IN GASES OF COMBUSTION

to the liberation of from 40 to 60 per cent of the heat in the fuel in the combustion space over the fuel bed.

The air that passes through the fuel bed is called *primary air* and the air that is admitted over the fuel bed in order to burn the volatile matter and CO is called *secondary air.*

TABLE 4. REPRESENTATIVE MAXIMUM CO_2 VALUE

Fuel	Theoretical CO_2	CO_2 Usually Attained in Practice
Coke	21.00	12-14
Anthracite	20 20	12-14
Bituminous Coal	18 20	13
No 2 Fuel Oil	15 00	10.5
No. 6 Fuel Oil	16.50	13.5
Natural Gas	12 00	9 7
Coke Oven Gas	11.00	8.5

This process of combustion is illustrated in Fig. 2[1]. The free oxygen of the air passes through the grate and the ash above it and burns the carbon in the lower 3 or 4 in. of the fuel bed forming carbon dioxide. This layer noted as the oxidizing zone is indicated by the symbols CO_2 and O_2. Some of the carbon dioxide of the oxidizing zone is reduced to carbon monoxide in the upper layer of the fuel bed noted as the reducing zone and indicated by the symbols CO_2 and CO. The gases leaving the fuel bed are mainly carbon monoxide, carbon dioxide, nitrogen and very little free oxygen. Free oxygen is admitted through the firing door in an attempt to burn carbon monoxide and the volatile combustible distilled from the freshly fired fuel.

The division of the total into primary and secondary air necessary to produce the same rate of burning and the same excess air depends on a number of factors which include size and type of fuel, depth of fuel bed, and size of fire-pot.

Size of the fuel is a very important factor in fixing the quantity of

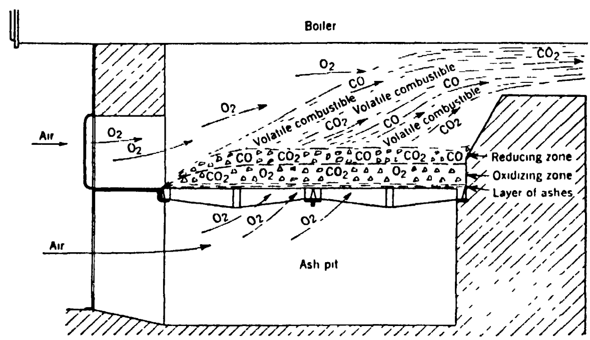


FIG 2. COMBUSTION OF FUEL IN A HAND-FIRED FURNACE

secondary air required for non-caking coals. With caking coals it is not so important because small pieces fuse together and form large lumps. Fortunately a smaller size fuel gives more resistance to air flow through the fuel bed and thus automatically causes a larger draft above the fuel bed, which draws in more secondary air through the same slot openings. In spite of this, a small size fuel requires a larger opening of the door slots; for a certain size for each fuel no slot opening is required, and for larger sizes too much excess air gets through the fuel bed.

It is impossible to establish a single rule for the correct slot opening for all types and sizes of fuels and for all rates of burning. Furthermore, the effect of slot opening is dependent on whether the ashpit damper is open or closed.

Bituminous coals require a large amount of secondary air during the period subsequent to a firing in order to consume the gases and to reduce the smoke. The smoke produced is a good indicator, and that opening is best which reduces the smoke to a minimum. Too much secondary air will cool the gases below the ignition point, and prove harmful instead of beneficial.

Secondary air that enters the combustion chamber too far removed from the zone of combustion will also be harmful, for the oxygen in the

[1]Hand Firing Soft Coal Under Power Plant Boilers (*U. S Bureau of Mines Technical Paper* No 80).

secondary air will not react with any unburned gases unless the mixture is subjected to high temperatures.

The air requirements of oil and gas burners are discussed in Chapter 10, Automatic Fuel Burning Equipment.

DRAFT REQUIREMENTS

The draft required to effect a given rate of burning the fuel is dependent on the following factors:

1. Kind and size of fuel.
2. Grate area.
3. Thickness of fuel bed.
4. Type and amount of ash and clinker accumulation.
5. Amount of excess air present in the gases.
6. Resistance offered by the boiler passes to the flow of the gases.
7. Accumulation of soot in the passes.

Insufficient draft will necessitate additional manipulation of the fuel bed and more frequent cleanings to keep its resistance down. Insufficient draft also restricts the control that can be accomplished by adjustment of the dampers.

The quantity of excess air present has a marked effect on the draft required to produce a given rate of burning. If the excess is caused by holes in the fuel bed, or an extremely thin fuel bed, it is often possible to produce a higher rate of burning by increasing the thickness of the bed. The thickness of the fuel bed should not, however, be increased too much because the increased draft resistance will reduce the rate of primary air supply and the rate of burning.

For amount of draft required see Chapter 9, Chimneys and Draft Calculations.

DRAFT REGULATION

Because of the varying heating load demands present in most installations it is necessary to vary the rate of fuel burning. The maintenance of the proper air supply for the various rates of burning is accomplished by regulation of the drafts. Correct and incorrect methods of draft regulation are shown in Fig. 3. The air enters through the ashpit draft door, firing door and by leaks in the setting, whereas the gases leave only through the uptake. By throttling the gases with the damper in the

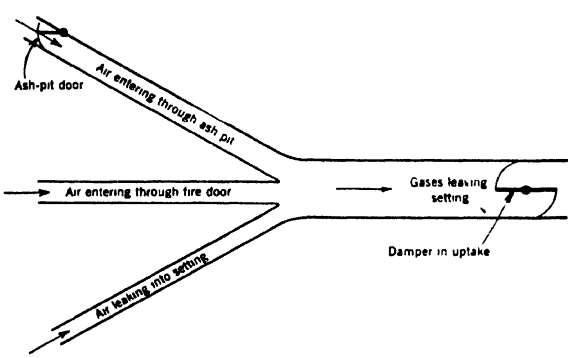


FIG. 3. CORRECT AND INCORRECT METHODS OF DRAFT REGULATION IN A HAND-FIRED FURNACE

uptake all the air entering by each of the three intakes is reduced in the same proportion. If the ashpit draft door is closed the air admitted through the ashpit is reduced while it is increased through the other two intake openings.

Methods of control of draft conditions when burning oil or gas are noted in Chapter 10, Automatic Fuel Burning Equipment.

CLASSIFICATION OF COALS

The complex composition of coal makes it difficult to classify it into clear-cut types. Its chemical composition is some indication but coals having the same chemical analysis may have distinctly different burning characteristics. Users are mainly interested in the available heat per pound of coal, in the handling and storing properties, and in the burning characteristics. A description of the relationship between the qualities of coals and these characteristics requires considerable space: a treatment applicable to heating boilers is given in a Bureau of Mines Bulletin[2].

Coal composition may be expressed by either an *ultimate* or *proximate* analysis. In the ultimate analysis the proportions of carbon, hydrogen, oxygen, nitrogen, sulphur, and ash are determined. This form of analysis is difficult to make and is used only for extremely close studies. The proximate analysis is more easily made and is satisfactory for most purposes. In this analysis, the proportions of moisture, volatile matter, fixed carbon, and ash are determined. Moisture is obtained by noting the loss of weight of a sample of coal when dried at about 220 F. To determine volatile matter, the dried sample is heated to about 1750 F in a closed crucible, and the loss of weight is noted. The sample is then burned in an open crucible, and the accompanying loss of weight represents the fixed carbon. The unburned residue is ash. Although determined separately, the sulphur content is frequently reported with a proximate analysis.

Other important qualities of coals are the screen sizes, ash fusion temperature, friability, caking tendency, and the qualities of the volatile matter. In considering these factors the following points are of interest. The volatile products given off by coals when they are heated differ materially in the ratios by weight of the gases to the oils and tars. No heavy oils or tars are given off by anthracite, and very small quantities are given off by semi-anthracite. As the volatile matter in the coal increases to as much as 40 per cent of ash and moisture-free coal, increasing amounts of oils and tars are released. For coals of higher volatile content, the relative quantity of oils and tars decreases, so it is low in the sub-bituminous coals and in lignite. The percentage of ash and its fusion temperature do not indicate the composition or distribution of its constituents.

A classification of coals is given in Table 5, and a brief description of the kinds of fuel is given in the following paragraphs, but it should be recognized that there are no distinct lines of demarcation between the kinds, and that they graduate into each other.

Anthracite is a clean, dense, hard coal which creates little dust in handling. It is comparatively hard to ignite but it burns freely when well started. It is non-caking, it burns uniformly and smokelessly with a short flame, and it requires no attention to the fuel bed between firings. It is capable of giving a high efficiency in the common types of hand-

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[2]Five Hundred Tests of Various Coals in Househeating Boilers (*U. S Bureau of Mines Bulletin* No 276).

TABLE 5. CLASSIFICATION OF COALS BY RANK[a]

Legend: F.C. = Fixed Carbon. V.M. = Volatile Matter. Btu = British thermal units.

CLASS	GROUP	LIMITS OF FIXED CARBON OR BTU MINERAL-MATTER-FREE BASIS	REQUISITE PHYSICAL PROPERTIES
I. Anthracite	1. Meta-anthracite	Dry F.C., 98 per cent or more (Dry V.M., 2 per cent or less)	Non-agglomerating[b]
	2. Anthracite	Dry F.C., 92 per cent or more and less than 98 per cent (Dry V.M., 8 per cent or less and more than 2 per cent)	
	3. Semi-anthracite	Dry F.C., 86 per cent or more and less than 92 per cent (Dry V.M., 14 per cent or less and more than 8 per cent)	
II. Bituminous[d]	1. Low volatile bituminous coal	Dry F.C., 78 per cent or more and less than 86 per cent (Dry V.M., 22 per cent or less and more than 14 per cent)	Either agglomerating[b] or non-weathering[f]
	2. Medium volatile bituminous coal	Dry F.C., 69 per cent or more and less than 78 per cent (Dry V.M., 31 per cent or less and more than 22 per cent)	
	3. High volatile *A* bituminous coal	Dry F.C., less than 69 per cent (Dry V.M., more than 31 per cent), and moist[c] Btu, 14,000[e] or more	
	4. High volatile *B* bituminous coal	Moist[c] Btu, 13,000 or more and less than 14,000[e]	
	5. High volatile *C* bituminous Coal	Moist Btu, 11,000 or more and less than 13,000[e]	
III. Sub-bituminous	1. Sub-bituminous *A* coal	Moist Btu, 11,000 or more and less than 13,000[e]	Both weathering and non-agglomerating[b]
	2. Sub-bituminous *B* coal	Moist Btu, 9500 or more and less than 11,000[e]	
	3. Sub-bituminous *C* coal	Moist Btu, 8300 or more and less than 9500[e]	
IV. Lignitic	1. Lignite	Moist Btu less than 8300	Consolidated
	2. Brown coal	Moist Btu less than 8300	Unconsolidated

[a]This classification does not include a few coals which have unusual physical and chemical properties and which come within the limits of fixed carbon or Btu of the high-volatile bituminous and sub-bituminous ranks. All of these coals either contain less than 48 per cent dry, mineral-matter-free fixed carbon or have more than 15,500 moist, mineral-matter-free Btu.

[b]If agglomerating, classify in low-volatile group of the bituminous class.

[c]Moist Btu refers to coal containing its natural bed moisture but not including visible water on the surface of the coal.

[d]It is recognized that there may be non-caking varieties in each group of the bituminous class.

[e]Coals having 69 per cent or more fixed carbon on the dry, mineral-matter-free basis shall be classified according to fixed carbon, regardless of Btu.

[f]There are three varieties of coal in the high-volatile *C* bituminous coal group, namely, Variety 1, agglomerating and non-weathering; Variety 2, agglomerating and weathering; Variety 3, non-agglomerating and non-weathering.

Adapted from *A.S.T.M.* Standards, 1937, Supplement, p. 145, *American Society for Testing Materials*, Philadelphia.

fired furnaces. A tabulation of the quality of the various anthracite sizes will be found in a Bureau of Mines Report[3].

Semi-anthracite has a higher volatile content than anthracite. It is not so hard and ignites somewhat more easily; otherwise its properties are similar to those of anthracite.

Semi-bituminous coal is soft and friable, and fines and dust are created by handling it. It ignites somewhat slowly and burns with a medium length of flame. Its caking properties increase as the volatile matter increases, but the coke formed is relatively weak. Having only half the volatile matter content of the more abundant bituminous coals it can be burned with less production of smoke, and it is sometimes called *smokeless coal.*

The term *bituminous coal* covers a large range of coals and includes many types having distinctly different composition, properties, and burning characteristics. The coals range from the high-grade bituminous coals of the East to the poorer coals of the West. Their caking properties range from coals which completely melt, to those from which the

[3]Quality of Anthracite as Prepared at Breakers, 1935 (*U. S. Bureau of Mines Report of Investigations*, R. I. 3283).

volatiles and tars are distilled without change of form, so that they are classed as non-caking or free-burning. Most bituminous coals are strong and non-friable enough to permit of the screened sizes being delivered free from fines. In general, they ignite easily and burn freely; the length of flame varies with different coals, but it is long. Much smoke and soot are possible, if improperly fired, especially at low rates of burning.

Sub-bituminous coals occur in the western states; they are high in moisture when mined and tend to break up as they dry or when exposed to the weather; they are liable to ignite spontaneously when piled or stored. They ignite easily and quickly and have a medium length flame, are non-caking and free-burning; the lumps tend to break into small pieces if poked; very little smoke and soot are formed.

Lignite is of woody structure, very high in moisture as mined, and of low heating value; it is clean to handle. It has a greater tendency than the sub-bituminous coals to disintegrate as it dries, and it also is more liable to spontaneous ignition. Freshly mined lignite, because of its high moisture, ignites slowly. It is non-caking. The char left after the moisture and volatile matter are driven off burns very easily, like charcoal. The lumps tend to break up in the fuel bed and pieces of char falling into the ashpit continue to burn. Very little smoke or soot is formed.

CLASSIFICATION OF COKES

Coke is produced by the distillation of the volatile matter from coal. The type of coke depends on the coal or mixture of coals used, the temperatures and time of distillation and, to some extent, on the type of retort or oven; coke is also produced as a residue from the destructive distillation of oil

High-temperature cokes. Coke as usually available is of the high-temperature type, and contains between 1 and 2 per cent volatile matter. High-temperature cokes are subdivided into ***beehive coke*** of which comparatively little is now sold for domestic use, ***by-product coke***, which covers the greater part of the coke sold, and ***gas-house coke***. The differences among these three cokes are relatively small; their denseness and hardness decrease and friability increases in the order named In general, the lighter and more friable cokes ignite and burn the more easily.

Low-temperature cokes are produced at low coking temperatures, and only a portion of the volatile matter is distilled off. Cokes as made by various processes under development have contained from 10 to 15 per cent volatile matter. In general, these cokes ignite and burn more readily than high-temperature cokes. The properties of various low-temperature cokes may differ more than those of the various high-temperature cokes because of the differences in the quantities of volatile matter and because some may be light and others briquetted.

Petroleum cokes, which are obtained by coking the residue left from the distillation of petroleum, vary in the amount of volatile matter they contain, but all have the common property of a very low ash content, which necessitates the use of refractory pieces to protect the grates from being burned

FIRING METHODS FOR ANTHRACITE

An anthracite fire should never be poked or disturbed, as this serves to bring ash to the surface of the fuel bed where it may melt into clinker.

Egg size is suitable for large fire-pots (grates 24 in. and over) if the fuel can be fired at least 16 in. deep. For best results this coal should be fired deeply.

Stove size coal is the proper size of anthracite for many boilers and furnaces used for heating buildings. It burns well on grates at least 16 in. in diameter and 12 in. deep. The fuel should be fired deeply and uniformly.

Chestnut size coal is in demand for fire-pots up to 20 in. in diameter, with a depth of from 10 to 15 in.

Pea size coal is often an economical fuel to burn. When fired carefully, pea coal can be burned on standard grates. Care should be taken to shake the grates only until the first bright coals begin to fall through the grates. The fuel bed, after a new fire has been built, should be increased in thickness by the addition of small charges until it is at least level with the sill of the fire-door. A satisfactory method of firing pea coal consists

of drawing the red coals toward the front end and piling fresh fuel toward the back of the fire-box.

Pea size coal requires a strong draft and therefore the best results generally will be obtained by keeping the choke damper open and regulating solely by means of the cold air check and the air inlet damper.

Buckwheat size coal for best results requires more attention than pea size coal, and in addition the smaller size of the fuel makes it more difficult to burn on ordinary grates. Greater care must be taken in shaking the grates than with the pea coal on account of the danger of the fuel falling through the grate. In house heating furnaces the coal should be fired lightly and more frequently than pea coal. When banking a buckwheat coal fire it is advisable after coaling to expose a small spot of hot fire by putting a straight poker down through the bed of fresh coal. This will serve to ignite the gas that will be distilled from the fresh coal and prevent delayed ignition within the fire-pot, which in some cases, depending upon the thickness of the bed of fresh coal, is severe enough to blow open the doors and dampers of the furnace. Where frequent attention can be given and care exercised in manipulation of the grates this fuel can be burned satisfactorily without the aid of any special equipment.

In general it will be found more satisfactory with buckwheat coal to maintain a uniform heat output and consequently to keep the system warm all the time, rather than to allow the system to cool off at times and then to attempt to burn the fuel at a high rate while warming up. A uniform low fire will minimize the clinker formation and keep the clinker in an easily broken up condition so that it readily can be shaken through the grate.

Forced draft and small mesh grates are frequently used for burning buckwheat anthracite. For greater convenience, domestic stokers are used.

Buckwheat anthracite No. 2, or rice size, is used principally in stokers of the domestic, commercial and industrial type. No. 3 buckwheat anthracite, or barley, has no application in domestic heating.

Standard anthracite sizing specifications are shown in Table 6[4]

TABLE 6. STANDARD ANTHRACITE SIZING SPECIFICATIONS

Size	Test Mesh, In.		Round Mesh			Maximum Impurities, Per Cent	
			Oversize	Undersize			
	Through	Over	Max Per Cent	Max Per Cent	Min Per Cent	Slate[a]	Bone
Broken	4 3/8	3 1/4	....	15	7 1/2	1 1/2	2
Egg	3 1/4 to 3	2 7/16	5	15	7 1/2	1 1/2	2
Stove	2 7/16	1 5/8	7 1/2	12 1/2	7 1/2	2	3
Nut	1 5/8	13/16	7 1/2	10	5	3	4
Pea	13/16	9/16	10	15	7 1/2	4	5
Buckwheat	9/16	5/16	10	15	7 1/2	12 Ash	
Rice	5/16	3/16	10	15	7 1/2	13 Ash	
Barley	3/16	3/32	10	20	10	....	

[a]When slate content on Broken to Pea inclusive is less than above standards, bone content may be correspondingly increased, but slate content specified above shall not be exceeded in any event and the total maximum impurities shall not exceed those above specified

[4]Approved and adopted by *Anthracite Committee*, State Street Building, Harrisburg, Pa.

FIRING METHODS FOR BITUMINOUS COAL

A commonly recommended procedure for firing domestic heating units, called the side-bank method, requires the movement of live coals to one side or the back of the grate, and placing the fresh fuel charge on the opposite side. The results are a more uniform release of volatile gases, and the subjection of these gases to the high temperature of the red coals. If the fresh charge is covered with a layer of fine coal, still better results may be obtained because of slower release of volatile matter.

Bituminous coal should never be fired over the entire fuel bed at one time. A portion of the glowing fuel should always be left exposed to ignite the gases leaving the fresh charge.

The importance of firing bituminous coal in small quantities at short intervals is discussed in a U. S. Bureau of Mines technical paper[5]. Better combustion is obtained by this method in that the fuel supply is maintained more nearly proportional to the air supply.

If the coal is of the caking kind the fresh charge will fuse into one solid mass which can be broken up with the stoking bar and leveled from 20 min to one hour after firing, depending on the temperature of the fire-box. Care should be exercised when stoking not to bring the bar up to the surface of the fuel as this will tend to bring ash into the high temperature zone at the top of the fire, where it will melt and form clinker. The stoking bar should be kept as near the grate as possible and should be raised only enough to break up the fuel. With fuels requiring stoking it may not be necessary to shake the grates, as the ash is usually dislodged during stoking.

It is acknowledged that it may be difficult to apply the outlined methods to domestic heating boilers of small size, especially when frequent attendance is impractical. The adherence to these methods insofar as practical, however, will result in better combustion.

The output obtained from any heater with bituminous coal will usually exceed that obtained with anthracite, since bituminous coal burns more rapidly than anthracite and with less draft. Bituminous coal, however, will usually require frequent attention to the fuel bed.

FIRING METHODS FOR SEMI-BITUMINOUS COAL

The *Pocahontas Operators' Association* recommends the central cone method of firing, in which the coal is heaped on to the center of the bed forming a cone, the top of which should be level with the middle of the firing door. This allows the larger lumps to fall to the sides, and the fines to remain in the center and be coked. The poking should be limited to breaking down the coke without stirring, and to gently rocking the grates. It is recommended that the slides in the firing door be kept closed, as the thinner fuel bed around the sides allows enough air to get through.

FIRING METHODS FOR COKE

Coke ignites less readily than bituminous coal and more readily than anthracite and burns rapidly with little draft. In order to control the air admitted to the fuel it is very important that all openings or leaks into the ashpit be closed tightly. A coke fire responds rapidly to the opening of the dampers. This is an advantage in warming up the system, but it also makes it necessary to watch the dampers more closely in order to

[5]Loc. Cit. Note 1.

prevent the fire from burning too rapidly. In order to obtain the same interval of attention as with other fuels a deep fuel bed always should be maintained when burning coke. The grates should be shaken only slightly in mild weather and should be shaken only until the first red particles drop from the grates in cold weather. The best size of coke for general use, for small fire-pots where the fuel depth is not over 20 in., is that which passes over a 1 in. screen and through a 1½ in. screen. For large fire-pots where the fuel can be fired over 20 in. deep, coke which passes over a 1 in. screen and through a 3 in. screen can be used, but a coke of uniform size is always more satisfactory. Large sizes of coke should be either mixed with fine sizes or broken up before using.

FURNACE VOLUME

The principal requirements for a *hand-fired furnace* are that it shall have enough grate area and correctly proportioned combustion space. The amount of grate area required is dependent upon the desired combustion rate.

The furnace volume is influenced by the kind of coal used. Bituminous coals, on account of their long-flaming characteristic, require more space in which to burn the gases of combustion completely than do the coals low in volatile matter. For burning high volatile coals provision should be made for mixing the combustible gases thoroughly, so that combustion is complete before the gases come in contact with the relatively cool heating surfaces. An abrupt change in the direction of flow tends to mix the gases of combustion more thoroughly. Anthracite requires comparatively little combustion space.

CLASSIFICATION OF OILS

The Commercial Standard Specifications for Fuel Oils (CS 12-40) of the *U. S. Department of Commerce* are given in Table 7. These specifications conform with *American Society for Testing Materials* Tentative Specifications for Fuel Oils D396-38T.

The specific gravity of oil is of interest in its relationship to the calorific value and these data are given in Table 8.

COMBUSTION OF OIL

With oil, as with any kind of fuel, efficient heat production requires that all combustible matter in the fuel shall be completely consumed and that it shall be done with a minimum of excess air. The combustion of oil is a rather rapid chemical reaction. Excess air provides an over supply of oxygen so that all of the oil will be completely oxidized and thus produce all the heat possible. The use of unreasonable quantities of air in excess of theoretical combustion requirements results in lowered efficiencies due to increased stack losses. Such losses, if not accompanied by unburned products of combustion, may be offset somewhat by increasing the secondary heating surfaces of the heat absorbing medium, boiler or furnace.

Oil is a highly concentrated fuel composed mainly of hydrogen and carbon. In its liquid form oil cannot burn. It must be converted into a gas or vapor by some means. If the excess air is to be kept within efficient limits, air must be supplied in carefully regulated quantities. The air and oil vapor must be thoroughly mixed to get a rapid and complete chemical reaction. The better the mixing, the less excess air will be

TABLE 7. DETAILED REQUIREMENTS FOR FUEL OILS[a]

Grade[b]	Flash Point Deg F		Pour Point Deg F	Water and Sediment Per Cent	Carbon Residue Per Cent	Ash Per Cent	Distillation Temperatures Deg F: 10 Per Cent Point	Distillation Temperatures Deg F: 90 Per Cent Point		Distillation Temperatures Deg F: End Point	Viscosity Seconds: Saybolt Universal at 100 F		Viscosity Seconds: Saybolt Furol at 122 F	
	Min.	Max.	Max	Max	Max	Max	Max.	Max	Min.	Max.	Max.	Min.	Max	Min.
No. 1 Fuel oil—a distillate oil for use in burners requiring a volatile fuel.	100 or Legal	165	0[c]	Trace	0.05 on 10% Residuum[d]		410			560[e]				
No. 2 Fuel oil—a distillate oil for use in burners requiring a moderately volatile fuel.	110 or Legal	190	10[c]	0.05	0.25 on 10% Residuum[f]		440	600						
No. 3 Fuel oil—a distillate oil for use in burners requiring a low viscosity fuel.	110 or Legal	230	20[c]	0.10	0.15 Straight			675	600[g]		45			
No. 5 Fuel oil—an oil for use in burners requiring a medium viscosity fuel.	130 or Legal			1.00		0.10						50	40	
No. 6 Fuel oil—an oil for use in burners equipped with preheaters permitting a high viscosity fuel.	150			2.00[h]									300	45

[a]Recognizing the necessity for low sulphur fuel oils used in connection with heat-treatment, non-ferrous metal, glass and ceramic furnaces and other special uses, a sulphur requirement may be specified in accordance with the following table

Grade of Fuel Oil	Sulphur, Max. Per cent
No. 1	0.5
No. 2	0.5
No. 3	0.75
No. 5	No Limit
No 6	No Limit

Other sulphur limits may be specified only by mutual agreement between the buyer and seller.

[b]It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

[c]Lower or higher pour points may be specified whenever required by conditions of storage or use. However, these specifications shall not require a pour point lower than 0 F under any conditions

[d]For use in other than sleeve type blue flame burners carbon residue on 10 per cent residuum may be increased to a maximum of 0 12 per cent. This limit may be specified by mutual agreement between the buyer and seller.

[e]The maximum end point may be increased to 590 F when used in burners other than sleeve type blue flame burners

[f]To meet certain burner requirements the carbon residue limit may be reduced to 0.15 per cent on 10 per cent residuum.

[g]The minimum distillation temperature of 600 F for 90 per cent may be waived if A.P.I gravity is 26 or lower.

[h]Water by distillation, plus sediment by extraction. Sum, maximum 2 0 per cent The maximum sediment by extraction shall not exceed 0.50 per cent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 per cent.

TABLE 8. APPROXIMATE GRAVITY AND CALORIFIC VALUE OF STANDARD GRADES OF FUEL OIL

Commercial Standard No.	Approximate Gravity Range A. P. I.	Calorific Value Btu Per Gallon
1	38-40	136,000
2	34-36	138,500
3	28-32	141,000
5	18-22	148,500
6	14-16	152,000

needed. The combustion must take place in a space that maintains the temperatures high so the reaction will be completed.

CLASSIFICATION OF GAS

Gas is broadly classified as being either *natural* or *manufactured*. Natural gas is a mechanical mixture of several combustible and inert gases rather than a chemical compound. Manufactured gas as distributed is usually a combination of certain proportions of gases produced by two or more processes. Representative properties of gaseous fuels commonly used in domestic heating are presented in Table 9.

Natural gas is the richest of the gases and contains from 80 to 95 per cent methane, with small percentages of the other combustible hydrocarbons. In addition, it contains from 0.5 to 5.0 per cent of CO_2, and from 1 to 12 or 14 per cent of nitrogen. The heat value varies from 1000 to 1200 Btu per cubic foot, the majority of natural gases averaging about 1000 Btu per cubic foot. Table 9 shows typical values for the four main oil fields, although values from any one field vary materially.

Table 9 also gives the calorific values of the more common types of manufactured gas. Most states have legislation which controls the distribution of gas and fixes a minimum limit to its heat content. The gross or higher calorific value usually ranges between 520 and 545 Btu per cubic foot, with an average of 535. A given heat value may be maintained and yet leave considerable latitude in the composition of the gas so that as distributed the composition is not necessarily the same in different districts, nor at successive times in the same district. However, in any community the variations in gas composition are held within suitable limits so that the performance of approved gas appliances will not be adversely affected.

COMBUSTION OF GAS

The majority of gas burners utilized in central domestic heating plants are of the Bunsen type and operate with a non-luminous flame. In this type of burner part of the air required for combustion is mixed with the gas as primary air, the air and gas mixture being fed to the burner ports. Additional secondary air is introduced around the flame by draft inspiration. In the luminous flame burner, which is sometimes used, all of the air for combustion is brought in contact with the flame as secondary air. This secondary air should be brought into intimate contact with the gas.

Some makes of burners use radiants or refractories to convert some of the energy in the gas to radiant heat. The radiants also serve as baffles in directing the flow of the products of combustion.

The quantity of air given in Table 9 is that required for theoretical combustion, but with a properly designed and installed burner the excess

TABLE 9. REPRESENTATIVE PROPERTIES OF GASEOUS FUELS BASED ON GAS AT 60 F AND 30 IN. HG.

Gas	Btu per Cu Ft		Specific Gravity, Air = 1.00	Air Required for Combustion, (Cu Ft)	Products of Combustion				Theoretical Flame Temperature, (deg F)
					Cubic Feet			Ultimate CO_2 Dry Basis	
	High (Gross)	Low (Net)			CO_2	H_2O	Total with N_2		
Natural gas—California	1200	1085	0.67	11.26	1.24	2.24	12.4	12.2	3610
Natural gas—Mid-Continental	970	870	0.57	9.17	0.97	1.92	10.2	11.7	3580
Natural gas—Ohio	1130	1025	0.65	10.70	1.17	2.16	11.8	12.1	3600
Natural gas—Pennsylvania	1130	1025	0.71	11.70	1.30	2.29	12.9	12.3	3620
Retort coal gas	570	510	0.42	5.00	0.50	1.21	5.7	11.2	3665
Coke oven gas	590	520	0.42	5.19	0.51	1.25	5.9	11.0	3660
Carbureted water gas	540	495	0.65	4.37	0.74	0.75	5.0	17.2	3815
Blue water gas	300	280	0.53	2.26	0.46	0.51	2.8	22.3	3800
Anthracite producer gas	135	125	0.85	1.05	0.33	0.19	1.9	19.0	3000
Bituminous producer gas	150	140	0.86	1.24	0.35	0.19	2.0	19.0	3160
Oil gas	575	510	0.35	4.91	0.47	1.21	5.6	10.7	3725

air can be kept low. In order to insure freedom from carbon monoxide under conditions which may obtain in installations, it is customary to design gas burning appliances for a supply of 30 to 35 per cent of excess air. In individual installations in which flue gas analyses are made, the excess air is sometimes reduced to approximately 20 per cent. The division of the air into primary and secondary is a matter of burner design, the pressure of gas available, and the type of flame desired.

The air gas ratio has a decided effect upon flame propagation. It is necessary that the gas will flow out of the burner ports fast enough so that the flame cannot travel back into the burner head, i.e. *flash back*, but the velocity must not be so high that it blows the flame away from the port.

The maximum and minimum flow speeds from burner ports which may be permitted are known to be very close together when air-gas mixtures in theoretical proportions are being supplied to the burner. As the air-gas ratio is lowered, and the mixture becomes more *gas rich*, the limiting speeds become farther apart, until with 100 per cent gas, in an all-yellow flame, flash back cannot occur and a much higher velocity is needed to blow off the flames.

SOOT

The deposit of soot on the flue surfaces of a boiler or heater acts as an insulating layer over the surface and reduces the heat transmission to the

water or air. The *Bureau of Mines Report of Investigations* No. 3272[6] shows that the loss of seasonal efficiency is not so great as has been believed and usually is not over 6 per cent because the greater part of the heat is transmitted through the combustion chamber surfaces. The *Bureau of Standards Report* BMS 54[7] points out that, although the decrease in efficiency of an oil fired boiler due to soot deposits is relatively small the attendant increase in stack temperature may be considerable.

The soot accumulation clogs the flues, reduces the draft, and may prevent proper combustion.

CONDENSATION AND CORROSION

Sulphur dioxide or trioxide formed by the combustion of fuels is the corroding element in flue gases and becomes active whenever moisture is available for the formation of sulphurous or sulphuric acid. It is therefore necessary to maintain a flue gas temperature above 175 F in all parts of appliances, and consequently it is not practicable to recover the latent heat in flue gases. Since some unpreventable condensation occurs during the warming-up period, it is important to design appliances so that all surfaces reach quickly a temperature above the dew-point of the flue gases or have corrosion resistant properties enabling them to withstand the corrosive effect.

DUSTLESS TREATMENT OF COAL

The practice of treating the more friable coals to allay the dust they create is increasing. The coal is sprayed with various petroleum products, a solution of calcium chloride or a mixture of calcium and magnesium chlorides.

The coal is usually treated at the mine, but sometimes by the local distributor just before delivery. The salt solutions are sprayed under high pressure, using from 2 to 4 gal or from 5 to 10 lb of the salt per ton of coal, depending on its friability and size. Oil for the dustless treatment of coal is also applied under high pressure, in concentrations of 1 to 8 qt per ton of coal, depending upon the characteristics of the coal and oil.

Dustless treatments which are of such a corrosive nature that they may damage coal handling or burning equipment should not be used.

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CHAPTER 9

Chimneys and Draft Calculations

Natural Draft, Mechanical Draft, Draft Control, Characteristics of Natural Draft Chimneys, Determining Chimney Sizes, General Equation, Domestic Chimneys, Chimneys for Gas Heating, Construction Details

A DRAFT, in the older sense, is a current of air and the draft of a furnace or boiler is that air current which flows through the fire-box and furnishes the oxygen for combustion. In engineering, however, the word *draft* has come to mean that pressure difference which causes this air current to flow and the word will be used in this sense in this chapter.

Draft is usually measured in inches of water and it is proper to speak of the draft in the fire-box or in the smoke breeching, etc., meaning the difference in pressure between the gases within and the air without those parts of a system. Draft is called positive when the pressure within such a part is less than that outside.

Draft is classified as natural and mechanical, depending on whether it is produced by a chimney or by a blower, and mechanical draft is further classified as induced or forced, depending on whether the air is drawn through or forced through the combustion chamber.

Chimneys can serve both to create a draft and to dispose of combustion products at a desirable height. For the latter purpose, chimneys, stacks, or, in the case of ships, funnels, are used in conjunction with mechanical-draft systems.

THEORETICAL DRAFT

If the air in one of two equal chimneys is heated while that in the other is not, the air in the heated chimney will be less heavy than that in the other chimney and a manometer or other pressure gage connecting the two at the bottom will indicate a pressure difference, called natural draft. The pressure of the air at the tops of the two chimneys will be equal, so that the pressure difference between them at the bottom will depend only on their height and the difference in density of the air they contain. The density of the air in either chimney is inversely proportional to its absolute temperature, so that the difference in pressure between them at the bottom will be proportional to their height and to the difference between the reciprocals of the absolute temperatures within them.

The pressure at the bottom of an unheated (and uncooled) chimney will be the same as that of the air outside, so that the unheated chimney can be dropped from the foregoing illustration. The manometer reading will be the same if its free connection is left open to the atmosphere.

These considerations in conjunction with those of barometric pressure and the difference in density of flue gases from that of air lead to the following formula:

$$D_t = 2.96\, HB_o \left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) \qquad (1)$$

where

H = height of chimney, feet.
B_o = existing barometric pressure, inches of mercury.
W_o = density of air at 0 F and 1 atmosphere pressure, pounds per cubic foot.
W_c = density of flue gas at 0 F and 1 atmosphere pressure, pounds per cubic foot.

T_o = temperature of air surrounding the chimney, degrees Fahrenheit absolute.
T_c = average or effective temperature of the gases in the chimney, degrees Fahrenheit absolute.

The quantity D_t, yielded by the formula, is the pressure difference between the gas inside and air outside of the chimney, in inches of water, when no flow occurs in the chimney. The quantity is variously known as the theoretical draft, the static draft or the computed draft. It is very useful in predicting and analyzing chimney performance, but it is seldom if ever attained in an actual chimney on account of the friction incident to gas flow, wind effects, etc.

AVAILABLE DRAFT

The available draft, D_a, *for large chimneys and stacks* has been estimated with apparent satisfaction in the past by means of formulae which in effect deduct an estimated friction loss from a theoretical draft determined as in Equation 1. The friction loss can be estimated by means of one of the formulae available for ducts, such as the Fanning equation. This procedure results in formulae for the available draft as follows:

For a *cylindrical* stack:

$$D_a = 2.96\, HB_o \left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.00126\, W^2 T_c f L}{D^5 B_o W_c} \qquad (2)$$

and for a *rectangular* stack:

$$D_a = 2.96\, HB_o \left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.000388\, W^2\, T_c f L\, (x + y)}{xy^3\, B_o\, W_c} \qquad (3)$$

where

D_a = available draft, inches, water gage.
H = height of chimney above grate, feet.
B_o = existing barometric pressure, inches of mercury.
W_o = density of air at 0 F, 1 atmosphere pressure.
W_c = density of flue gas at 0 F, 1 atmosphere pressure.
T_o = temperature of atmosphere, degrees Fahrenheit absolute.
T_c = temperature of flue gas, degrees Fahrenheit absolute.
W = flue gas flow rate, pounds per second.
f = coefficient of friction.
L = length of friction duct (= H approximately), feet.
D = minimum diameter of round chimney, feet.
x and y = length and width of cross-section of rectangular chimney, feet.

The following notes are intended to facilitate the use of Equations 2 and 3.

1. The *barometric pressure*, represented by B_o, is the actual pressure at the site of the chimney and not the pressure reduced to sea level datum.

In general, the barometric pressure decreases approximately 0.1 in. Hg per 100 ft increase in elevation.

2. The ***unit weight of a cubic foot*** of chimney gases at 0 F and sea level barometric pressure is given by the equation:

$$W_c = 0.131 CO_2 + 0.095 O_2 + 0.083 N_2 \qquad (4)$$

In this equation CO_2, O_2 and N_2 represent the percentages of the parts by volume of the carbon dioxide, oxygen and nitrogen content, respectively, of the gas analysis. For ordinary operating conditions, the value of W_c may be assumed at 0.09.

The density effect on the chimney gases due to superheated water vapor resulting from moisture and hydrogen in the fuel, or due to any air infiltrations in the chimney proper are disregarded. Though water vapor content is not disclosed by Orsat analysis, its presence tends to reduce the actual weight per cubic foot of chimney gases.

3. The *atmospheric temperature* is the actual observed temperature of the outside air at the time the analysis of the operating chimney is made. The mean atmospheric temperature in the temperate zone is approximately 62 F.

4. The *chimney gas temperature* decreases from the breeching connection to the top of the stack. This drop in temperature depends upon the material and construction of the stack, its tightness or freedom from leaks, its area, its height, and the velocity of the gases through it. The same chimney will suffer different temperature losses depending upon the capacity under which it is working and the variable atmospheric conditions. No general equation covering all these variables has been suggested, but from observations on chimneys varying in diameter from 3 to 16 ft and in height from 100 to 250 ft Equation 5 was deduced[1]:

$$T_c = \frac{3.13\, T_1 \left[\left(\frac{H_b}{3} \right)^{0.96} - 1 \right]}{H_b - 3} \tag{5}$$

where

T_1 = absolute temperature at the center of the connection from the breeching, degrees Fahrenheit.

H_b = the height of the stack above center line connection to breeching, feet.

5. The *coefficient of friction* between the chimney gases and a sooted surface has been taken by many workers in this field as a constant value of 0.016 for the conditions involved. This value, of course, would be less for a new unlined steel stack than for a brick or brick-lined chimney, but in time the inside surface of all chimneys regardless of the materials of construction becomes covered with a layer of soot, and thus the coefficient of friction has been taken the same for all types of chimneys and in general constant for all conditions of operation. For reasons of simplicity and convenience to the reader, this constant value of 0.016 has been employed in the development of the various special equations and charts shown in this chapter.

In important chimney design, especially when the construction or the materials are unusual, it is recommended that use be made of Reynolds' number[2] in determining the friction factor, f.

To illustrate the use of Equation 2, the following problem is solved by its use.

Example 1. Determine the available draft of a natural draft chimney 200 ft in height and 10 ft in diameter operating under the following conditions: atmospheric temperature, 62 F; chimney gas temperature, 500 F; sea level atmospheric pressure, B_o = 29.92 in. Hg; atmospheric and chimney gas density, 0.0863 and 0.09, respectively; coefficient of friction, 0.016; length of friction duct, 200 ft. The chimney discharges 100 lb of gases per second.

Substituting these values in Equation 2 and reducing:

$$D_a = 2.96 \times 200 \times 29.92 \times \left(\frac{0.0863}{522} - \frac{0.09}{960} \right) - \frac{0.00126 \times 100^2 \times 960 \times 0.016 \times 200}{10^6 \times 29.92 \times 0.09}$$

$$= 1.27 - 0.14 = 1.13 \text{ in.}$$

Fig. 1 shows the variation in the available draft of a typical 200 ft by 10 ft chimney operating under the general conditions noted in Example 1. When the chimney is under static conditions and no gases are flowing, the available draft is equal to 1.27 in. of water, the theoretical intensity. As the amount of gases flowing increases, the available draft decreases until it becomes zero at a gas flow of 297 lb per second, at which point the draft loss due to friction is equal to the theoretical intensity. The point of maximum draft and zero capacity is called shut-off draft, or point of

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[1]Notes on Power Plant Design, by E. F. Miller and James Holt (*Massachusetts Institute of Technology*, 1930).

[2]A.S.H.V.E. RESEARCH REPORT No. 1105—Frictional Resistance to the Flow of Air in Straight Ducts, by F. C. Houghten, J. B. Schmieler, J. A. Zalovcik and N. Ivanovic (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 35) and for more complete discussion see Flow of Fluids in Closed Conduits, by R. J. S. Pigott (*Mechanical Engineering*, August, 1933).

impending delivery, and corresponds to the point of shut-off head of a centrifugal pump. The point of zero draft and maximum capacity is called the wide open point and corresponds to the wide open point of a centrifugal pump. A set of operating characteristics may be developed for any size chimney operating under any set of conditions by substituting the proper values in Equation 2 and then plotting the results in the manner shown in Fig. 1.

Fig. 2 is a typical chimney performance chart giving the available draft for various gas flow rates and sizes of chimney. This chart is based on an atmospheric temperature of 62 F, a chimney gas temperature of 500 F, a unit chimney gas weight of 0.09 lb per cubic foot, sea level atmospheric pressure, a coefficient of friction of 0.016, and a friction duct length equal to the height of the chimney above the grate level. These curves may be

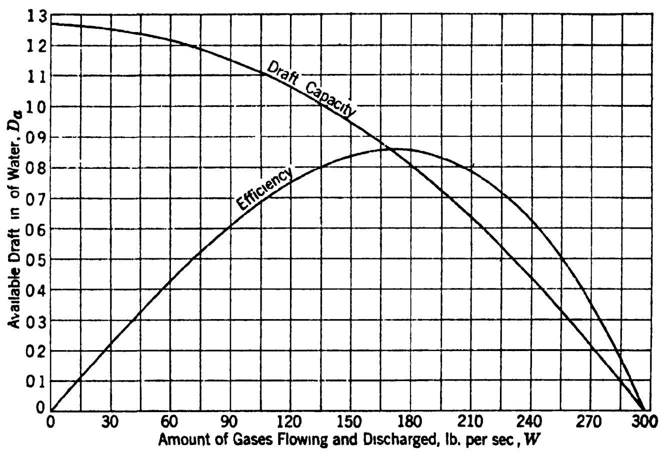


FIG. 1. TYPICAL SET OF OPERATING CHARACTERISTICS OF A NATURAL DRAFT CHIMNEY

used for general operating conditions. For specific conditions, a new chart may be prepared from Equation 2 or 3.

DETERMINING CHIMNEY SIZES

If the required performance for a proposed chimney is known and *if a chimney-gas velocity is assumed*, Equation 2 can be transposed to yield the necessary height and an equation can be developed for the required diameter. These operations result in the following equations:

$$H = \frac{D_r}{2.96B_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.184fW_cB_oV^2}{T_cD}} \tag{6}$$

The weight of gas per second, $W = 12.075 \frac{D^2 VB_o W_c}{T_c}$ from which

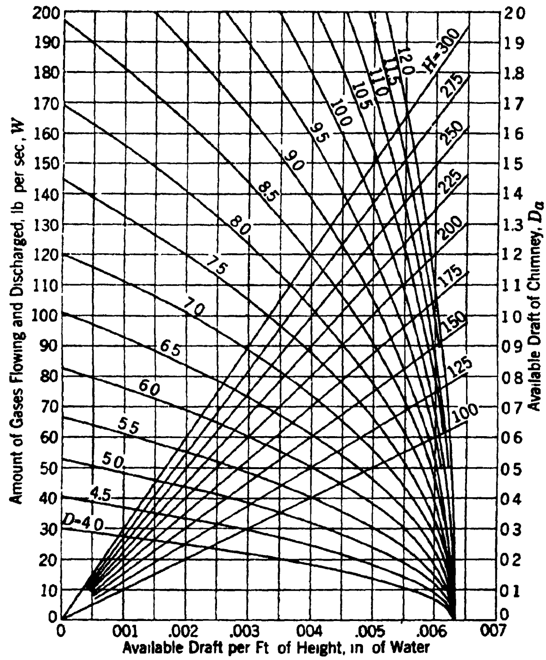
$$D = 0.288\sqrt{\frac{WT_c}{B_oW_cV}} \tag{7}$$

where

H = required height of chimney above grate, feet.
D = required minimum diameter of chimney, feet.

V = chimney gas velocity, feet per second.
D_r = total required draft.

For large chimneys, it is usual to assume that total construction cost is least when the product HD (height × diameter) is minimum. On this assumption, the product of Equations 6 and 7 can be differentiated and



To solve a typical example: Proceed horizontally from a Weight Flow Rate point to intersection with diameter line; from this intersection follow vertically to chimney height line; from this intersection follow horizontally to the right to Available Draft scale. Starting from a point of Available Draft, take steps in reverse order.

FIG. 2. CHIMNEY PERFORMANCE CHART

the differential set equal to zero to find the minimum. Solution for velocity then yields the following equation:

$$V_e = \left(\frac{0.772 T_c \left(\frac{W_o}{T_o} - \frac{W_c}{T_c} \right) \sqrt{\frac{WT_c}{B_o W_c}}}{f W_c} \right)^{2/5} \tag{8}$$

where V_e = economical chimney gas velocity, feet per second.

Equations 6, 7 and 8 can of course be simplified if values are assumed for some of the factors in it. Some typical figures for boiler plants are as follows:

Average chimney gas temperature 500 F .... T_c = 960 F absolute
Average atmospheric temperature 62 F .... T_o = 522 F absolute
Average coefficient of friction 0.016 .... f = 0.016
Average chimney, gas density, 0 F, 1 Atmosphere .... W_c = 0.09 lb per cubic foot
Barometer reading, sea level .... B_o = 29.92 in. Hg

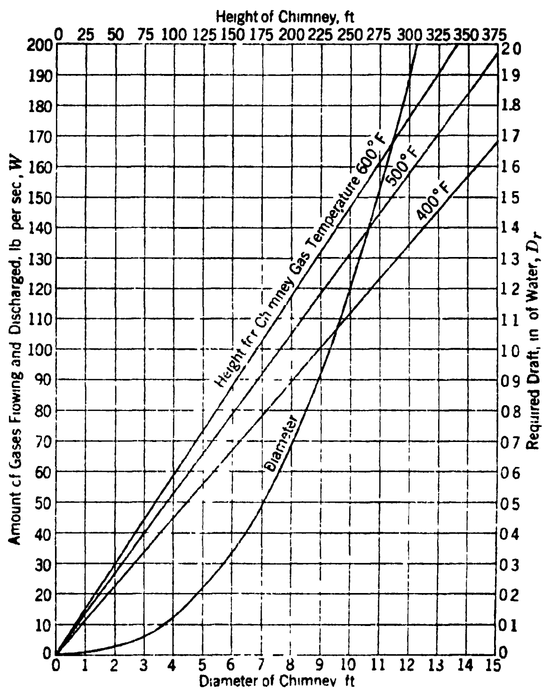
When these values are substituted in Equations 8, 7 and 6 respectively, the results are:

$$V_e = 13.7W^{1/5} \tag{9}$$

$$D = 1.5W^{2/5} \tag{10}$$

$$H = 190D_r \tag{11}$$

Fig. 3 gives the economical chimney sizes for various amounts of gases



Diameter values also for gas temperatures of 400, 500 and 600 F.

FIG. 3. ECONOMICAL CHIMNEY SIZES

flowing and for required draft intensities computed from Equations 9, 10 and 11. They are based on the operating factors used in reducing Equations 6, 7 and 8 to their simpler form. The sizes shown by the curves in the chart should be used for general operating conditions only, or for installations where the required data necessary for an exact determination are difficult or impossible to secure. Whenever it is possible to secure accurate data, or the anticipated operating conditions are fairly well known, the required size should be determined from Equations 6, 7 and 8.

The recommended minimum inside dimensions and heights of chimneys for small and medium size installations are given in Table 1.

FACTORS AFFECTING REQUIRED DRAFT

The foregoing considerations deal with chimney size selection when the required draft and flue gas volume and temperature are known. The

required draft is, of course, equal to the sum of all the resistances to gas flow from the ash pit door to and including the chimney connection.

Fig. 4 presents information on the fuel-bed draft loss for various kinds of coal burned at different rates and rough generalizations can be given for the losses in the flue passages of boiler or furnace, but, on account of the great differences in such devices, more reliable data on their flue gas volume temperature and flue resistance should be obtained for design purposes from their respective manufacturers.

Flue gases encounter resistance to flow in breechings or smoke pipes and this can probably be treated with sufficient accuracy by means of the method used for air ducts. (See Chapter 31). The friction in straight ducts can be estimated by means of the last terms of foregoing Equations 2 and 3.

Also, the temperature of flue gases falls during passage through breechings or flue pipes. For uninsulated surfaces this probably can be ade-

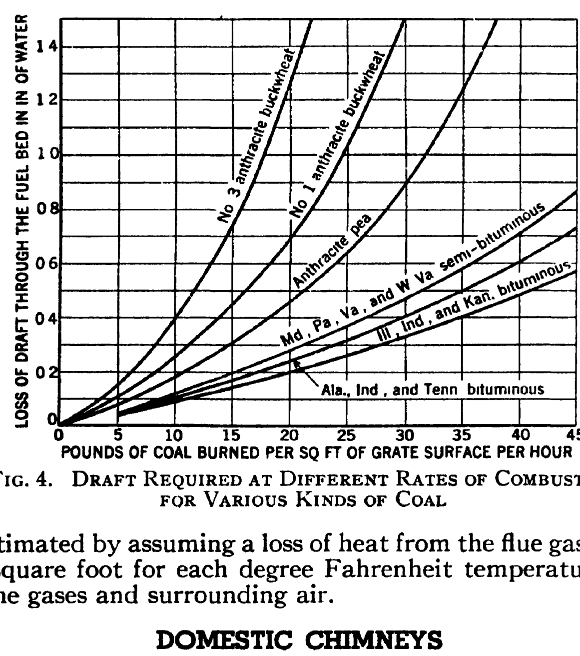


FIG. 4. DRAFT REQUIRED AT DIFFERENT RATES OF COMBUSTION FOR VARIOUS KINDS OF COAL

quately estimated by assuming a loss of heat from the flue gas of 3 Btu per hour per square foot for each degree Fahrenheit temperature difference between the gases and surrounding air.

DOMESTIC CHIMNEYS

Experience has shown that the chimney sizes given in Table 1 are in general adequate for heating boilers and furnaces with the capacities indicated, when such devices are hand fired and that chimneys of these sizes are likely to be architecturally congruous with houses or buildings which require heating devices of such capacities.

Sometimes it is desirable to use a smaller or shorter chimney than that indicated in Table 1. In such cases it is necessary to estimate whether a chimney of limited height can be made to produce sufficient draft in the particular case or whether mechanical draft must be employed.

It should be remembered that mechanically fired devices, oil burners and stokers, are equipped with blowers so that, with these devices, the chimney is not required to overcome the resistance of a fuel bed. Nevertheless, a draft in the fire box, of about 0.03 in. of water is considered

Table 1. Recommended Minimum Chimney Sizes for Heating Boilers and Furnaces[a]

Warm Air Furnace Capacity in Sq In. of Leader Pipe	Steam Boiler Capacity Sq Ft of Radiation	Hot Water Heater Capacity Sq Ft of Radiation	Nominal Dimensions of Fire Clay Lining in Inches	Rectangular Flue		Round Flue		Height in Ft Above Grate
				Actual Inside Dimensions of Fire Clay Lining in Inches	Actual Area Sq In.	Inside Diameter of Lining in Inches	Actual Area Sq In.	
790	590	973	8½x13	7x11½	81			35
1000	690	1,140				10	79	
	900	1,490	13x13	11¼x11¼	127			
	900	1,490	8½x18	6¾x16¼	110			
	1,100	1,820				12	113	40
	1,700	2,800	13x18	11¼x16¼	183			
	1,940	3,200				15	177	
	2,130	3,520	18x18	15¾x15¾	248			
	2,480	4,090	20x20	17¼x17¼	298			45
	3,150	5,200				18	254	50
	4,300	7,100				20	314	
	4,600	7,590	20x24	17x21	357			
	5,000	8,250	24x24	21x21	441			55
	5,570	9,190		24x24[b]	576			60
	5,580	9,200				22	380	
	6,980	11,500				24	452	65
	7,270	12,000		24x28[b]	672			
	8,700	14,400		28x28[b]	784			
	9,380	15,500				27	573	
	10,150	16,750		30x30[b]	900			
	10,470	17,250		28x32[b]	896			

[a]This table is taken from the A S H.V E Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).

[b]Dimensions are for unlined rectangular flues.

desirable so that any small openings in the fire box or flue passages will result in leakage of air in, and not leakage of combustion products out, of such parts. This is not to be taken to condone leaks in fire boxes. Such leaks adversely affect plant efficiency.

OBSERVED TEST PERFORMANCE

The observed performances of some brick chimneys[3] are given in Tables 2 and 3.

The tests on which these data are based were made at various outside temperatures as shown and, to make them comparable among themselves, the observed drafts were corrected to 32 F, 1 atmosphere pressure, by the formula:

$$O_2 = O_1 + S_2 - S_1 \qquad (12)$$

where

S_1 = computed static (theoretical) draft, experimental conditions.

S_2 = computed static (theoretical) draft, standard conditions.

O_1 = observed draft, experimental conditions.

O_2 = observed draft corrected to standard conditions.

It will be noted that the observed draft exceeded the computed static draft during some observations on the shorter chimneys. This is mainly attributed to the draft producing effect of the hot gases immediately above the chimney. By means of a manometer it was found that a

[3]Observed Performance of Some Experimental Chimneys, by R. S. Dill, P. R. Achenbach and J. T. Duck (A.S H V.E. Transactions, Vol 48, 1942, p 351).

TABLE 2. TEMPERATURE AND DRAFT IN 9 IN. BY 9 IN. MASONRY CHIMNEY[a]

Chimney Height, Ft		Fuel Oil Equivalent[c] of Chimney Gases		Flue Gas Temp at Inlet, F	Average Chimney Gas Temp. F	Outside Temp. F	Observed Draft Inch Water	Computed Static Draft Inch Water	Corrected Draft[b]		Indicated Friction Loss Inch Water
Nominal	Height Above Thimble	Fuel Oil Gal/Hr	CO_2 Per Cent						Observed Inch Water	Computed Static Inch Water	
35	31.5	0.5	10	236	142	68	0.040	0.053	0.076	0.089	+0.013
35	31.5	1.5	8	200	158	81	0.043	0.055	0.086	0.098	+0.012
35	31.5	0.5	10	400	224	77	0.081	0.095	0.128	0.142	+0.014
35	31.5	1.5	8	400	278	80	0.092	0.117	0.139	0.164	+0.025
35	31.5	0.5	10	1000	488	77	0.163	0.191	0.206	0.234	+0.028
35	31.5	1.5	8	1000	601	78	0.169	0.214	0.216	0.261	+0.045
30	27.5	0.5	10	250	160	83	0.043	0.049	0.070	0.076	+0.006
30	27.5	1.5	8	250	195	84	0.055	0.064	0.097	0.106	+0.009
30	27.5	0.5	10	400	213	68	0.075	0.084	0.107	0.116	+0.009
30	27.5	1.5	8	400	269	72	0.094	0.108	0.125	0.139	+0.014
30	27.5	0.5	10	1000	460	64	0.146	0.169	0.174	0.197	+0.023
30	27.5	1.5	8	1000	608	72	0.168	0.193	0.204	0.228	+0.025
25	22.3	0.5	10	250	171	92	0.034	0.037	0.072	0.076	+0.003
25	22.3	1.5	8	200	171	94	0.034	0.036	0.074	0.076	+0.002
25	22.3	0.5	10	400	243	78	0.066	0.073	0.097	0.105	+0.006
25	22.3	1.5	8	400	301	79	0.085	0.091	0.116	0.123	+0.006
25	22.3	0.5	10	1000	504	79	0.124	0.137	0.154	0.167	+0.013
25	22.3	1.5	8	1000	642	80	0.146	0.159	0.177	0.190	+0.013
20	17.2	0.5	10	250	179	94	0.029	0.032	0.059	0.062	+0.003
20	17.2	1.5	8	200	188	94	0.031	0.032	0.059	0.060	+0.001
20	17.2	0.5	10	400	270	81	0.058	0.062	0.082	0.086	+0.004
20	17.2	1.5	8	400	297	55	0.075	0.081	0.087	0.093	+0.006
20	17.2	0.5	10	600	330	50	0.086	0.091	0.095	0.100	+0.005
20	17.2	1.5	8	600	425	55	0.104	0.106	0.116	0.118	+0.002
20	17.2	0.5	10	1000	526	37	0.120	0.130	0.123	0.134	+0.010
20	17.2	1.5	8	915	588	44	0.129	0.134	0.135	0.141	+0.005
15	11.3	0.5	10	250	182	87	0.023	0.023	0.041	0.041	+0.000
15	11.3	1.5	8	200	180	84	0.025	0.024	0.042	0.041	−0.001
15	11.3	0.5	10	400	258	32	0.053	0.055	0.054	0.056	+0.002
15	11.3	1.5	8	400	293	43	0.056	0.057	0.060	0.061	+0.001
15	11.3	0.5	10	600	369	34	0.068	0.071	0.069	0.071	+0.003
15	11.3	1.5	8	600	428	40	0.072	0.076	0.075	0.078	+0.004
15	11.3	0.5	10	1000	577	32	0.094	0.092	0.094	0.092	−0.002
15	11.3	1.5	8	1000	654	55	0.092	0.090	0.100	0.098	−0.002

[a]Actual inside dimensions of flue lining 7¼ × 7¼ in.
[b]Corrected for outside temperature 32 F. Barometer 29.92 in. Hg.
[c]½ gal per hour of fuel oil burned with 10 per cent CO_2 produces 18.4 cfm or 1.38 lb per minute of flue gases (corrected to 70 F, 1 atmosphere pressure); 1½ gal per hour of fuel oil burned with 8 per cent CO_2 produces 67.5 cfm or 5.06 lb per minute of flue gas (corrected to 70 F, 1 atmosphere pressure).

TABLE 3. TEMPERATURE AND DRAFT IN 9 IN. BY 13 IN. MASONRY CHIMNEY[a]

Chimney Height, Ft		Fuel Oil Equivalent[c] of Chimney Gases		Flue Gas Temp. at Inlet, F	Average Chimney Gas Temp. F	Outside Temp. F	Observed Draft Inch Water	Computed Static Draft Inch Water	Corrected Draft[b]		Indicated Friction Loss Inch Water
Nominal	Height Above Thimble	Fuel Oil Gal/Hr	CO_2 Per Cent						Observed Inch Water	Computed Static Inch Water	
35	31.5	0.5	10	204	125	78	0.035	0.034	0.078	0.077	−0.001
35	31.5	1.5	8	206	155	82	0.047	0.052	0.093	0.098	+0.005
35	31.5	1.5	8	600	355	84	0.140	0.141	0.193	0.194	+0.001
35	31.5	0.5	10	600	272	78	0.104	0.117	0.152	0.165	+0.013
35	31.5	0.5	10	1000	438	77	0.163	0.178	0.205	0.221	+0.015
35	31.5	1.5	8	1000	542	84	0.189	0.201	0.236	0.247	+0.012
30	27.5	0.5	10	260	155	83	0.040	0.045	0.081	0.086	+0.005
30	27.5	1.5	8	202	159	83	0.048	0.047	0.089	0.088	−0.001
30	27.5	0.5	10	605	278	80	0.089	0.103	0.129	0.144	+0.014
30	27.5	1.5	8	595	378	72	0.130	0.144	0.161	0.175	+0.014
30	27.5	0.5	10	1007	438	91	0.135	0.148	0.179	0.191	+0.013
30	27.5	1.5	8	995	609	89	0.166	0.168	0.225	0.227	+0.002
25	22.3	0.5	10	258	155	86	0.034	0.034	0.069	0.069	0.000
25	22.3	1.5	8	200	160	86	0.039	0.037	0.074	0.072	−0.002
25	22.3	0.5	10	600	312	86	0.086	0.090	0.122	0.126	+0.004
25	22.3	1.5	8	592	382	83	0.106	0.110	0.140	0.144	+0.004
25	22.3	0.5	10	1007	462	85	0.120	0.127	0.154	0.161	+0.007
25	22.3	1.5	8	1007	607	84	0.151	0.153	0.184	0.185	+0.002
20	17.2	0.5	10	280	175	79	0.030	0.028	0.062	0.060	−0.002
20	17.2	1.5	8	250	192	81	0.044	0.041	0.069	0.066	−0.003
20	17.2	0.5	10	598	324	74	0.069	0.079	0.089	0.099	+0.010
20	17.2	1.5	8	604	410	74	0.090	0.094	0.111	0.115	+0.004
20	17.2	0.5	10	1015	496	79	0.094	0.107	0.115	0.128	+0.013
20	17.2	1.5	8	1000	645	74	0.122	0.127	0.140	0.147	+0.005
15	11.3	0.5	10	270	172	88	0.022	0.021	0.041	0.040	−0.001
15	11.3	1.5	8	200	166	85	0.022	0.020	0.040	0.038	−0.002
15	11.3	0.5	10	604	356	86	0.055	0.052	0.073	0.070	−0.003
15	11.3	1.5	8	599	431	84	0.066	0.062	0.082	0.078	−0.004
15	11.3	0.5	10	1000	560	82	0.066	0.074	0.082	0.091	+0.008
15	11.3	1.5	8	1000	682	85	0.083	0.082	0.101	0.100	−0.001

[a]Actual inside dimensions of flue lining 6½ × 11 in.

[b]Corrected for outside temperature 32 F. Barometer 29.92 in. Hg

[c]½ gal per hour of fuel oil burned with 10 per cent CO_2 produces 18.4 cfm or 1.38 lb per minute of flue gases (corrected to 70 F, 1 atmosphere pressure); 1½ gal per hour of fuel oil burned with 8 per cent CO_2 produces 67.5 cfm or 5.06 lb per minute of flue gas (corrected to 70 F, 1 atmosphere pressure).

measurable draft existed in this gas column for some distance above the chimney top. However, the temperatures in the chimney were measured with unshielded thermocouples and the actual gas temperatures may have been higher for this reason than the observed temperatures on which the computations of draft were based. The tests were made in calm weather.

Tests were made at the National Bureau of Standards to find the draft produced by round, metal smoke pipe set in a vertical position to act as chimneys. Curves are presented in Fig. 5 showing the computed static or theoretical draft for short chimneys for various heights and temperatures. For this purpose, the density of chimney gases was assumed

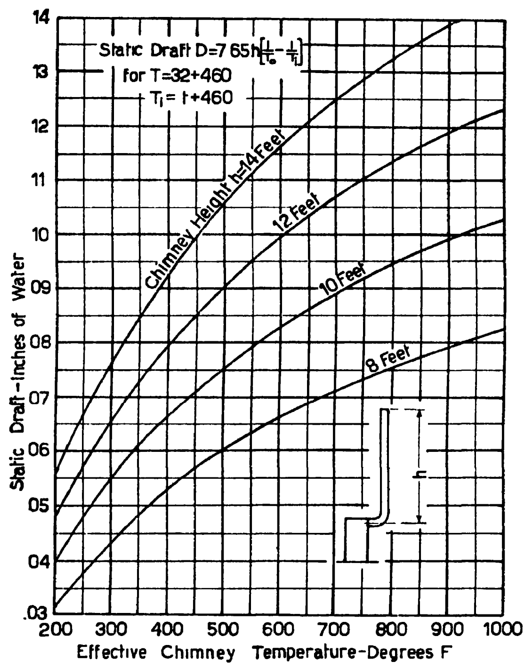


Fig. 5. Computed Static Draft for Short Chimneys

to be the same as that of air at the same condition, since the error thus introduced was not considered important in this case. The results of the tests showed that the following procedures would yield the available draft for 6-in. flue pipe used as a chimney within 10 per cent for the range shown and for fuel burning rates from about one-quarter to three-quarters of a gallon of oil per hour.

Using the temperature at the smoke collar of the heater, find the static draft corresponding to the available chimney height. Then:

1. If the chimney is bare, multiply the static draft by 0.76 to find the available draft.

2. If the chimney is insulated with 1 in. of air-cell material with a ¼-in. air space, multiply the static draft by 0.85 to find the available draft.

3. If the chimney is insulated with 1 in. of air-cell material with a 1-in. air space, open at top and bottom for ventilation, multiply the static draft by 0.81 to find the available draft.

4. If the chimney is insulated with 1 in. of air-cell material and has a 1-in. air space closed at top and bottom to prevent ventilation, multiply the static draft by 0.85 to find the available draft.

The use of the 1-in. air-cell asbestos insulation in the tests discussed is not to be construed as an approval of such insulation in all cases in regard to fire resistance. Several laboratories are working on the fire resistance aspects of the problem but definite rules are not yet available. For coal- or oil-burning devices, a bare smoke pipe is probably safe if kept a couple of feet or more from any woodwork and the better the pipe is insulated, or the lower its temperature, the nearer it can be placed to combustible materials.

DRAFT REQUIREMENTS OF DOMESTIC APPLIANCES

Typical flue-gas temperatures and drafts required at rated output for several kinds of domestic heating devices or appliances[4] are contained in Table 4.

TABLE 4 DRAFTS REQUIRED BY TYPICAL DOMESTIC HEATING DEVICES OR APPLIANCES

Device	Draft, Inches Water	Stack Temperature Deg F
Space Heater, Oil Burning, Pot Burner	0.06 to 0.08	1000
Warm Air Furnace, Oil Burning, Pot Burner	0.06	860
Warm Air Furnace, Hand Fired	0.06[b]	900
Floor Furnace, Oil Burning, Pot Burner	0.06	860
Mechanical Oil Burner, Less than 5 gph	0.03[a]	......
Mechanical Oil Burner, More than 5 gph	0.05[a] or less	......
Cooking Stove, Solid Fuel	0.04[b]	400
Space Heater, Coal Burning	0.06[b]	900

[a]Draft in fire-box
[b]For chestnut sized anthracite

CHIMNEYS FOR GAS HEATING

Heating appliances designed to burn gas as well as appliances converted to gas burning, except those equipped with power type burners, are always equipped with a draft hood attached to the flue outlet of the appliance. This draft hood is required if the appliance is to meet the approval requirements of the *American Gas Association* and the *American Standards Association* and is essential for safe operation. It is designed to prevent excessive chimney draft which would lower appliance efficiency, to prevent a blocked flue or a down draft in the chimney from impairing combustion, to provide a relief opening for the products of combustion during down draft or blocked flue conditions, and to prevent spillage of the products of combustion to the space surrounding the appliance if there is a chimney draft equivalent to that provided by a 3-ft chimney. As the draft hood is designed without moving parts, the relief opening is always open and consequently some air is drawn into the chimney. While the air drawn in lowers the gas temperature in the chimney, it also lowers the dew-point of the gases and tends to prevent condensation of moisture.

The products of complete combustion of gas are water vapor (H_2O) and carbon dioxide (CO_2) with a trace of sulphur trioxide (SO_3). Sulphur usually burns to the trioxide in the presence of an iron catalyst. The

[4]*National Bureau of Standards Commercial Standards:* CS101-43 Oil-Burning Space Heaters Equipped With Vaporizing Pot-Type Burners, CS75-42 Automatic Mechanical Oil Burners Designed for Domestic Installations, CS(E)104-43 Warm Air Furnaces Equipped With Vaporizing Pot-Type Burners; and *Trade Standards:* TS3536a Solid-Fuel-Burning Forced Air Furnaces, TS3518 Oil-Burning Floor Furnaces Equipped With Vaporizing Pot-Type Burners

volume of water vapor in the flue products from natural or coke oven gas is about twice the volume of carbon dioxide. It is extremely important that the chimney be tight and resistant to corrosion not only of moisture, but also of dilute sulphur trioxide.

Vitreous tile linings with joints which prevent retention of moisture and linings made of non-corrosive materials are advantageous. The protection of unlined chimneys has been investigated and the results indicate that after the loose material has been removed, the spraying with a water emulsion of asphalt chromate will provide excellent protection.

Advice regarding recommended practice and materials for flue connections and chimney linings can usually be obtained from the local gas company and should be given careful consideration.

Since a gas designed appliance must be able to operate at rated input (plus 10 or 15 per cent) without chimney connection, and without producing carbon monoxide, the only function of the chimney is to remove the products of combustion from the room. The chimney provides draft to overcome the friction in the flue pipe and chimney, but does not draw air into the appliance.

Chimneys for venting appliances designed for burning gas can therefore be low in height, but must have adequate area. The height is usually established by the building height. Chimney sizes are usually selected on the basis of Btu input to the appliance. One chart[5] designed to facilitate selection is shown in Fig. 6. The assumptions made in preparing the chart as well as its limitations should be noted carefully.

Since Fig. 6 has been prepared for circular flues, relative capacities for rectangular and elliptical flues[6] are shown in Figs. 7 and 8.

When a flue is connected to several appliances, the number of horizontal runs of various sizes which may be substituted for the single run having a diameter equal to that of the flue may be obtained from Table 5.

TABLE 5. EQUIVALENT FLUE PIPE SIZES[a]

Diameter of Horizontal Runs	Size of Flue						
	3	4	5	6	8	10	12
3	1	2	3	5	9	12	22
4		1	2	3	5	7	11
5			1	2	3	4	7
6				1	2	3	5
8					1	2	3

[a]Comfort Heating (*American Gas Association*)

CONSTRUCTION DETAILS

For general data on the construction of chimneys reference should be made to the Building Code recommended by the *National Board of Fire Underwriters*, Article XI, Sections 1101 to 1105, in which the following are some of the important provisions listed in the 1943 edition:

(*a*) Chimneys erected within or attached to a structure shall be constructed of brick, of solid block masonry, or of reinforced concrete.

[5]Comfort Heating, 1938, p. 71 (*American Gas Association*).
[6]Loc. Cit. Note 5, p. 74.

(*b*) Chimneys shall extend at least 3 ft above the highest point where they pass through the roof of the building and at least 2 ft higher than any ridge within 10 ft of such chimney.

(*c*) Every such chimney shall be properly capped with brick, terra cotta, stone, cast-iron, concrete or other approved non-combustible, weatherproof material.

(*d*) Chimneys shall be wholly supported on approved masonry or self-supporting fireproof construction.

(*e*) No such chimney shall be corbeled from a wall more than 6 in.; nor shall such chimney be corbeled from a wall which is less than 12 in. in thickness unless it projects equally on each side of the wall; provided that in the second story of two-story dwellings

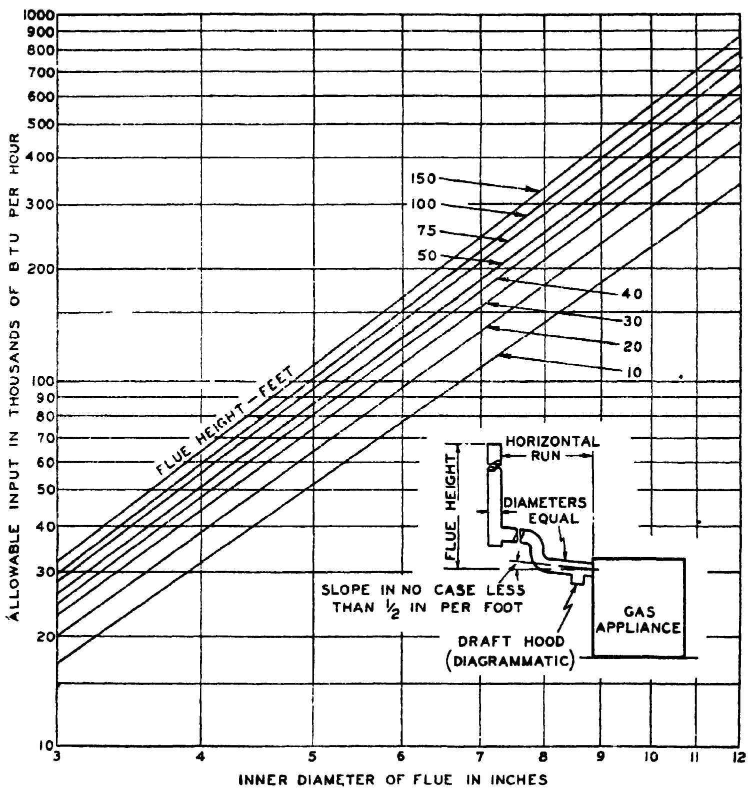


FIG. 6. ALLOWABLE BTU INPUT TO CIRCULAR FLUES FOR DOMESTIC GAS APPLIANCES WITH DRAFT HOODS

Notes:

1 Chart is based on: average flue temperature of 150 F, outside temperature of 60 F, barometric pressure of 30 in. Hg. 100 per cent excess air and 100 per cent dilution at draft hood

2 Based on terra-cotta lined flues. With rough brick flues, capacities are 15 per cent less.

3. Based on condition that horizontal run is not greater than 20 ft except for a flue height less than 20 ft, in which case the horizontal run is not to have greater length than the height of the flue

4. Two long radius elbows are included in the horizontal run, the diameter of which is equal to that of the flue.

5. Each additional elbow reduces the allowable horizontal run by a length in feet equal to the diameter in inches.

6. When the horizontal run has an effective length in excess of that given (or additional elbows) the next larger size of flue should be chosen. It is desirable that long horizontal runs be insulated to reduce heat loss of flue products and to conserve draft.

7. Capacities should be reduced 3.5 per cent for each 1000 ft above sea level.

corbeling of chimneys on the exterior of the enclosing walls may equal the wall thickness. In every case the corbeling shall not exceed 1 in. projection for each course of brick projected.

(*f*) No change in the size or shape of a chimney, where the chimney passes through the roof, shall be made within a distance of 6 in. above or below the roof joists or rafters.

(*g*) Smoke flues for warm air, hot water and low pressure steam heating furnaces shall have walls not less than 8 in. thick; the walls may be of solid masonry using brick, stone or concrete, or of solid moulded or solid cast chimney units of concrete, or of burned clay, or of suitably reinforced solid concrete cast in place; provided that for stone masonry other than sawed or dressed stone in courses, the thickness shall be not less than 12 in. The walls shall be properly bonded, or tied with non-corrosive metal anchors. In dwellings and buildings of like heating requirements the thickness of the chimney walls may be reduced to not less than 3¾ in. when lined with a flue lining conforming to the Code requirements.

(*h*) Required flue linings shall be made of fire clay or other refractory clay to withstand the action of flue gases and to resist, without softening or cracking, the temperatures to which they will be subjected, but not less than 2,000 F, or of cast-iron of ap-

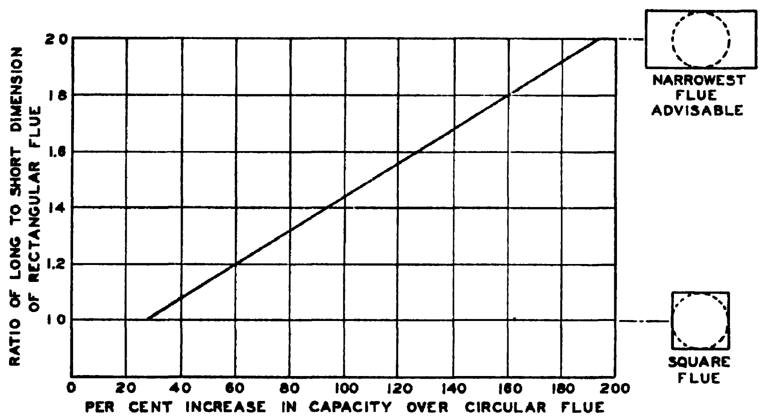


FIG. 7. CAPACITY OF A RECTANGULAR FLUE, HAVING ITS MINIMUM WIDTH EQUAL TO THE DIAMETER OF A CIRCULAR FLUE, COMPARED WITH THE CAPACITY OF THE CIRCULAR FLUE

proved quality, form and construction. Approved corrosion resistant linings may be used in flues for gas appliances.

(*i*) Required clay flue linings shall be not less than ⅝ in. thick for the smaller flues and increasing in thickness for the larger flues.

(*j*) Flue linings shall be built ahead of the construction of the chimney as it is carried up, carefully bedded one on the other in mortar as hereinafter specified with close fitting joints left smooth on the inside.

(*k*) Flue linings shall start from a point not less than 8 in. below the intake They shall extend, as nearly vertically as possible, for the entire height of the chimney. It is recommended that flue linings be extended 4 in. above the top or cap of the chimney

(*l*) Only Portland cement mortar, cement lime mortar or fire clay mortar shall be used in setting flue linings.

For *gas appliances* the Building Code specifies lined chimneys and metal smoke stacks for all appliances which may be converted readily to the use of solid or liquid fuel and also for all boilers and furnaces except those having a flue gas temperature not exceeding 550 F at the outlet of the draft hood when burning gas at the manufacturer's rating and which may therefore be connected to *Type B* vent piping. Approved *Type B* vent piping is non-combustible, corrosion resistant piping of adequate strength and heat insulating value, and having bell and spigot or other acceptable joints.

All flue mortar for flues or vent pipes from gas burning appliances shall be acid resisting.

GENERAL CONSIDERATIONS FOR CHIMNEYS

The draft of domestic chimneys may be subject to a variety of influences not usually encountered in power chimneys[7]. Horizontal winds have an aspirating effect as they cross the chimney and are an aid to draft. However, surrounding objects, such as trees or other buildings, may affect the direction of the wind at the chimney top and may even direct it down the chimney, tending to reduce the draft or even to cause it to be negative. Although the chimney should extend well above the highest part of the roof, it is impracticable to carry it much beyond this point.

It is also important to consider the source of the air supply for proper combustion. Usually the boiler or furnace is located in the basement.

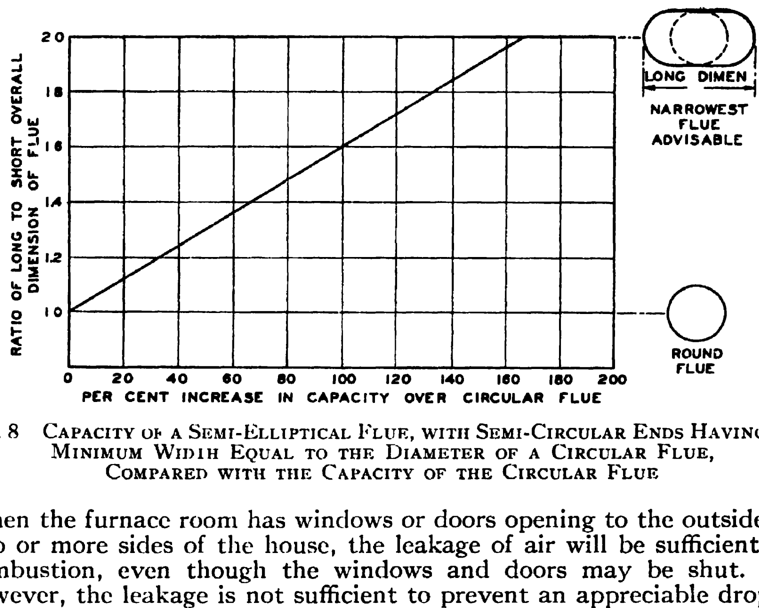


FIG. 8 CAPACITY OF A SEMI-ELLIPTICAL FLUE, WITH SEMI-CIRCULAR ENDS HAVING ITS MINIMUM WIDTH EQUAL TO THE DIAMETER OF A CIRCULAR FLUE, COMPARED WITH THE CAPACITY OF THE CIRCULAR FLUE

When the furnace room has windows or doors opening to the outside on two or more sides of the house, the leakage of air will be sufficient for combustion, even though the windows and doors may be shut. If, however, the leakage is not sufficient to prevent an appreciable drop of pressure in the furnace room below that of the air outside, the chimney draft will be reduced by the difference between the atmospheric pressure outside and that inside the boiler room. In case the boiler room is fairly tight and is open to the outside on only one side of the house, then the draft will be affected in windy weather even with windows or doors open. If the wind is blowing toward the boiler room the draft will be increased, but if blowing in the opposite direction the draft may be decreased.

It is not to be assumed that increasing the cross-section area of a chimney will always effect a cure for poor draft. The opposite result may be experienced because of the cooling effect of the larger area. This reduces the theoretical draft and the velocity of the gases, and affords a greater opportunity for counter currents in the chimney. Sometimes the only practical remedy for a chimney with bad draft, when the chimney is

[7]Chimneys and Draft (Chapter 32 in Winter Air Conditioning, by S. Konzo, published by *National Warm Air Heating and Air Conditioning Association*, 1939).

of the proper size and is affected by conditions beyond control, is to resort to mechanical draft. This can often be done at small expense and the arrangement can be such that the fan or blower need be operated only when conditions are bad.

Two or more chimneys, either large or small, should never be connected together especially near the bottom. Hot gases in an inverted U-tube thus formed will be in unstable equilibrium. Cold air will descend through one such chimney, from the top, and drive the hot gases out of the other and thus annul the draft.

More than one device can be served by one chimney. Batteries of boilers are commonly connected to a single chimney in power plants. However, if two or more chimneys are used, each chimney should be used separately for part of the boilers, and not connected in manifold with another chimney, in order to avoid the difficulty described previously.

In domestic installations it is sometimes necessary to serve a space heater or cooking stove and a water heater with the same chimney flue. This is not desirable, especially for low chimneys, since doors left open on one device while it is un-fired will tend to annul the draft on another device. Gas burning devices, with their draft hoods and lack of draft dampers, are especially bad in this respect. The traditional method of avoiding this with brick chimneys has been to construct multiple-flue chimneys, so that each fuel-burning device could be served by a separate opening. If two devices must be served by one flue-opening in a chimney, their connections to the chimneys should not be located opposite each other. The connection from the larger device should be reasonably low down and that from the smaller, up near the ceiling, so that each device can be serviced as well as possible, regardless of the treatment of the other.

Excessive height in a chimney does no harm but means for controlling the draft are more than ordinarily essential if the chimney is too large in capacity. Coal-burning devices often have air leaks around the fire-box and the draft doors sometimes fit poorly so that the fire cannot be controlled at a low rate. Perhaps the simplest remedy for such cases is the barometric damper which admits air into the flue pipe and thus reduces the draft.

Directions for building chimneys for fireplaces are contained in Department of Agriculture Farmers' Bulletin No. 1230.

It is considered bad practice to connect any heating device to a fireplace flue unless the fireplace is effectively sealed.

CHAPTER 10

Automatic Fuel Burning Equipment

Classification of Stokers, Combustion Process and Adjustments, Furnace Design, Classification of Oil Burners, Combustion Chamber Design, Classification of Gas-Fired Appliances

AUTOMATIC mechanical equipment for the combustion of solid, liquid, and gaseous fuels is considered in this chapter.

MECHANICAL STOKERS

A mechanical stoker is a device that feeds a solid fuel into a combustion chamber, provides a supply of air for burning the fuel under automatic control and, in some cases, incorporates a means of removing the ash and refuse of combustion automatically. Coal can be burned more efficiently by a mechanical stoker than by hand firing because the stoker provides a uniform rate of fuel feed, better distribution in the fuel bed and positive control of the air supplied for combustion.

CLASSIFICATION OF STOKERS ACCORDING TO CAPACITY

Stokers may be classified according to their coal feeding rates. The following classification has been made by the *U. S. Department of Commerce*, in cooperation with the *Stoker Manufacturers Association.*

Class 1. Capacity under 61 lb of coal per hour.
Class 2. Capacity 61 to 100 lb of coal per hour.
Class 3. Capacity 101 to 300 lb of coal per hour.
Class 4. Capacity 300 to 1200 lb of coal per hour.
Class 5. Capacity 1200 lb of coal per hour and over.

Class 1 Stokers

These stokers are used primarily for home heating and are designed for quiet, automatic operation. Simple, trouble-free construction and attractive appearance are desirable characteristics of these small units.

A common stoker in this class (Fig. 1) consists essentially of a coal hopper, a screw for conveying the coal from the hopper to the retort, a fan which supplies the air for combustion, a transmission for driving the coal feed worm, and an electric motor for supplying power for coal feed and air supply.

Air for combustion is admitted to the fuel through tuyeres at the top of the retort, and in this class, the retort may be either round or rectangular. Stokers in this class are made for burning anthracite, bituminous, semi-bituminous, and lignite coals, and coke. The *U. S. Department of Commerce* has issued commercial standards for household anthracite stokers[1].

Units are available in either the hopper type, as shown in Fig. 1 or in the bin-feed type as shown in Figs. 2 and 3. Some stokers, particularly those designed for use with anthracite automatically remove ash from the ash pit and deposit it in an ash receptacle as shown in Fig. 3. Most of the bituminous models, however, require removal of the ash from the fuel bed after it is fused into a clinker.

[1]Domestic Burners for Pennsylvania Anthracite (Underfeed Type), (*U. S. Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS48-40).

Stokers in this class feed coal to the furnace intermittently in accordance with temperature or pressure demands. A special control is used to insure sufficient stoker operation to maintain a fire during periods when no heat is required.

Stoker-Fired Boiler and Furnace Units

Boilers, air conditioners, and space heaters especially designed for stokers are available having design features closely coordinating the heat absorber and the stoker. Although efficient and satisfactory performance can be obtained from the application of stokers to existing boilers and furnaces, some of the combination stoker-fired units (Fig. 4) are more compact and attractive in appearance.

Class 2 and 3 Stokers

Stokers in this class are usually of the screw feed type without auxiliary

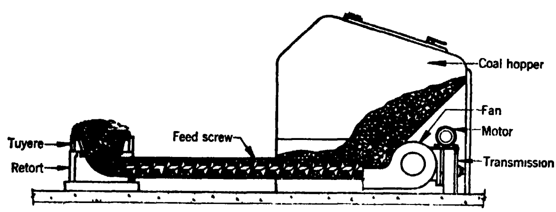


FIG. 1. UNDERFEED STOKER, HOPPER TYPE, CLASS 1

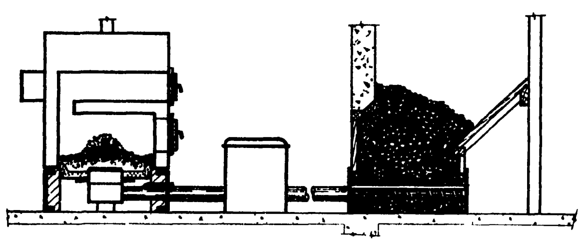


FIG. 2. UNDERFEED STOKER, BIN FEED TYPE, CLASS 1

plungers or other means of distributing the coal. They are used extensively for heating plants in apartments and hotels, also, for industrial plants.

They are of the underfeed type and are available in both the hopper type, as illustrated in Fig. 5, and the bin feed type, shown in Fig. 6. These units also are built in plunger feed type with an electric motor or a steam or hydraulic cylinder coal feed drive.

Stokers in this class are available for burning all types of anthracite, bituminous and lignite coals. The tuyere and retort design varies according to the fuel and load conditions. Stationary type grates are used on bituminous models and the clinkers formed from the ash accumulate on the grates surrounding the retort.

Anthracite stokers in this class are equipped with moving grates which discharge the ash into a pit below the grate. This ash pit may be located on one or both sides of the grate and on some installations is of sufficient capacity to hold the ash for several weeks' operation.

Class 4 Stokers

Stokers in this group vary widely in details of design and several methods of feeding coal are employed. The underfeed stoker is widely used, although a number of the overfeed types are used in the larger sizes. Bin-feed, as well as hopper models, are available in both underfeed and overfeed types.

Class 5 Stokers

The prevalent stokers in this field are: (1) underfeed side cleaning, (2) underfeed rear cleaning, (3) overfeed flat grate, and (4) overfeed inclined grate.

Underfeed side cleaning stokers are made in sizes up to approximately 500 boiler horsepower. They are not so varied in design as those in the smaller classes, although the principle of operation is similar. A stoker of this type is illustrated in Fig. 7.

The rear cleaning underfeed stoker is usually of the multiple retort design and is used in some of the largest industrial plants and central power stations. Zoned air control has been applied on these stokers, both longitudinally and transversely of the grate surface.

The overfeed flat grate stoker is represented by the various chain—or traveling-grate stokers. A typical traveling-grate stoker is illustrated in Fig. 8.

Another distinct type of overfeed flat-grate stoker is the spreader (Figs. 9 and 10) type in which coal is distributed either by rotating paddles or by air over the entire grate surface. This type of stoker is adapted to a wide range of fuels and has a wide application on small sized fuels, and on fuels such as lignites, high-ash coals, and coke breeze.

The overfeed inclined-grate stoker operates on the same general combustion principle as the flat-grate stoker, the main difference being that rocking grates, set on an incline, are provided in the former to advance the fuel during combustion.

Combustion Process

In anthracite stokers of the Class 1 underfeed type, burning takes place entirely within the stoker retort. The refuse of combustion spills over the edge of the retort into an ash pit or receptacle from which it may be removed either manually or automatically.

Larger underfeed anthracite stokers operate on the same principle, except that the retort is rectangular and the refuse spills over only one or two sides of the grate. Anthracite for stoker firing is usually the No. 1 buckwheat or No. 2 buckwheat size.

Because the majority of the smaller bituminous coal stokers operate on the underfeed principle, a general description of their operation is given. When the coal is fed into the retort, it moves upward toward the zone of combustion and is heated by conduction and radiation from the burning fuel in the combustion zone. As the temperature of the coal rises, it gives off moisture and occluded gases, which are largely non-combustible. When the temperature increases to around 700 or 800 F the coal particles become plastic, the degree of plasticity varying with the type of coal.

A rapid evolution of the combustible volatile matter occurs during and directly after the plastic stage. The distillation of volatile matter continues above the plastic zone where the coal is coked. The strength and porosity of the coke formed will vary according to the size and characteristics of the coal. While some of the ash fuses into particles on the surface

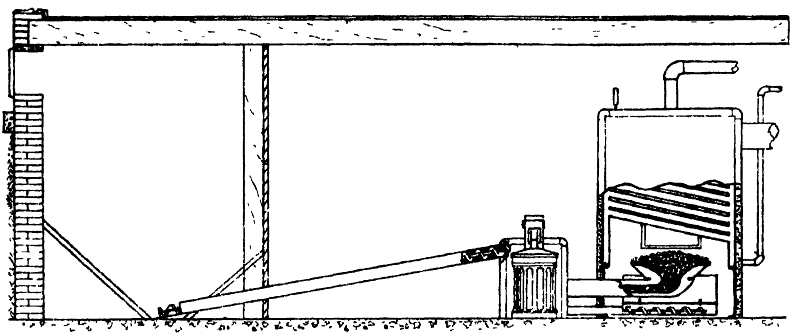


FIG. 3. UNDERFEED ANTHRACITE STOKER WITH AUTOMATIC ASH REMOVAL, BIN TYPE

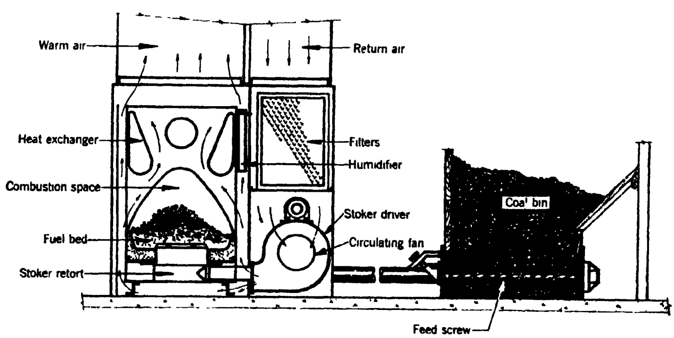


FIG. 4. STOKER-FIRED WINTER AIR CONDITIONING UNIT

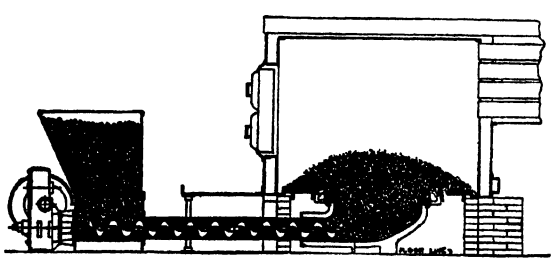


FIG. 5. UNDERFEED SCREW STOKER, HOPPER TYPE, CLASS 2, 3 OR 4

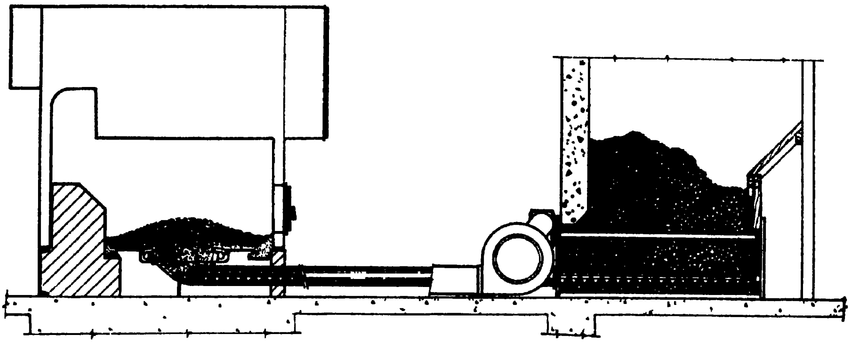


FIG 6. UNDERFEED SCREW STOKER, BIN TYPE, CLASS 2, 3 OR 4

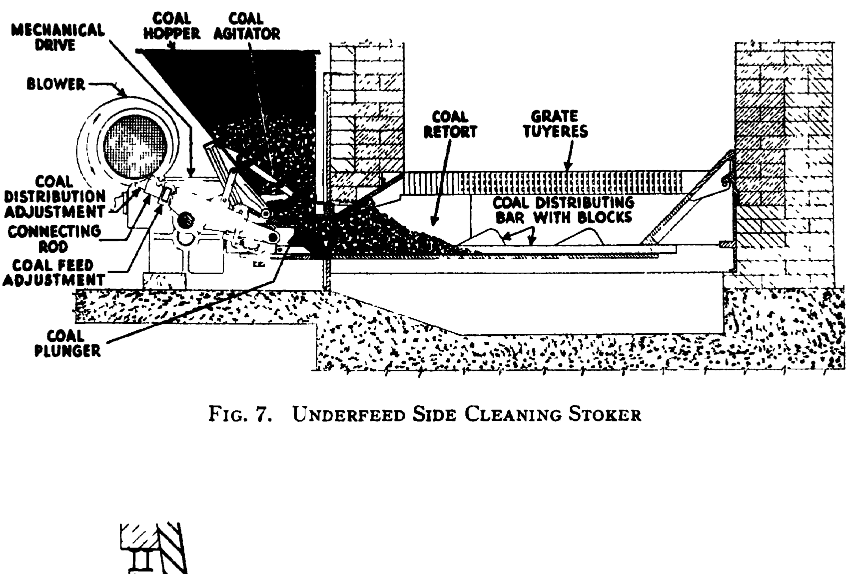


FIG. 7. UNDERFEED SIDE CLEANING STOKER

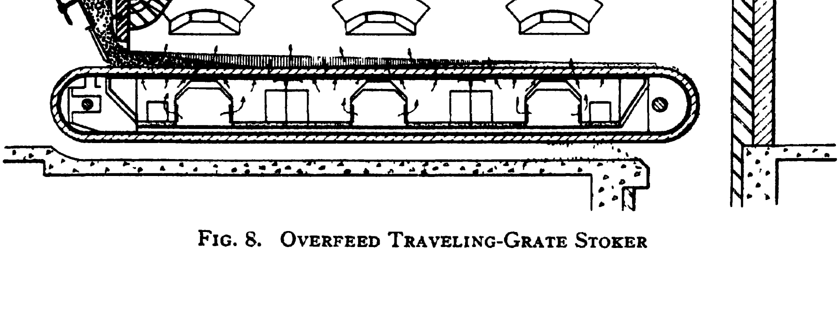


FIG. 8. OVERFEED TRAVELING-GRATE STOKER

of the coke as it is released, most of it remains on the hearth or grates and as this ash layer becomes thicker with time, that portion exposed to the higher temperatures surrounding the retort fuses into a clinker. The temperature in the fuel bed, the chemical composition and homogeneity of the ash, and the time of heating govern the degree of fusion.

Most bituminous coal stokers of Classes 1, 2, 3 and 4 require manual removal of the ash in clinker form.

In the underfeed side-cleaning stokers the fuel is introduced at the front of the furnace to one or more retorts, and is advanced away from the retort as combustion progresses, while finally the ash is disposed of at the sides. This type of stoker is suitable for all bituminous coals while in the smaller sizes it is suitable for small sizes of anthracite. In this type of stoker the fuel is delivered to a retort beneath the fire and is raised into the fire. During this process the volatile gases are released, are mixed with air, and pass through the fire where they are burned. The ash may be continuously or periodically discharged.

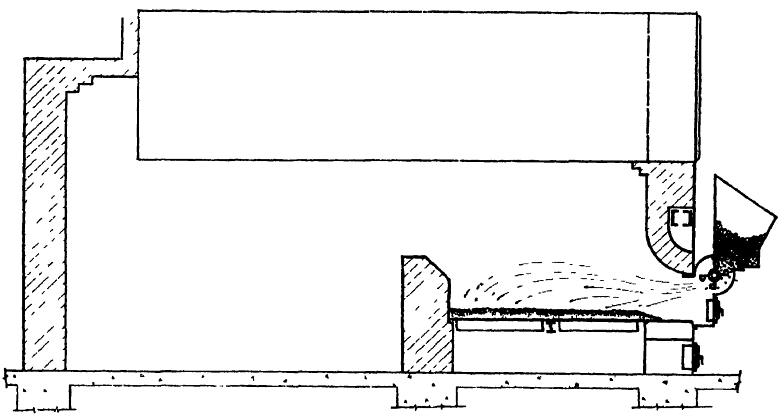


FIG. 9. OVERFEED SPREADER STOKER (ROTOR TYPE)

The underfeed rear-cleaning stoker accomplishes combustion in much the same manner as the side-cleaning type, but consists of several retorts placed side by side and filling up the furnace width, while the ash disposal is at the rear. In principle, its operation is the same as the side cleaning underfeed type.

Overfeed flat-grate stokers receive fuel at the front of the grate in a layer of uniform thickness and move it horizontally to the rear of the furnace. Air is supplied under the moving grate to carry on combustion at a sufficient rate to complete the burning of the coal near the rear of the furnace. The ash is carried over the back end of the stoker into an ash pit beneath. This type of stoker is suitable for small sizes of anthracite or coke breeze, and also for bituminous coals, the characteristics of which make it desirable to burn the fuel without disturbing it. This type of stoker requires an arch over the front of the fuel bed to maintain ignition of the incoming fuel. Frequently, a rear combustion arch is also required.

In addition to the use of rocking grates, the overfeed inclined-grate stoker is provided with an ash plate on which ash is accumulated and dumped periodically. This type of stoker is suitable for all types of

coking fuels but preferably for those of low volatile content. Its grate action keeps the fuel bed broken up thereby allowing for free passage of air. Because of its agitating effect on the fuel it is not desirable for badly clinkering coals. It usually should be provided with a front arch to ignite the volatile gases.

Combustion Adjustments

The coal feeding rate and air supply to the stoker should be regulated so as to maintain a balance between the load demand and the heat liberated by the fuel. Under such conditions no manual attention to the fuel bed should be required, other than the removal of clinker in stokers which operate on this principle of ash removal.

As in all combustion processes, the maintenance of the correct proportions of air and fuel is essential. It is desirable to supply the minimum amount of air required to properly burn the fuel at the rate of feed.

While there may be only slight variations in the rate at which the coal is being fed due to variations in the size or density of the coal, there may

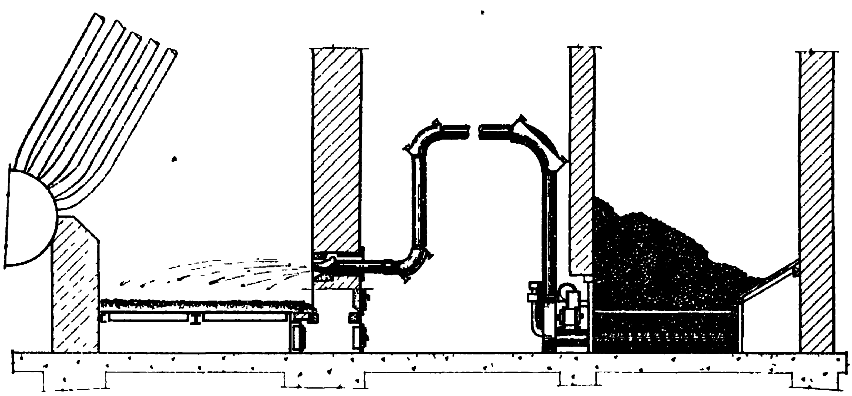


FIG. 10. OVERFEED SPREADER STOKER (PNEUMATIC TYPE)

be wide variations in the rate of air flow as the result of changes in fuel bed resistance. These changes in resistance may be caused by changes in the porosity of the fuel bed due to variations in size or friability of the coal, ash and clinker accumulation, and variations in depth of the fuel bed. Because of this variable fuel bed resistance, many bituminous stokers, even in the smaller domestic sizes, incorporate air controls which automatically compensate for these changes in resistance and maintain a constant air fuel ratio.

It is desirable on most stoker installations to provide automatic draft regulation in order to reduce air infiltration and provide better control during the banking, or off, periods of the stoker. The efficiency of combustion may be determined by analyzing the flue gases as explained in Chapter 8.

Furnace Design

Although there is considerable variation in stoker, boiler, and furnace design, the stoker industry, from long-time experience, has established certain rules for the proportioning of furnaces for domestic, and commercial stokers. The stoker installer, and designer of stoker-fired equipment should give careful consideration to these factors.

The *Stoker Manufacturers Association* has published standard recommendations on setting heights for stokers having capacities up to 1200 lb of coal per hour[2].

The empirical formulas for determining these setting heights are:

For burning rates up to 100 lb coal per hour

$$H = 0.1125\ B + 15.75 \qquad (1)$$

For burning rates from 100 to 1200 lb coal per hour

$$H = 0.03\ B + 24 \qquad (2)$$

where

H = minimum setting height, inches, for steel boilers. For cast-iron boilers height may be $\frac{7}{8} H$.

B = burning rate coal per hour, pounds.

Standards for minimum firebox dimensions and base heights have been formulated by the *Stoker Manufacturers Association* as shown in Fig. 11[3].

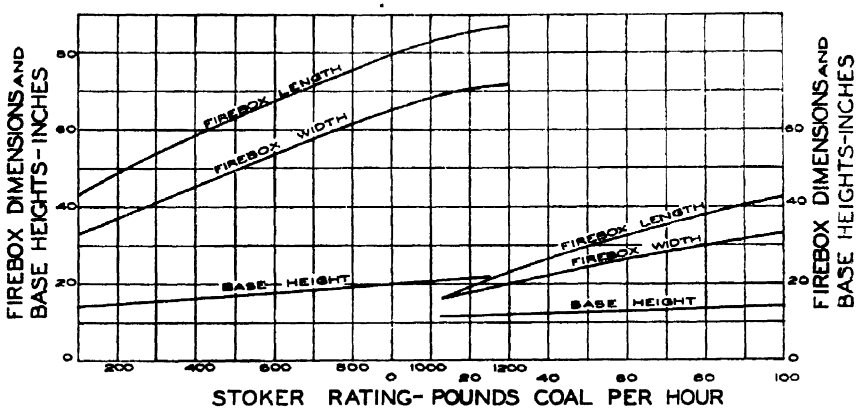


FIG. 11. SUGGESTED MINIMUM FIREBOX DIMENSIONS AND BASE HEIGHTS[a]

[a]For reference in selecting or designing boilers and furnaces for stoker firing. Dimensions shown are for net inside clearance at grate level using coal with heating value of not less than 12,000 Btu per pound Under certain conditions smaller fireboxes will permit satisfactory performance but these dimensions are preferred normal minimums

In considering these recommendations, it should be understood that they show the average recommended minimum. There are many factors affecting the proper application of stokers to various types of boilers and furnaces, and, in certain instances, setting height or firebox dimensions shown in the standards may be modified without impairing performance. Such modification rests with the experience of the installer, or designer, with a particular stoker, the type of fuel used, and the construction of the boiler or furnace.

Rating and Sizing Stokers

The capacity or rating of small underfeed stokers is usually stated as the burning rate in pounds of coal per hour. Codes for establishing uni-

[2]*Stoker Manufacturers Association Manual:* Industry Standards, Recommended Practices, Technical Information. Published by *Stoker Manufacturers Association*, 307 N. Michigan Ave., Chicago 1, Ill

[3]Loc. Cit. Note 2.

form methods of rating anthracite, and bituminous coal stokers have been adopted by the *Stoker Manufacturers Association*[4].

The Association also has adopted a uniform method of selecting stokers that is published in convenient tables and charts[5]. The required capacity of the stoker is calculated as follows:

$$\frac{\text{Load (Btu per hour)}}{\text{Heating value of coal (Btu per pound)} \times \text{over-all efficiency of stoker and boiler or furnace}} = \begin{array}{l}\text{Stoker burning rate}\\ \text{required (pounds of}\\ \text{coal per hour)}\end{array}$$

In determining the total load placed on a stoker-fired boiler by a steam or hot water heating system, a piping and pick-up factor of 1.33 is commonly used in sizing the stoker, but this factor should be increased at times due to unusual conditions.

Controls

The heat delivery from the stoker of the smallest household type to the largest industrial unit can be regulated accurately with fully automatic controls. The smaller heating applications are controlled normally by a thermostat placed in the building to be heated. Limit controls are supplied to prevent excessive temperature or pressure being developed in the furnace or boiler and refueling controls are used to maintain ignition during periods of low heat demand. Automatic low water cut-outs are recommended for use with all automatically-fired steam boilers. (See Chapter 33).

DOMESTIC OIL BURNERS

An oil burner is a mechanical device for producing heat automatically from liquid fuels. Two methods are employed for the preparation of the oil for the combustion process; atomization, and vaporization. The simpler types of burners depend upon the natural chimney draft for supplying the air for combustion. Other burners provide mechanical air supply or a combination of atmospheric, and mechanical. Ignition is accomplished by an electrical spark or hot wire, or by an oil or gas pilot. Some burners utilize a combination of these methods. Continuously operating burners may use manual ignition. Burners of different types operate with luminous or non-luminous flame. Operation may be intermittent, continuous with high-low flame, or continuous with graduated flame.

CLASSIFICATION OF BURNERS

Domestic oil burners may be classified by type of design or operation into the following groups: pressure atomizing or gun, rotary, and vaporizing or pot. These are further classified as mechanical draft, and natural draft.

Pressure Atomizing (Gun Type)

Gun type burners may be divided into two classes, low-pressure, and high-pressure atomization. In the first group, a mixture of oil and primary air is pumped as a spray through the nozzle at a pressure of 2 to 7 lb per square inch. Secondary air is supplied by a fan. Ignition is

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[4]Code for Determination of Rated Capacities of Anthracite Underfeed Stokers, adopted June 1, 1944, and a Code for Determination of Rated Capacities of Bituminous Underfeed Stokers, adopted May 3, 1944 See *Stoker Manufacturers Association Manual.*

[5]Loc. Cit. Note 2.

obtained by means of a high-voltage electric spark used alone, or as primary-ignition for a gas pilot.

The high-pressure atomizing type, illustrated in Fig. 12, is characterized by an air tube, usually horizontal, with oil supply pipe centrally located in the tube and arranged so that a spray of atomized oil is introduced, at about 100 lb per square inch, and mixed in the combustion chamber with the air stream emerging from the air tube. A variety of patented shapes is employed at the end of the air tube to influence the direction and speed of the air and thus the effectiveness of the mixing process.

This type of burner utilizes a fan to supply the air for combustion, and ignition is established by a high-voltage electric spark that may be operative continuously while the burner is running, or just at the beginning of the running period. Gun type burners operate on the intermittent *on-off* principle, and with a luminous flame.

The combustion process is completed in a chamber constructed of

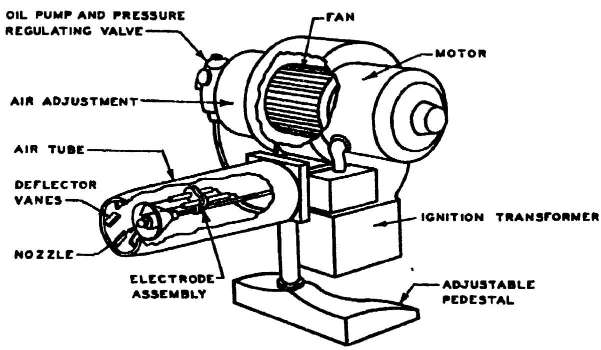

FIG 12. HIGH-PRESSURE ATOMIZING OIL BURNER

refractory material, or stainless steel, this being a part of the in: Pressure-atomizing burners generally use the distillate oils, No. 1, 2 or 3 grade. (See Chapter 8).

Rotary Type

This class of burners may be divided into two groups: vertical, and horizontal. Most of the smaller rotary burners are of the vertical type, and use the lighter distillate oils, No. 1 or 2 grade.

The most distinguishing feature of vertical rotary burners is the principle of flame application. These burners are of two general types: the center flame and wall flame. In the former type (Fig. 13), the oil is atomized by being thrown from the rim of a revolving disc or cup and the flame burns in suspension with a characteristic yellow color. Combustion is supported by means of a bowl-shaped chamber or hearth. The wall flame burner (Fig. 14) differs in that combustion takes place in a ring of stainless steel or refractory material, which is placed around the hearth. Dependent upon combustion adjustment, these burners may operate with either a semi-luminous or non-luminous flame.

Both types of vertical rotary burners are further characterized by their installation within the ash pit of the boiler or furnace. Various types of

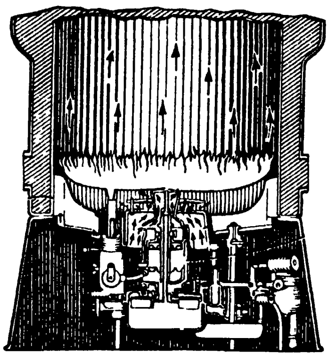

FIG 13. CENTER FLAME VERTICAL ROTARY BURNER

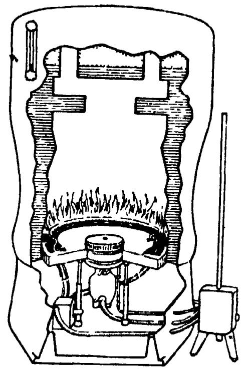

FIG. 14. WALL FLAME VERTICAL ROTARY BURNER

ignition are utilized, gas and electric, either spark or hot wire. The air for combustion is supplied partially by natural draft, and partially by fan effect of the central spinner element.

Horizontal rotary burners are used principally to burn the heavier oils, Nos. 5 and 6 grades, principally in larger commercial and industrial installations, although domestic sizes are available. Such burners are of the mechanical atomizing type, using rotating cups which throw the oil from the edge of the cup at high velocity into the surrounding stream of air delivered by the blower (Fig. 15).

Horizontal rotary burners commonly use a combination electric-gas ignition system, or are lighted manually. Primary air for combustion is supplied by a blower, and secondary air, often introduced through a checkerwork in the combustion chamber, is controlled by chimney draft. These burners operate with a luminous flame, usually on high-low or continuous setting.

In larger installations, burners may be installed in multiple in a common combustion chamber. Because of the high viscosity oils used in these

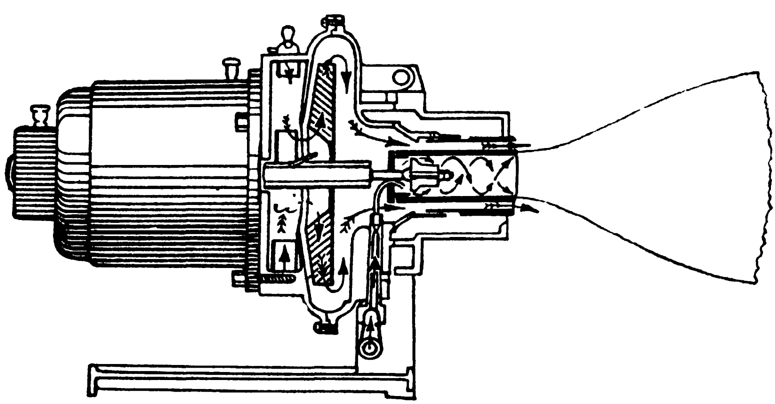

FIG. 15. HORIZONTAL ROTATING CUP OIL BURNER

burners, it is customary to preheat the oil between the tank and the burner. Preheating when delivering from tank car, or truck is often required in cold weather. As with pressure atomizing burners, a refractory combustion chamber, built within the boiler furnace, is a part of the installation.

Vaporizing Burners

This type burner transforms the oil into a combustible vapor by the application of heat from a plate, or cracking chamber. The class may be subdivided into burners supplying combustion air mechanically, and those utilizing the natural chimney draft. Vaporizing pot-type burners are restricted to the use of the lighter distillates.

These burners are designed for continuous or intermittent operation, with manual ignition, and the rate of burning is controlled by a metering valve. Flame may be luminous or non-luminous, dependent upon adjustment. A burner of this type is illustrated in Fig. 16.

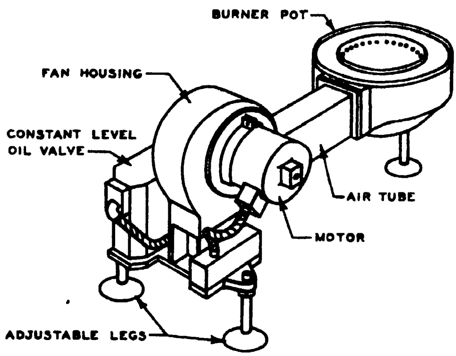

FIG. 16. VAPORIZING POT-TYPE BURNER

Vaporizing burners of the natural draft type are used as the firing device in integral space heaters, water heaters, or furnace units. Only those with full mechanical draft are suitable for conversion installation

Oil-Fired Boiler and Furnace Units

Boilers and furnaces especially designed for oil burners are available. This type of equipment usually has more heating surface, better flue proportions and gas travel, than conventional boilers or furnaces. Some of the better conversion installations, however, equal the unit type in performance.

Operating Requirements for Mechanical Draft Oil Burners

The *U. S. Department of Commerce* in conjunction with the oil burner industry has established commercial standards for automatic mechanical draft oil burners for domestic installations which cover installation requirements and performance tests[6].

[6]Automatic Mechanical Draft Oil Burners Designed for Domestic Installations (*U. S Department of Commerce, National Bureau of Standards*, Commercial Standard No CS75-42). Flue Connected Oil Burning Space Heaters Equipped with Vaporizing Pot Type Burners (*U. S. Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS101-43). Warm-Air Furnaces Equipped with Vaporizing Pot-Type Oil Burners (*U. S. Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS(E)104-43).

Combustion Process

Efficient combustion must produce a clean flame and use a relatively small excess of air, *i.e.*, between 25 and 50 per cent. This can be done only by vaporizing the oil quickly and completely, and mixing it vigorously with air in a combustion chamber hot enough to support the combustion. A vaporizing burner prepares the oil, for combustion, by transforming the liquid fuel to the gaseous state by the application of heat. This is accomplished before the oil vapor mixes with air to any extent and if the air and oil vapor temperatures are high and the fire pot hot, a clear blue flame is produced.

In an atomizing burner the oil is mechanically separated into very fine particles so that the surface exposure of the liquid to the radiant heat of the combustion chamber is vastly increased and vaporization proceeds quickly. The result is the ability to burn more and heavier oil within a given combustion space. Because the air enters the combustion chamber with the liquid fuel particles, mixing, vaporization and burning occur all at once in the same space. This produces a luminous flame. A deficient amount of air is indicated by a dull red or dark orange flame with smoky tips.

An excessive supply of air may produce a brilliant white flame or a short ragged flame with incandescent sparks flashing through the combustion space. While extreme cases may be detected, it is not possible to distinguish, by eye, the effect of the finer adjustment which competent installation requires.

Combustion Adjustments

The present-day oil burner with mechanical oil and air supply, properly installed and equipped with an automatic draft regulator, is capable of maintaining efficient combustion for an appreciable period following the initial adjustments of oil and air. Eventually certain changes may occur, however, that will cause the per cent of excess air to decrease below allowable limits. A decrease in air supply while the oil delivery remains constant or an increase in oil delivery while the air supply remains constant will make the mixture of oil and air too rich for clean combustion. The more efficient the adjustment the more critical it will be. The oil and air supply rates must remain constant.

The following factors may influence the oil delivery rate: (1) changes in oil viscosity due to temperature change or variations in grade of oil delivered, (2) erosion of atomizing nozzle, (3) fluctuations in by-pass relief pressures, and (4) possible variations in methods of atomization. Any change due to partial stoppage of oil delivery will increase the proportion of excess air. This will result in less heat, reduced economy and possibly a complete interruption of service.

The following factors may influence the air supply: (1) changes in combustion draft due to a variety of causes (*i.e.*, changes in chimney draft because of weather changes, seasonal changes, back drafts, failure or inadequacy of automatic draft regulator, use of chimney for other purposes, possible stoppage of the chimney and changes in draft resistance of boiler due to partial stoppage of the flues), and (2) changes in air inlet adjustments at the fan.

Air leakage into the boiler or furnace setting should be reduced to a minimum. The amount of air leakage will be determined by the draft in the combustion chamber. It is important that this draft should be reduced as low as is consistent with the proper disposal of the gases of

combustion. When using mechanical draft burners with average conditions, the combustion chamber draft should not be allowed to exceed 0.02-0.05 in. water. An automatic draft regulator is very helpful in maintaining such values.

Even though a fan is generally used to supply the air for combustion, in most oil burners, the importance of a proper chimney should not be overlooked. The chimney should have sufficient height and size to insure that the draft will be uniform within the limits given above if maximum efficiency throughout the heating season is to be maintained.

Measurement of the Efficiency of Combustion

Since efficient combustion is based upon a clean flame and definite proportions of oil and air employed, it is possible to determine the results by analyzing the combustion gases. It is usually sufficient to analyze only for carbon dioxide (CO_2). A showing of 10 to 12 per cent indicates the best adjustment if the flame is clean. Most of the good installations show from 8 to 10 per cent CO_2. Taking into account the potential hazard of low excess air (high CO_2), a setting to give 10 per cent CO_2 constitutes a reasonable standard for most oil burners.

Combustion Chamber Design

With burners requiring a refractory combustion chamber the size and shape should be in accordance with the manufacturer's instructions. It is important that the chamber shall be as nearly air tight as is possible, except when the particular burner requires a secondary supply of air for combustion.

The atomizing burner is dependent upon the surrounding heated refractory or firebrick surfaces to vaporize the oil and support combustion. Unsatisfactory combustion may be due to inadequate atomization and mixing. A combustion chamber can only compensate for these things to a limited extent. If liquid fuel continually reaches some part of the firebrick surface, a carbon deposit will result. The combustion chamber should enclose a space having a shape similar to the flame but large enough to avoid flame contact. The nearest approach in practice is to have the bottom of the combustion chamber flat, but far enough below the nozzle to avoid flame contact, the sides tapering from the air tube at the same angle as the nozzle spray and the back wall rounded. A plan view of the combustion chamber resembles in shape the outline of the flame. In this way as much firebrick as possible is close to the flame so it may be kept hot. This insures quick vaporization, rapid combustion and better mixing by eliminating dead spaces in the combustion chamber. An overhanging arch at the back of the fire pot is sometimes used to increase the flame travel and give more time for mixing and burning, and sometimes to prevent the gases from going too directly into the boiler flues. When good atomization and vigorous mixing are achieved by the burner, combustion chamber design becomes a less critical matter. Where secondary air is used, combustion chamber design is quite important. When installing some of the vertical rotary burners the manufacturer's instructions must be followed carefully when installing the hearth, as in this class successful performance depends upon this factor.

Boiler Settings

As the volume of space available for combustion is a determining factor in oil consumption, it is general practice to remove grates and extend the combustion chamber downward to include or even exceed the

ash pit volume; in new installations the boiler may be raised to make added volume available. Approximately 1 cu ft of combustion volume should be provided for every developed boiler horsepower, and in this volume from 1.5 to 2.5 lb of oil per hour can properly be burned. This corresponds to an average liberation of about 38,000 Btu per cubic foot per hour. At times much higher fuel rates may be satisfactory. For best results, care should be taken to keep the gas velocity below 40 fps. Where checkerwork of brick is used to provide secondary air, good practice calls for about 1 sq in. of opening for each pound of oil fired per hour. Such checkerwork is best adapted to flat flames, or to conical flames that can be spread over the floor of the combustion chamber. The proper bricking of a large or even medium sized boiler for oil firing is important and frequently it is advisable to consult an authority on this subject. The essential in combustion chamber design is to provide against flame impingement upon either metallic or firebrick surfaces. Manufacturers of oil burners usually have available detailed plans for adapting their burners to various types of boilers, and such information should be utilized.

Controls

Controls for oil burner operation, including devices for the safety and protection of a boiler or furnace, are fully described in Chapter 33.

GAS-FIRED HEATING EQUIPMENT

A gas burner is defined by the *American Gas Association* as "a device for the final conveyance of the gas, or a mixture of gas and air, to the combustion zone." Burners used for domestic heating are of the atmospheric injection, yellow flame, or power burner types.

The use of gas has resulted in the production of a number of types of domestic gas heating appliances, and systems. These may be classified in types designed for central heating plants, and those for unit application. Gas-designed units and conversion burners are available for the several kinds of central systems. Unit heaters, space heaters and circulators may be had for installation in the space being heated.

Central Heating Systems

Boilers and furnaces specially designed for gas-firing incorporate design features for obtaining maximum efficiency and performance. Small flue passes to secure good heat transfer, the use of materials resistant to the corrosive effects of products of combustion, and draft hoods are notable features. Control equipment includes gas pressure regulators, thermostatic pilots and limit controls designed to protect the appliance and to insure safety of operation. A boiler designed for gas-burning is illustrated in Fig. 17.

Conversion burners are usually complete burner and control units designed for installation in existing boilers and furnaces. Burner heads are of circular or rectangular shape in order to fit in space available. The control equipment is generally the same as for gas boilers and furnaces. Various baffles made of clay radiants or metal are used for the purpose of guiding the products of combustion along the heating surface in the firebox or flues. Automatic air dampers are supplied on many models to prevent flow of air into the firebox when the burner is not operating.

One form of central heating system is the warm air floor furnace[7]. The

[7]Gas Floor Furnaces, Gravity Circulating Type (*U S Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS99-42).

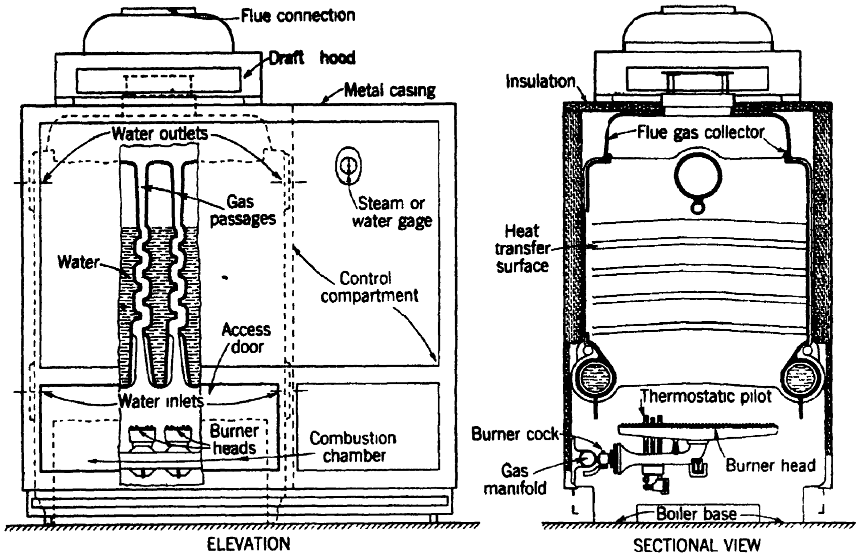

FIG. 17. GAS-FIRED BOILER

use of these furnaces is adaptable to mild climates. They are used for heating first floors, or where heat is required in only one or two rooms. A number may be used to provide heat for the entire building where all rooms are on the ground floor, thus giving the heating system flexibility. With the usual type the register is installed in the floor, the heating element and gas piping being suspended below.

Unit Type Heaters

Space heaters may be used for auxiliary heating, but in many cases are installed for furnishing heat to entire buildings. With the exception of wall heaters, they are portable. Although it is desirable to connect space heaters with solid piping, use of flexible tubing is frequently resorted to, particularly in the portable types. Where flexible tubing is used, a gas shut-off on the heater is not permitted and only *American Gas Association* approved tubing should be used.

Parlor heaters or *circulators* are usually of the cabinet type. They heat the room entirely by convection, *i.e.*, the cold air of the room is drawn in near the base, passes up inside the jacket around a heating section, and out of the heater at, or near, the top. These heaters cause a continuous circulation of the air in the room during the time they are in operation. The burners are located in the base at the bottom of an enclosed combustion chamber. The products of combustion pass around baffles within the heating element, and out the flue at the back near the top. They are well adapted not only for residence room heating but also for stores and offices.

Radiant heaters give off considerable portion of their heat in the form of radiant energy emitted by an incandescent refractory that is heated by a Bunsen flame. They are made in numerous shapes and designs and in sizes ranging from two to fourteen or more radiants. An atmospheric burner is supported near the center of the base. Others have a group of small atmospheric burners supported on a manifold attached to the base.

Most radiant heaters are portable; however, there are also types which are encased in a jacket that fits into the wall with a grilled front.

Gas-fired steam and hot water radiators are other types of room heating appliances. They are made in a large variety of shapes and sizes and are similar in appearance to the ordinary steam or hot water radiator. A separate combustion chamber is provided in the base of each radiator and is usually fitted with a one-piece burner. They may be secured in either the vented or unvented types, and with steam pressure, thermostatic or room temperature controls.

Warm air radiators are similar in appearance to steam or hot water radiators. They are usually constructed of sheet metal hollow sections. The products of combustion circulate through the sections and are discharged from a flue or into the room, depending upon whether the radiator is of the vented or unvented type.

Unit heaters are used extensively for heating large spaces such as stores, garages, and factories. These heaters consist of a burner, heat exchanger, fan for distributing the air, draft hood, thermostatic pilot and controls for burners and fan. They are usually mounted in an elevated position from which the heated air is directed downward by louvres. Some unit heaters are suspended from the ceiling, and others are free-standing floor units of the heat tower type.

Combustion Process and Adjustments

Most domestic gas burners are of the atmospheric injection (Bunsen) type in which primary air is introduced, and mixed with the gas in the throat of the mixing tube. A ratio of about 3 parts primary air to 1 part gas for manufactured gas, and a 5½ to 1 ratio for natural gas, are generally used as theoretical values. The amount of excess air required in practice depends upon several factors, notably; uniformity of air distribution and mixing, direction of gas travel from burner, and the height and temperature of combustion chamber.

Secondary air is drawn into gas appliances by natural draft. As with other fuels, excess secondary air constitutes a loss, and should be reduced to a proper minimum, which usually cannot be less than 25 to 35 per cent if the appliance is to meet *A.S.A.* approval. Yellow flame burners depend upon secondary air, alone, for combustion.

The flame produced by atmospheric injection burners is non-luminous. Air shutter adjustments for manufactured gas should be made by closing the air shutter until yellow flame tips appear and then by opening the air shutter to a final position at which the yellow tips just disappear. This type of flame obtains ready ignition from port to port and also favors quiet flame extinction. When burning natural gas the air adjustment is generally made to secure as blue a flame as obtainable.

Little difficulty should be had in maintaining efficient combustion when burning gas. The fuel supply is normally held to close limits of variation in pressure and calorific value and the rate of heat supply is nominally constant. Because the force necessary to introduce the fuel into the combustion chamber is an inherent factor of the fuel, no draft by the chimney is required for this purpose. The use of a draft hood insures the maintenance of constant low draft condition in the combustion chamber with a resultant stability of air supply. A draft hood is also helpful in controlling the amount of excess air and preventing back drafts that might extinguish the flame. (See Chapter 8.)

Due to the use of draft hoods and gas pressure regulators both the input and combustion conditions of gas appliances are maintained quite uniform until deposits of dirt, corrosion, or scale accumulate in the air inlet openings, burner ports or on the heating surface. Periodic cleaning is, therefore, necessary to keep any gas appliance in proper operating condition.

Measurement of the Efficiency of Combustion

The efficiency of combustion may be judged from the percentage of carbon dioxide (CO_2), oxygen (O_2) and carbon monoxide (CO) in the flue gases. The CO_2 and O_2 may be obtained by means of an Orsat apparatus but the CO must be determined by more accurate equipment. It is customary to use simple indicators to determine whether CO is present and to make adjustments of the appliances to reduce the CO below 4/100 of one per cent before continuing tests in which the CO_2 and O_2 can then be found by use of the Orsat apparatus. Since the ultimate CO_2 for any gas depends on the total hydrogen content the quality of the combustion should not be judged from the value of the CO_2 in the flue gas without reference to the ultimate CO_2 obtainable. Practical values of CO_2 will usually be from 8 to 14 per cent depending on the gas used.

Ratings for Gas Appliances

Input rating for a gas appliance is established by demonstrating that the appliance can meet the Approval Requirements of the *A.S.A.* The tests are conducted at the *A.G.A. Testing Laboratories.* Output rating is determined from the approved input and an average efficiency stated in the Approval Requirements and represents the heat available at the outlet of the appliance.

Sizing Gas-Fired Heating Plants

Although gas-burning equipment usually is completely automatic, maintaining the temperature of rooms at a predetermined figure, there are some manually controlled installations. In order to overcome effectively the starting load and losses in piping, a manually-controlled gas boiler should have an output as much as 100 per cent greater than the equivalent standard radiation which it is expected to serve.

Boilers under thermostatic control, however, are not subject to such severe pick-up loads and consequently, it is possible to use a lower selection factor. For a gas-fired boiler or furnace under thermostatic control a factor of 20 per cent is usually sufficient for pick-up allowance.

The factor to be allowed for loss of heat from piping, will vary somewhat, the proportionate amount of piping installed being greater for small installations than for large ones. Selection factors to be added to the installed radiation under thermostatic control are discussed in Chapter 12.

Appliances used for heating with gas should bear the approval seal of the *A.G.A. Testing Laboratories* on the manufacturer's nameplate, together with the official input and output ratings. It is not permissible to operate a gas heating unit above its stated rating. It may be necessary to operate below this rating if the appliance is installed at elevations more than 3,000 ft above sea level.

Installations should be made in accordance with recommendations shown in the publications of the *American Gas Association.*

Controls

Temperature controls for gas burners are described in Chapter 33. Some central heating plants are equipped with push-button or other manual control. The main gas valve may be of either the snap action or throttling type. Automatic electric ignition is available for gas heating equipment.

FUEL BURNING RATES

The burning rate for automatic fuel burning devices is determined by the gross heat output required of the boiler, or furnace, to carry the net heating load plus allowances for system losses, and pick-up. General values for these allowances previously have been noted. Detailed information for piping and pick-up allowances for steam, and hot water systems is given in Chapter 12. Similar data for warm air systems are given in Chapters 18 and 19.

When the gross output, operating efficiency, and heat value of the fuel are known, the required rate of burning can be determined. Figs. 18, 19 and 20 afford a convenient method for finding rates of burning for the several fuels.

For a given efficiency, the rate of fuel burning is directly proportional to the load and therefore these charts can be extended by moving the decimal points the same number of digits in both vertical and horizontal scales.

The correct fuel burning rate can be determined directly from the several charts for oil or gas burning installations, as these customarily operate on a strictly intermittent basis. These fuel burning devices usually introduce the fuel at a single fixed rate during the *on* periods and this rate should be sufficient to carry the gross load. In the case of coal stokers, which are usually capable of variable rates of firing, it is desirable to operate at as low a rate as weather conditions will permit, but the maximum firing rate of the stoker should be sufficient to carry the gross load. This rate may be determined by the same method as used for oil or gas.

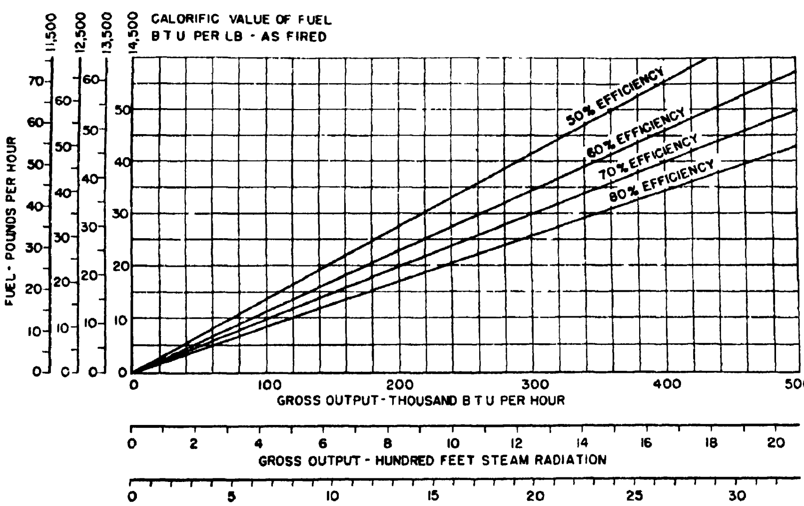

FIG. 18. COAL FUEL BURNING RATE CHART

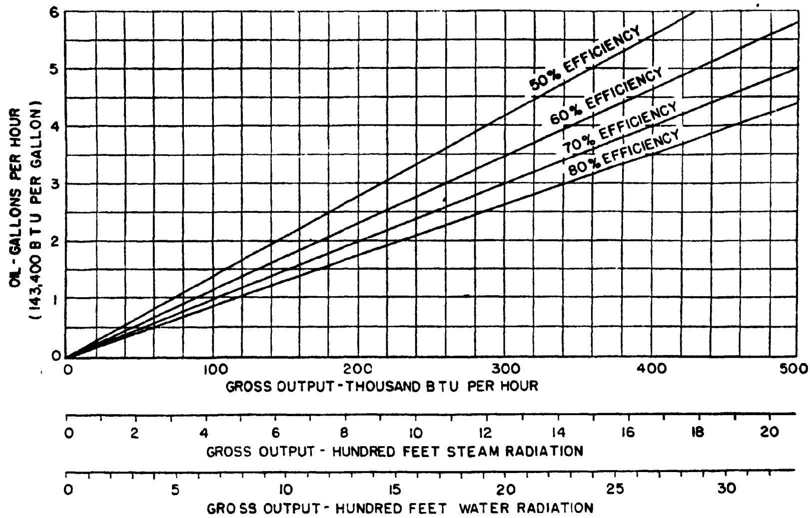

FIG. 19. OIL FUEL BURNING RATE CHART[a]

[a]This chart is based upon No. 3 oil having a heat content of 143,400 Btu per gallon If other grades of oil are used multiply the value obtained from this chart by the following factors No 1 oil (139,000 Btu per gallon) 1.032; No. 2 oil (141,000 Btu per gallon) 1.017; No 4 oil (144,500 Btu per gallon) 0 992; No 5 oil (146,000 Btu per gallon) 0.982; and No 6 oil (150,000 Btu per gallon) 0 956.

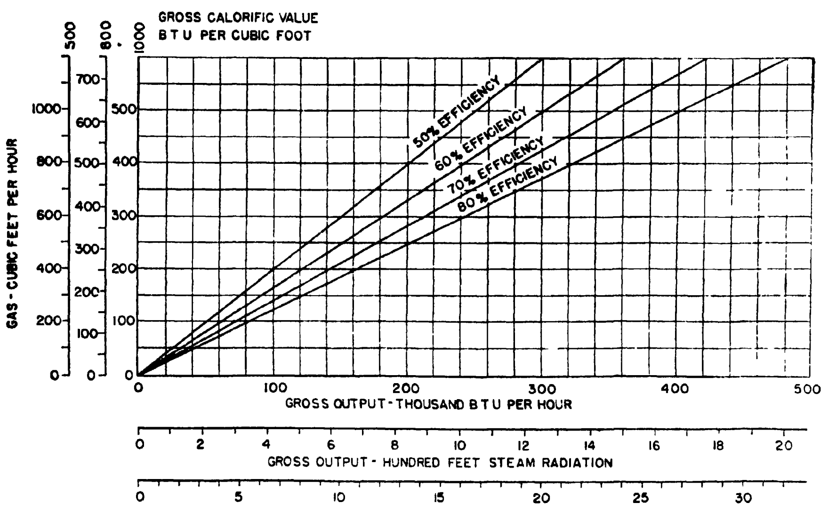

FIG. 20. GAS FUEL BURNING RATE CHART

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CHAPTER 11

Estimating Fuel Consumption

Fuel Consumption Records, Calculated Heat Loss Estimation Method, Degree-Day Method, Unit Fuel Consumption per Degree-Day, Degree-Day as an Operating Unit, Maximum Demands and Load Factors

MANY methods are in use for estimating in advance of actual operation the anticipated heat or fuel consumption of heating plants over long or short periods. With suitable modification in procedure these same general methods are frequently useful in checking the degree of effectiveness with which heat or fuel is utilized during plant operation.

In applying any of these estimating methods to the consumption of a particular building plant it should be noted that (*a*) reliable records of past heat or fuel consumptions of the building under consideration will usually produce more trustworthy estimates of future consumptions than will any data obtained by averages or from other similar buildings; (*b*) where no past records exist useful data can sometimes be obtained from records of similar buildings with similar plants *in the same locality;* (*c*) records of consumption, which are averages from many types of plants in many types of buildings in various localities, can produce no better than an average estimate which may be far from accurate; (*d*) estimates based on computed heat losses without the benefit of operating data are wholly dependent on how well the computation represents the actual facts.

Estimates based on computed heat losses alone are frequently the only ones possible to obtain, especially where new equipment is put into unusual buildings and there is a scarcity of records and an absence of experience data. Such estimates also have to be made where direct information is not obtainable as, for example, if a survey is being made without the assistance or knowledge of the building operator and thus without information as to the actual consumption. Estimates of this kind are also useful in some cases where a *relative* standard of performance is desired to serve as a base of comparisons in a campaign of fuel utilization.

In interpreting and evaluating heat or fuel consumption estimates as well as in their preparation, it is well to realize that any estimating method used will produce a more reliable result over a long period operation than over a short period. Nearly all of the methods in common use will give trustworthy results over a full *annual* heating season, and in some cases such estimates will prove consistent within themselves for monthly periods. As the period of the estimate is shortened there is more chance that some factor not allowed for in the estimating method will become controlling and thus give discrepant and even ridiculous results.

Of the various estimating methods in use attention is directed in this discussion to but two as they are illustrative of all, viz: (1) calculated heat loss method, and (2) degree-day method.

CALCULATED HEAT LOSS METHOD

This method is theoretical and assumes constant temperatures for very definite hours each day throughout the entire heating season. It does not take into account factors which are difficult to evaluate such as opening of windows, abnormal heating of the building, poor heating systems, winter heat gains, such as sun effect, and many others.

In order to apply this method the hourly heat loss from the building under maximum load, or design condition, is computed following the principles discussed in Chapters 4 and 5 and the method described and illustrated in Chapter 6.

In some cases, however, depending on the presence of interior partitions, the computed heat loss is modified when used for estimating the heat or fuel consumption. If the building has no interior walls or partitions then, by the method of Chapters 5 and 6, the infiltration losses are calculated by using only half the total window crack. In such a building the calculated loss need not be modified in order to prepare heat or fuel estimates by this method. Where the building does contain interior walls or partitions instead of using as the calculated heat loss (H) which is equal to the sum of the transmission losses (H_t) and the infiltration losses (H_i), it is more desirable to let $H = H_t + \frac{H_i}{2}$.

In predicting fuel consumption for heating a building by the Calculated Heat Loss Method, the general equation is:

$$F = \frac{H\,(t - t_a)\,N}{E\,(t_d - t_o)\,C} \tag{1}$$

where

F = quantity of fuel or energy required (in the units in which C is expressed).

H = calculated heat loss, Btu per hour, during the design hour, based on t_o and t_d $\left(\text{generally } H = H_t + H_i \text{ but may on occasion equal } H_t + \frac{H_i}{2}\right)$.

t = average inside temperature maintained during heating period, degrees Fahrenheit.

t_a = average outside temperature through estimate period, degrees Fahrenheit (for cities with an Oct. 1-May 1 heating season, see Table 1, Chapter 6).

t_d = inside design temperature, degrees Fahrenheit (usually 70 F).

t_o = outside design temperature, degrees Fahrenheit (see Table 1 in Chapter 6).

N = number of heating hours in estimate period (for an Oct. 1—May 1 heating season, 212 days × 24 hr = 5088).

E = efficiency of utilization of the fuel over the period, expressed as a decimal; not the efficiency at peak or rated load condition.

C = heating value of one unit of fuel or energy.

Although the assumption of an Oct. 1-May 1 heating season is reasonably accurate in the well-populated New York-Chicago zone, it is not valid as far north as Minneapolis nor farther south than Washington, D. C. and St. Louis. Consequently, it is suggested that allowance be made for this variation, especially in the far north or southern cities.

Example 1. A residence in Chicago is to be heated to 70 F from 6 A.M. to 10 P.M. and 55 F from 10 P.M. to 6 A.M. The calculated hourly heat loss is 120,000 Btu per hour based on 70 F inside at −10 F outside. If the building is to be heated by metered steam, how many pounds would be required during an average heating season?

Solution. The heating value of steam may be taken as 1000 Btu per pound, and since it is purchased steam, the efficiency can be assumed as 100 per cent. From Table 1, Chapter 6, t_a = 36.4 F. The average inside temperature is:

$$\frac{(16 \times 70) + (8 \times 55)}{24} = 65 \text{ F.}$$

Substituting in Equation 1:

$$F = \frac{120{,}000\,(65 - 36.4)\,5088}{1.00\,[70 - (-10)]\,1000} = 218{,}275 \text{ lb.}$$

Example 2. How much would the fuel cost to heat the building in Example 1 during an average heating season with coal at \$8 per ton and with a calorific value of 11,000 Btu per pound, assuming that the seasonal efficiency of the plant was 55 per cent?

Solution. Substituting in Equation 1: $F = \dfrac{120{,}000\,(65 - 36.4)\,5088}{0.55\,[70 - (-10)]\,11{,}000} = 36{,}079$ lb = 18 tons, which, at \$8 per ton, costs \$144.

Several time-saving procedures have been devised for quickly estimating the heat consumption of one and two-story residences in order that fuel estimates can be predicted more quickly from Equation 1. A graphical method of calculating heat losses has been developed[1] which makes possible a quick solution if the gross wall, ceiling, or floor areas and respective transmission coefficients are known.

The Federal Housing Administration has originated a short-cut formula for residential heat loss determinations which makes use of the floor area and three selected transmission coefficients. Equation 2 is for a one-story residence and Equation 3 is intended for two-story structures.

$$H_1 = A\,(0.45 + U_w + U_c + U_f)\,(t_d - t_o) \tag{2}$$

$$H_2 = A\,(0.45 + 1.2\,U_w + 0.5\,U_c + 0.5\,U_f)\,(t_d - t_o) \tag{3}$$

where

H_1 = heat loss from one-story residence, Btu per hour.
H_2 = heat loss from two-story residence, Btu per hour.
A = floor area, square feet.
U_w = coefficient of transmission for outside wall
U_c = coefficient transmission for ceiling from air in rooms to air in attic space (for ventilated attics).
U_f = coefficient of transmission for floor from air in rooms to air in basement.
t_d = inside design temperature, degrees Fahrenheit
t_o = outside design temperature, degrees Fahrenheit.

Both the graphical method and short-cut formula have been found to give accurate and consistent results for the average residence, but if precise estimates are required, the procedure outlined in Chapter 6 should be used.

Example 3 What will be the estimated fuel cost per year of heating a building with gas, assuming that the calculated hourly heat loss is 92,000 Btu based on 0 F, which includes 26,000 Btu for infiltration? The design temperatures are 0 F and 72 F. The normal heating season is 210 days, and the average outside temperature during the heating season is 36.4 F. The seasonal efficiency will be 75 per cent. The heating plant will be thermostatically controlled, and a temperature of 55 F will be maintained from 11 P.M. to 7 A.M. Assume that the price of gas is 7 cents per 100,000 Btu of fuel consumption, and disregard the loss of heat through open windows and doors.

Solution. The average hourly temperature is:

$$t_a = \frac{(72 \times 16) + (55 \times 8)}{24} = 66.3 \text{ F.}$$

The maximum hourly heat loss will be:

$$H = 92{,}000 - \frac{26{,}000}{2} = 79{,}000 \text{ Btu.}$$

$$M = \frac{79{,}000\,(66.3 - 36.4) \times 24 \times 210}{100{,}000 \times 0.75 \times (72 - 0)} = 2204.6 \text{ hundred thousand Btu.}$$

2204.6 × \$0.07 = \$154.32 = estimated fuel cost per year of heating building.

In the case of gravity warm air heating installations, the load is usually expressed in square inches of leader pipe. This can be converted into hourly heat loss by multiplying by the factors in Table 1.

[1]Graphical Method of Calculating Heat Losses, by Paul D. Close (A.S.H.V E. TRANSACTIONS, Vol. 49, 1943, p. 345).

TABLE 1. HEAT CARRYING CAPACITY OF GRAVITY WARM AIR FURNACE ROUND LEADER PIPES

175 F Register Temperature

LEADER PIPE	BTU PER HOUR AT DESIGN CONDITIONS PER SQ IN. OF LEADER PIPE
First floor	111
Second floor	167
Third floor	200

Example 4. What would be the total gas consumption over a full heating season of a gas-fired gravity warm air furnace designed according to the Code[2], and with four 12 in. and two 8 in. round leaders to the first floor and six 10 in. leaders to the second floor, if the gas has a heating value of 500 Btu per cubic foot, the plant operates at a 70 per cent seasonal efficiency and is designed to maintain an average inside temperature of 65 F when it is 10 F outside in a city where the average outside temperature is 45 F and the heating season is 5088 hr long?

Solution. The area of the round leaders is: 12 in., 113 sq in.; 10 in., 79 sq in.; and 8 in., 50 sq in. From Table 1 the total Btu transmitted is:

First Floor: $[(4 \times 113) + (2 \times 50)] \times 111 = 61{,}272$ Btu per hour.
Second Floor: $(6 \times 79) \times 167 = 79{,}158$ Btu per hour.

Total 140,430 Btu per hour.

Substituting this total heat loss value as H in Equation 1 gives:

$$F = \frac{140{,}430\ (65 - 45)\ 5088}{0.70\ (70 - 10)\ 500} = 680{,}483 \text{ cu ft gas.}$$

DEGREE-DAY METHOD

This method is based on consumption data which have been taken from buildings in operation, and the results computed on a degree-day basis. While this method may not be as theoretically correct as the Calculated Heat Loss Method, it is considered by many to be of more value for practical use.

The amount of heat required by a building depends upon the outdoor temperature, if other variables are eliminated. Theoretically it is proportional to the difference between the outdoor and indoor temperatures. Some years ago the *American Gas Association*[3] determined from experiment in the heating of residences that the gas consumption varied directly as the difference between 65 F and the outside temperature. In other words, on a day when the temperature was 20 deg below 65 F, twice as much gas was consumed as on a day when the temperature was 10 deg below 65 F. For any one day there exists as many degree-days as there are degrees Fahrenheit difference in temperature between the mean temperature for the day and 65 F when the mean temperature is less than 65 F.

The normal or average number of degree-days which have occurred over a long period of years, and such averages, by months, on a 65 F basis are given for various United States and Canadian cities in Table 2. The United States values were computed from daily mean temperatures

[2]Standard Gravity Code for the Design and Installation of Gravity Warm Air Heating Systems (11th edition), and the Technical Code for the Design and Installation of Mechanical Warm Air Heating Systems, may be obtained from the *National Warm Air Heating and Air Conditioning Association*, 145 Public Square, Cleveland, Ohio

[3]See Industrial Gas Series, House Heating, (*third edition*) published by the *American Gas Association*.

TABLE 2. NORMAL DEGREE-DAYS FOR CITIES IN THE UNITED STATES AND CANADA[a]

STATE	CITY	JAN.	FEB.	MAR	APR	MAY	JUNE	JULY	AUG	SEPT	OCT.	NOV.	DEC.	Total
Ala.	Birmingham	586	502	318	133	23	1	0	1	10	111	351	582	2618
	Mobile	392	317	178	54	3	0	0	0	1	45	203	374	1567
Ariz.	Phoenix	404	261	152	47	7	0	0	0	0	18	168	389	1446
Ark.	Fort Smith	761	623	393	159	34	1	0	0	12	130	409	708	3230
	Little Rock	700	584	370	148	30	1	0	0	10	121	382	659	3005
Calif.	Los Angeles	276	232	209	158	104	27	1	0	5	43	110	225	1390
	San Francisco	464	340	314	269	252	196	198	180	122	140	242	426	3143
Colo	Denver	1026	900	792	523	271	62	9	8	124	415	721	1012	5863
	Grand Junction	1230	887	667	381	152	22	1	1	60	352	745	1149	5647
Conn.	New Haven	1103	1024	848	521	222	46	3	12	89	341	660	1010	5879
D.C.	Washington	929	842	637	345	104	14	0	2	43	254	558	870	4598
Fla.	Jacksonville	295	241	131	39	3	0	0	0	0	24	142	286	1161
Ga.	Atlanta	662	568	389	173	32	2	0	1	12	130	393	640	3002
	Savannah	395	335	200	68	6	1	0	0	1	46	209	386	1647
Idaho	Boise	1065	840	680	441	248	87	10	17	135	394	717	1025	5659
Ill.	Chicago	1219	1084	871	536	260	67	7	8	84	334	709	1108	6287
	Springfield	1145	988	732	382	130	15	1	3	64	290	666	1047	5463
Ind.	Evansville	942	826	588	289	82	5	0	1	32	207	540	875	4387
	Indianapolis	1104	982	751	414	155	22	2	4	65	299	663	1026	5487
Iowa	Des Moines	1326	1143	849	449	165	27	2	6	99	357	768	1200	6391
	Sioux City	1410	1216	913	488	194	37	3	10	124	405	841	1268	6909
Kan.	Dodge City	1050	878	669	354	134	16	1	3	58	276	644	994	5077
	Topeka	1104	932	666	331	112	11	0	2	54	258	623	1008	5101
Ky.	Lexington	964	862	650	353	123	14	1	3	47	258	600	916	4791
	Louisville	931	824	595	302	92	7	0	1	34	218	549	875	4428
La.	New Orleans	319	251	130	32	1	0	0	0	0	23	147	305	1208
	Shreveport	522	421	243	83	10	0	0	0	4	71	274	499	2127
Me.	Eastport	1353	1242	1090	778	532	297	159	148	274	528	831	1219	8451
	Portland	1296	1184	1003	668	375	135	29	46	179	459	792	1172	7338
Md	Baltimore	922	844	650	347	98	12	0	2	34	230	531	852	4522
Mass.	Boston	1101	1027	852	538	248	66	8	16	98	338	651	1000	5943
Mich.	Detroit[b]	1228	1141	943	569	252	56	7	15	109	387	753	1120	6580
	Marquette	1471	1358	1208	801	493	220	87	102	257	556	927	1306	8786
Minn.	Duluth	1726	1504	1267	809	514	232	82	100	289	636	1066	1541	9766
	Minneapolis	1597	1379	1082	580	255	59	8	24	163	484	943	1415	7989
Miss.	Vicksburg	498	413	239	85	9	0	0	0	5	77	268	479	2073
Mo.	Kansas City	1083	921	657	327	106	11	0	2	50	242	599	986	4984
	St. Louis	999	865	616	305	88	7	0	1	36	218	558	917	4610
	Springfield	971	835	599	306	105	11	1	2	45	232	560	900	4567
Mont.	Havre	1548	1374	1109	623	338	129	28	56	275	604	1010	1380	8474
Neb.	Lincoln	1254	1070	798	411	161	23	1	6	84	328	732	1142	6010
	Omaha	1285	1098	813	414	152	21	1	4	84	326	741	1163	6102
Nev.	Winnemucca	1126	887	763	539	320	109	11	23	190	502	802	1099	6371
N. H.	Concord	1339	1212	1005	633	310	98	19	46	189	492	827	1221	7391
N. J.	Atlantic City	944	893	762	489	215	37	1	2	40	250	552	864	5049
N M.	Santa Fe	1094	892	786	544	297	62	10	15	129	451	772	1072	6124
N. Y.	Albany	1276	1174	955	549	221	44	4	14	116	410	756	1139	6658
	Buffalo[b]	1220	1168	1010	672	352	90	15	24	125	410	747	1102	6935
	New York	1025	958	785	467	176	29	1	4	51	276	600	934	5306
N. C.	Raleigh	699	618	436	213	47	5	0	1	17	155	417	673	3281
	Wilmington	528	481	328	147	25	2	0	0	5	92	310	514	2432
N. D.	Bismarck	1720	1478	1187	650	335	104	20	44	240	605	1059	1527	8969
Ohio	Cincinnati	1009	905	681	374	132	16	1	3	52	274	613	953	5013
	Cleveland	1141	1074	891	558	255	56	8	14	92	354	688	1040	6171
	Columbus	1082	982	762	436	167	26	2	6	68	318	674	1013	5536
Okla.	Oklahoma City	850	696	459	204	56	3	0	0	22	156	461	791	3698
Ore.	Baker	1221	994	839	600	411	210	56	73	261	540	849	1165	7219
	Portland	776	622	530	370	241	106	29	29	109	298	540	729	4379
Pa.	Philadelphia	957	885	696	379	117	16	0	2	36	236	548	877	4749
	Pittsburgh[b]	1043	971	767	448	170	29	3	7	69	324	656	979	5466
S. C.	Charleston	450	386	245	84	7	1	0	0	1	48	227	421	1870
	Columbia	568	488	312	129	18	1	0	0	6	98	330	554	2504

TABLE 2. NORMAL DEGREE-DAYS FOR CITIES IN THE UNITED STATES AND CANADA[a] (CONCLUDED)

State Province	City	Jan.	Feb.	Mar.	Apr	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec	Total
S. D.	Huron	1586	1366	1044	579	264	66	9	21	174	507	966	1415	7997
	Rapid City	1288	1152	980	607	333	104	15	29	188	503	843	1183	7225
Tenn.	Knoxville	774	669	478	229	55	3	0	0	20	189	497	751	3665
	Memphis	709	605	389	159	32	1	0	0	13	127	383	660	3078
	Nashville	785	681	476	222	54	2	0	0	19	170	470	741	3620
Tex.	El Paso	611	433	288	107	15	1	0	0	6	90	369	618	2538
	Fort Worth	582	470	269	100	16	1	0	0	5	76	288	549	2356
	Houston	366	279	144	31	3	0	0	0	1	29	164	343	1360
	San Antonio	382	290	148	38	4	0	0	0	0	32	169	361	1424
Utah	Modena	1191	944	811	569	336	93	7	11	156	503	833	1151	6605
	Salt Lake City	1084	868	708	450	232	62	4	5	95	377	712	1040	5637
Vt.	Burlington	1444	1327	1117	679	330	98	22	47	196	510	866	1294	7930
Va.	Lynchburg	829	735	545	290	82	12	1	2	37	232	524	793	4082
	Norfolk	709	654	492	257	65	6	0	0	9	132	397	664	3385
	Richmond	812	727	548	281	74	9	0	1	27	199	491	775	3944
Wash.	Seattle	765	644	595	440	304	162	71	71	174	372	558	708	4864
	Spokane	1136	935	748	488	282	113	21	37	185	483	816	1061	6305
W. Va.	Elkins	1030	950	774	492	233	59	15	23	114	403	724	997	5814
	Parkersburg	975	888	671	371	129	170	1	3	55	287	619	922	5091
Wis.	Green Bay	1498	1337	1099	663	325	89	17	39	176	495	891	1327	7956
	LaCrosse	1500	1291	1001	533	222	50	7	22	154	459	865	1338	7442
	Milwaukee	1329	1181	969	621	340	100	13	18	120	409	788	1198	7086
Wyo.	Cheyenne	1188	1070	991	726	455	160	42	48	247	592	880	1147	7549
	Lander	1423	1198	1000	674	406	150	28	44	263	630	1020	1401	8237
Alta.	Calgary	1674	1428	1240	750	496	270	124	186	450	744	1170	1395	9,927
	Edmonton	1829	1512	1302	720	434	270	124	186	450	713	1230	1519	10,289
B. C.	Vancouver	899	756	713	510	341	180	62	31	270	496	660	837	5,755
Man.	Winnipeg	2139	1820	1581	810	465	90	......	62	270	744	1320	1829	11,130
N.B.	Moncton	1519	1428	1178	810	465	210	......	93	300	620	930	1333	8,886
N. S.	Halifax	1302	1176	1085	780	496	210	......	......	210	496	780	1147	7,682
Ont.	Ottawa	1674	1484	1271	690	279	30	......	.....	210	589	990	1457	8,674
	Port Arthur	1829	1624	1426	900	558	240	62	86	360	713	1140	1550	10,488
	Toronto	1333	1204	1209	720	372	60	......	......	180	558	870	1209	7,715
P.E.I.	Charlottetown	1178	1120	1209	870	529	210	......	......	600	558	870	1240	8,384
Que.	Montreal	1581	1428	1209	720	310	......	......	......	180	558	960	1395	8,341
	Quebec	1705	1484	1333	870	434	120	......	31	270	651	1050	1519	9,467
Sask.	Saskatoon	2108	1820	1581	810	465	210	62	155	450	806	1290	1736	11,493

[a]Computed from daily mean temperatures recorded by U S Weather Bureau over a 43-year period from 1899 to 1941 Data for Canadian cities abstracted from *Heating & Ventilating*, October, 1939 The National Joint Committee on Weather Statistics, in cooperation with the U. S Weather Bureau, is preparing revised Degree-Day Normals which will be available in 1945

[b]Data for these cities and possibly other localities are based on readings taken at more than one official Weather Station during the 43-year period of analysis and are subject to local verification, as these figures are being examined for possible revision by the U. S Weather Bureau

recorded by the Weather Bureau over a 43-year period[4] from 1899 to 1941. The number of degree-days for a calendar day of a given year was obtained by taking the difference between 65 F and the mean temperature determined from a reading of the maximum and minimum thermometers for a particular locality. The daily normal was established by taking an average of the 43 daily degree-day figures. These daily values were then added to obtain a monthly normal, and the yearly or seasonal degree-day figure was established by taking a summation of the monthly values. In general, attempts to apply the degree-day method to fuel consumption over a period of less than a month are of questionable value.

[4]Received from U. S. Weather Bureau Sept. 1943

If the degree-days occurring each day are totaled for a reasonably long period, the fuel consumption during that period as compared with another period will be in direct proportion to the number of degree-days in the two periods. Consequently, for a given installation, the fuel consumption can be calculated in terms of fuel used per degree-day for any sufficiently long period and compared with similar ratios for other periods to determine the relative operating efficiencies with the outside temperature variable eliminated.

Studies made by the *National District Heating Association*[5] of the metered steam consumption of 163 buildings located in 22 different cities and served with steam from a district heating company substantiate the fact that the 65 F base originally chosen by the gas industry is approximately correct.

Formula for Degree-Day Method

The general equation for calculating the probable fuel consumption by the degree-day method is:

$$F = U \times N \times D \tag{4}$$

where

F = fuel consumption for the estimate period.

U = unit fuel consumption, or quantity of fuel used per degree-day per ***building load unit***.

N = number of ***building load units*** (when available use calculated heat loss instead of actual amount of radiation installed).

D = number of degree-days for the estimate period.

Values of N depend on the particular building for which the estimate is being prepared and must be found by surveying plans, by observation, or by measurement of the building. Values of U for use in this equation are the unit fuel consumptions per degree-day and are obtained as a result of the collection of operating information. Certain of this information is presented later but before referring to these data attention is directed to the nature of the unit.

Unit Fuel Consumptions per Degree-Day

The quantity of fuel used per degree-day in a given heating plant can be reduced to a unit basis in terms of quantity of fuel or steam per degree-day per square foot of radiation, per cubic foot of heated building space, or per thousand Btu hourly heat loss at design conditions. A less frequently used basis is quantity of fuel per degree-day per square foot of floor area. In fact any convenient unit can be used to relate the consumption to the degree-day and to the building.

The choice of these units requires explanation and some discrimination and judgment. The use of heated space in preference to the gross building cubage is obviously more accurate for this purpose. The gross cubage includes the outer walls and certain percentages of attic and basement space which are usually unheated. The net heated space is usually about 80 per cent of the gross volume and can be calculated from the latter if it cannot be measured. The cubical content is not considered accurate as a basis of comparison due to differences in types of construction, exposure, and ratio of exposed area to cubical contents.

The calculated heat loss or its equivalent square foot of calculated radiator surface may be used as the unit. The use of the unit equivalent direct radiation is of questionable value when referring to heat transfer

[5]Report of Commercial Relations Committee, *Proceedings, National District Heating Association*, 1932

surfaces as applied to warm air furnace or central air conditioning systems. In view of all these considerations it is believed that the unit based on *thousands of Btu of hourly calculated heat loss for the design hour* is probably the most desirable although the one most widely used seems to be *units of fuel per degree-day per square foot of equivalent direct radiator surface.* The equivalent heating load for the hot water supply is not included in the latter unit, but it generally includes the piping load.

Since this unit is the one most widely used at present the unit fuel consumptions given in succeeding paragraphs of this chapter make use of this unit to a considerable extent, although it should be understood that most of these units of consumption can be transposed as desired.

Estimating Gas Consumption

Values of the Unit Fuel Consumption Constant (U) for gas are given in Table 3 for various gas heating values, and different types and sizes of heating plants. They are based on an inside design temperature of 70 F and an outside design temperature of 0 F and apply only to these conditions. For other design conditions corrections must be made as given in Table 6.

The factors in Table 3, as corrected if necessary, are satisfactory for regions having 3500 to 6500 degree-days per heating season. In regions with less than 3500 degree-days the unit gas consumption is higher than given; where over 6500, the unit is less than given. Ten per cent addition or deduction in these cases is recommended by *A.G.A.* publications. Estimates for industrial buildings where low inside temperatures are maintained cannot be made from this table.

For gas heating values other than those given in Table 3, simply interpolate or extrapolate. It will also be noted that Table 3 applies only to small installations. In general the larger the installation the smaller the unit gas consumption becomes and the values in the table should be used with care, if at all, in large gas-burning installations.

Example 5. Estimate the gas required to heat a building located in Chicago, Ill., which has 6287 degree-days and a gas heating value of 800 Btu per cubic foot. The calculated heating surface requirements are 1000 sq ft of hot water radiation based on design temperature of −10 F and 70 F.

Solution. From Table 3, the fuel consumption for a design temperature of 0 F with 800 Btu gas is found to be 0.085 cu ft of gas per degree-day per square foot of hot water radiation. From Table 6, the correction factor is 0.875 for −10 F outside design temperature, hence $0.875 \times 0.085 = 0.07438$. By Equation 4,

$$F = 0.07438 \times 1000 \times 6287 = 467{,}000 \text{ cu ft.}$$

Estimating Oil Consumption

Unit fuel consumption factors for oil, similar to those for gas in Table 3 are given in Table 4. The factors in Table 4 apply only to an inside design temperature of 70 F and an outside design temperature of 0 F. For other outside design temperatures, the constants in Table 4 must be multiplied by the values in Table 6 as explained under Estimating Gas Consumption.

Values given in Table 4 assume the use of oil with a heating value of 140,000 Btu per gallon. For other heating values, multiply the values in Table 4 by the ratio of 140,000 divided by the heating value per gallon of fuel being used.

Example 6. Estimate the seasonal oil consumption of an oil-fired boiler in a building located in Minneapolis having a calculated heat loss of 192,000 Btu per hour, burning

TABLE 3. UNIT FUEL CONSUMPTION CONSTANTS (U) FOR GAS[a]

Based on 0 F Outside Temperature, 70 F Inside Temperature, and 8-Hour Reduction to 60 F.

Heating Value of Gas Btu per Cu Ft	Hot Water: Cu Ft Gas per Degree-Day per Sq Ft Radiator			Steam: Cu Ft Gas per Degree-Day per Sq Ft Radiator			Warm Air: Cu Ft Gas per Degree-Day per 1000 Btu Hourly Design Heat Loss	
	Up to 500 Sq Ft	500 to 1200 Sq Ft	Over 1200 Sq Ft	Up to 500 Sq Ft	300 to 700 Sq Ft	Over 700 Sq Ft	Gravity	Fan Systems
500	0.142	0.135	0.128	0.242	0 231	0.220	0 855	0 820
535	0 132	0.126	0.120	0.226	0 215	0 206	0 800	0 766
800	0 089	0 085	0 081	0.151	0 144	0 137	0 534	0 513
1000	0 071	0.068	0.065	0 121	0 115	0 110	0.428	0.410
1 Therm = 100,000 Btu	Gas Consumption in Therms per Degree-Day							
	0 000708	0 000675	0.000642	0.00121	0 00115	0 00110	0 00428	0 00409

[a]Abstracted from Comfort Heating. *American Gas Association*, 1938

TABLE 4. UNIT FUEL CONSUMPTION[a] CONSTANTS (U) FOR OIL[b]

Based on 0 F Outside Temperature, 70 F Inside Temperature, and 8-Hour Reduction to 55 F.

Unit	Efficiency in Per Cent: 40	50	60	70	80
Gal Oil per Sq Ft Steam Radiator	0.00172	0.00137	0 00114	0 00098	0 00086
Gal Oil per Sq Ft Hot Water Radiator	0 00108	0.00086	0.00072	0 00062	0.00054
Gal Oil per 1000 Btu per Hour Heat Loss	0.00715	0 00571	0.00476	0 00409	0 00358

[a]Based on a heating value of 140,000 Btu per gallon.

[b]Abstracted by permission from *Degree-Day Handbook* (Second Edition, 1937), by C Strock and C. H. B. Hotchkiss.

TABLE 5. UNIT FUEL CONSUMPTION[a] CONSTANTS (U) FOR COAL[b]

Based on 0 F Outside Temperature, 70 F Inside Temperature, and 8-Hour Reduction to 55 F

Unit	Efficiency in Per Cent: 40	50	60	70	80
Lb Coal per Sq Ft Steam Radiator	0.0200	0.0160	0.0133	0.0114	0.0100
Lb Coal per Sq Ft Hot Water Radiator	0.0125	0 0100	0.0084	0.0072	0.0063
Lb Coal per 1000 Btu per Hour Heat Loss	0.0825	0.0666	0.0550	0.0471	0.0412

[a]Based on a heating value of 12,000 Btu per pound.

[b]Abstracted by permission from *Degree-Day Handbook*, (Second Edition, 1937), by C. Strock and C. H. B. Hotchkiss.

144,000 Btu per gallon oil and operating at a seasonal efficiency of 60 per cent. The outside design temperature for Minneapolis is −20 F, and the inside design temperature is 70 F.

Solution. From Table 4, under 60 per cent efficiency and opposite the bottom column, the value of U is found to be 0.00476 gal per 1000 Btu hourly heat loss for 0 F outside temperature.

The correction factor for −20 F outside design temperature from Table 6 is 0.778. Solving, $0.778 \times 0.00476 = 0.00370$. Making a further correction for the heating value:

$0.0037 \times \frac{140,000}{144,000} = 0.0036$ gal per 1000 Btu per hour calculated heat loss per degree-day.

From Table 2, the normal degree-days for Minneapolis is 7989 Since U is expressed in 1000 Btu, N is equal to 192. Substituting in Equation 4:

$$F = 0.0036 \times 7989 \times 192 = 5525 \text{ gal.}$$

Estimating Coal or Coke Consumption

Coal or coke consumption estimates are made in exactly the same procedure as for oil. Values of U are given in Table 5 which only apply to inside design temperatures of 70 F and an outside design temperature of 0 F. A correction must be made for other conditions by use of the multiplying factors in Table 6. Data in Table 5 are based on 12,000 Btu per pound coal and for other heating values of coal they must be multiplied by the ratio of 12,000 divided by the heating value of fuel used.

TABLE 6. CORRECTION FACTORS FOR OUTSIDE DESIGN TEMPERATURES[a]

OUTSIDE DESIGN TEMP DEG F	INSIDE DESIGN TEMP DEG F	MULTIPLY VALUES IN TABLES 3, 4 AND 5 BY
−20	70	0.778
−10	70	0 875
0	70	1 000
+10	70	1.167
+20	70	1.400

[a] The multipliers in Table 6, which are high for mild climates and low for cold regions are not in error as might appear The unit figures in Tables 3, 4 and 5 are per square foot of radiator or thousand Btu heat loss per degree-day For equivalent buildings and heating seasons, those in warm climates have lower design heat losses and smaller radiator quantities than those in cold cities Consequently, the *unit* figure, in quantity of fuel per square foot of radiator per degree-day, is larger for warm localities than for colder regions Since the northern cities have more radiator surface per given building and a higher seasonal degree-day total than cities in the south, the total fuel per season will be larger for the northern city.

Example 7. A building in Marquette, Mich., has an hourly heat loss at design conditions of 240,000 Btu per hour. Based on an inside design temperature of 70 F and an outside design temperature of −20 F, what will be the estimated normal seasonal coal consumption for heating if 12,000 Btu per pound fuel is burned at a 50 per cent seasonal efficiency, and what part of the total will be used during November, December, and January?

Solution. From Table 5, U is 0.0666 lb of coal per 1000 Btu per hour heat loss. Correcting for the outside design temperature of −20 F from Table 6, the value of U is $0.778 \times 0.0666 = 0.0518$. From Table 2, D is 8786 and from the problem, N is 240.

Substituting in Equation 4:

$$F = 0.0518 \times 240 \times 8786 = 109,200 \text{ lb.}$$

Fuel used over any period is, according to the theory of the degree-day, proportional to the number of degree-days during the period. From Table 2, the average numbers of degree-days for November, December, and January in Marquette are 927, 1306, and 1471, a total of 3704. The yearly total is 8786, so that during these three months the estimated consumption is:

$$\frac{3704}{8786} \times 109,200 = 46,200 \text{ lb.}$$

Estimating Steam Consumption

In estimating steam consumption the efficiency is generally assumed at 100 per cent. If for low pressure steam an average heating value of 1000 Btu per pound of steam is used no correction is necessary. In comparing values from different cities, correction should be made for design temperature (see Table 6) when the unit figures are in terms of square foot of radiator or 1000 Btu per hour calculated heat loss, but not when the values are in terms of building volume or floor space.

Consideration has been given to the difference in steam utilization of different types of buildings and Table 7 shows actual average units for

TABLE 7. STEAM CONSUMPTION OF BUILDINGS WITH VARIOUS TYPES OF OCCUPANCY[a]

Type of Building	No. Bldgs.	Average Volume Heated Space 1000 Cu Ft	Steam for Heating: Lb per DD per 1000 Cu Ft	Average Hours of Occupancy
Office	334	2160	0.685	12.1
Office and Bank	49	3000	0.577	13.1
Office and Printing	8	1895	1.230	17.7
Office and Theater	7	4950	0.412	12.9
Office and Stores or Shops	26	1615	0.617	13.2
Bank	16	806	0.786	11.7
Department Store	63	3400	0.385	11.1
Stores	73	310	0.624	10.4
Loft	63	865	0.588	10.0
Warehouse	24	2230	0 459	9.4
Hotel and Club	73	1795	0 990	22.3
Apartment or Residence	51	1425	0.962	21.8
Theater	22	1240	0 482	12.9
Garage	13	1540	0.202	21.4
Manufacturing	19	1350	0.808	9.5
Church	9	656	0.532	7.9
Hospital	4	3306	1.194	22.0
School	8	1115	0.592	11.5
Municipal or Federal	15	3215	0.587	15.6
Lodge, Gym, Hall or Auditorium	12	880	0.390	12.4
Miscellaneous	7	1387	0.479	21.4
Total or Average	896	1890	0 651	13.4

[a]Principles of Economical Heating, *National Association of Building Owners and Managers*

these various types. These figures were obtained from operating results in 896 buildings located in all sections of the United States. Being averages, and for small groups in each type, the figures may need considerable modification to allow for local variations. It should be especially noted that the steam used for heating hot water is not included in the values given in Table 7.

Example 8. A store in Philadelphia with a heating system designed to maintain 70 F inside in 0 F weather has 250,000 cu ft of heated space. What would be the estimated average yearly steam consumption of purchased steam for heating?

Solution. According to Table 7, a store would use 0.624 lb of steam per degree-day per 1000 cu ft heated space. From Table 2, Philadelphia has 4749 degree-days per normal year. Inserting in Equation 4:

$$F = 0.624 \times 250 \times 4749 = 740{,}000 \text{ lb of steam.}$$

Degree-Day as an Operating Unit

The use of the degree-day as a fuel *predicting* unit, explained in the foregoing, is perhaps the lesser of its two most used applications. It is also widely used as a unit for eliminating the weather (temperature) variable for existing plants from month to month or year to year. The degree-days given in Table 2 are *normals* or averages. For operating data

TABLE 8. HEAT CONSUMPTION RECORD FOR COMPARISON

HEATING SEASON 1942-43

COL. 1	COL. 2	COL. 3	COL. 4	COL. 5	COL. 6	COL. 7
	TOTAL CONSUMPTION	CONSUMPTION FOR HEATING	AVG MEAN TEMP	DEG DAYS 65 F BASE	LB/DEG DAY	LB/DEG DAY/ M CU FT
Sept	337,500	170,500	65	146	1,170	0.575
Oct	834,200	667,200	53	339	1,966	0.970
Nov	1,446,600	1,279,600	44	641	1,990	0.982
Dec	2,176,400	2,009,400	25	1,233	1,630	0.804
Jan	2,332,200	2,165,200	22	1,297	1,670	0.822
Feb	2,131,100	1,964,100	28	1,106	1,775	0.888
Mar	2,021,900	1,854,900	31	1,032	1,799	0.885
Apr	1,241,500	1,074,500	43	647	1,660	0.818
May	672,500	505,500	55	303	1,670	0.822
June	258,600	91,600	...	50	1,830	0.905
July	188,400	...	...	...	...	...
Aug	180,100	...	...	...	...	
Total	13,821,000	...	...	...	...	...

HEATING SEASON 1943-44

COL. 1	COL. 2	COL. 3	COL. 4	COL. 5	COL. 6	COL. 7
	TOTAL CONSUMPTION	CONSUMPTION FOR HEATING	AVG. MEAN TEMP.	DEG DAYS 65 F BASE	LB/DEG DAY	LB/DEG DAY/ M CU FT
Sept	330,200	146,200	61	167	875	0.431
Oct	887,100	703,100	52	410	1,718	0.845
Nov	1,525,200	1,341,200	39	812	1,653	0.815
Dec	2,045,500	1,861,500	28	1,120	1,660	0.817
Jan	1,933,400	1,749,400	30	1,044	1,670	0.825
Feb	1,990,200	1,806,200	30	1,111	1,624	0.800
Mar	1,984,100	1,800,100	31	1,021	1,760	0.868
Apr.	...	...	...	...	...	...
May		...	...	...	...	...
June		...	...	...	...	...
July		...	...	...	...	...
Aug	...	...	...	...	...	...
Total	...	...	...	...	...	...

If, for example, the heat consumption in March, 1943, is compared with that in March, 1944, it will be found that in the latter the steam consumption is 1799 − 1760 = 39 lb less which is a decrease of 2 2 per cent.

the actual number occurring for a given month and year in a city under consideration is recorded. Since fuel consumption is proportional to the number of degree-days, plant operators frequently compute each month, the fuel burned per degree-day by the heating plant. The resulting unit figure, by eliminating the outside temperature variable, indicates whether the operating efficiency of the plant is above or below the previous month or year.

The figures in Table 8 illustrate a typical example of a method of

using the degree-day for making heating comparisons for one building for two consecutive heating seasons. The heat quantity figures inserted are pounds of steam, but a similar comparison could be made using pounds of coal, gallons of oil, or cubic feet of gas.

For such a comparison, a two-year record is often used, as shown in Table 8. The year under consideration may then be compared, month by month, with the previous year. Column 3, Consumption for Heating, would be used if the same fuel is used for heating and process steam. Some reasonable figure must be assumed for the process requirement and should be deducted from the amount shown in column 2. This would leave in column 3 only the fuel chargeable to heating. The degree-day values in column 5 are obtainable from the local Weather Bureau. Figures in column 6 are obtained by dividing corresponding values in column 3 by the degree days in column 5. The heating index in column 6 is, then, a figure of heat consumption, corrected for outdoor temperature, and should be relatively constant month by month. Column 7 in

Table 9. Building Load Factors and Demands of Some Detroit Buildings

Building Classification	Load Factor	Lb of Demand per Hour per Sq Ft of Equivalent Installed Radiator Surface
Clubs and Lodges	0.318	0.184
Hotels	0.316	0.207
Printing	0.287	0.217
Offices	0.263	0.209
Apartments	0.255	0.225
Retail Stores	0.238	0.182
Auto Sales and Service	0.223	0.248
Banks	0.203	0.158
Churches	0.158	0.152
Department Stores	0.138	0.145
Theaters	0.126	0.151

Table 8 may be used if the heat consumption is to be compared on a building volume basis with average values shown in Table 7.

MAXIMUM DEMANDS AND LOAD FACTORS

In one form of district heating rates, a portion of the charge is based upon the maximum demand of the building. The maximum demand may be measured in several different ways. It may be taken as the instantaneous peak or as the rate of use during any specified interval. One method is to take the average of the three highest hours during the winter. These figures are available for a number of buildings in Detroit, as shown in Table 9[6].

These maximum demands were measured by an attachment on the condensation meter and therefore represent the amounts of condensation passed through the meter in the highest hours, rather than the true rate at which steam is supplied. There might be slight differences in these two quantities due to time lag and to storage of condensate in the system, but wherever this has been investigated it has been found to be negligible.

The load factor of a building is the ratio of the average load to the maximum load and is an index of the utilization. Thus, in Table 9, the

[6]The Heat Requirements of Buildings, by J. H. Walker and G. H Tuttle (A.S H.V.E. Transactions, Vol. 41, 1935, p. 171).

theaters, operating for short hours, have a load factor of 0.126 as compared with the figure of 0.318 for clubs and lodges.

SEASONAL EFFICIENCY

The task of predicting fuel consumption within reasonably accurate limits is a simple one where sufficient experience data are available for the fuel in question. Such data can be analyzed to the point where average unit factors can be determined and expressed in such terms as, for example, average gallons of oil actually burned per square foot of calculated steam radiator surface per degree-day. The unit U can be inserted directly in Equation 2 without reference to efficiency. Such experience factors are available for gas (see Table 3) and for district steam (Table 7), but not for coal or oil.

Since values of U are not available for oil or coal, an assumed seasonal efficiency E must be used. Selection of a value for this E must be made with caution, for its use implies a meaning not commonly associated with the word efficiency and consequently is frequently misleading.

The input of heat to a building consists not only of the energy in the fuel but that from occupants, the sun, appliances, processes, and all other sources. In many cases these make up, over a period, an important percentage of the total heat required, and if they are not taken into account a calculation of *efficiency* can show a figure over 100 per cent.

For this and other reasons the actual seasonal efficiency is a difficult thing to determine. Published data are widely scattered and insufficient. From the available published material it is found that the seasonal efficiency varies over a wide range, depending on the fuel used, and it varies widely even for a given fuel. For example, in a recent survey of 30 houses in one locality there was found a variation of from 45 to 75 per cent in the utilization efficiency depending on the fuel[7].

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[7]Heat Losses and Efficiencies of Fuels in Residential Heating, by R A. Sherman and R. C. Cross (A S H V E TRANSACTIONS, Vol 43, 1937, p 185)

CHAPTER 12

Heating Boilers

Construction, Types of Heating Boilers, Boiler Design Considerations, Testing and Rating Codes, Boiler Efficiency, Rating of Boilers, Selection of Boilers, Space Limitations, Connections and Fittings, Erection, Operation and Maintenance

STEAM and hot water boilers for low pressure heating are built in a wide variety of types and sizes, many of which are illustrated in the *Catalog Data Section*. They are made of steel or cast-iron.

CONSTRUCTION

The only code governing the construction of low-pressure heating steel and cast-iron boilers that has gained recognition on a national basis is the *A.S.M.E.* Boiler Construction Code for Low Pressure Heating Boilers. Many states and municipalities have their own codes which apply locally but these are usually patterned after the *A.S.M.E.* Code.

The maximum allowable working pressures are limited by the *A.S.M.E.* Code to 15 psi for steam and 30 psi for hot water heating boilers. Hot water boilers may be used for higher working pressures for heating purposes or for hot water supply when designed and tested for the higher pressure.

TYPES OF HEATING BOILERS

Heating boilers are classified in a number of different ways, such as:

(*a*) According to materials of construction. These are steel and cast-iron. Very few non-ferrous boilers are made.

(*b*) According to the fuels for which the boilers are designed. These are coal, hand fired or stoker fired; oil; gas; or wood. Some boilers are designed specifically for one fuel but many boilers are designed for more than one fuel.

(*c*) According to the specific purpose or application for which the boiler is used, such as space heating or domestic hot water supply.

(*d*) According to the design or construction of the boiler such as, sectional, round, fire-tube, water-tube, magazine feed, Scotch, etc

Cast-Iron Boilers

Cast-iron boilers are generally classified as:

(*a*) Square or rectangular boilers with vertical sections and rectangular grates commonly known as sectional boilers.

(*b*) Round boilers with horizontal pancake sections and circular grates

Cast-iron boilers are usually shipped in sections and assembled at the place of installation. In the majority of boilers the sections are assembled with push nipples and tie rods. Many sectional boilers are provided with large push nipples at top to permit the circulation of water between adjacent sections at both the water line and bottom of the boiler, which is necessary to enable the use of an indirect water heater with the boiler for summer-winter hot water supply. Round and sectional boilers may be increased in size by the addition of sections and corresponding plate work.

Capacities of cast-iron boilers range generally from capacities required for small residences up to about 12,000 sq ft of steam radiation. There

are a few boilers made with capacities up to 18,000 sq ft of steam radiation. For larger loads, boilers must be installed in multiple.

Steel Boilers

Steel boilers may be of the fire-tube type, in which the gases of combustion pass through the tubes and the boiler water circulates around them, or of the water-tube type, in which the gases circulate around the tubes and the water passes through them.

Either the fire-tube or water-tube type may be designed with integral water jacketed furnaces or arranged for refractory lined brick or refractory lined jacketed furnaces. Those with integral water jacketed furnaces are called portable firebox boilers and are the most commonly used type. They are usually shipped in one piece, ready for piping. Refractory furnaces are usually installed in refractory lined furnace boilers after they are set in place.

Capacities of steel boilers range from those required for small residences up to about 35,000 sq ft of steam radiation.

Boilers for Different Fuels

Either cast-iron or steel boilers may be designed to burn one kind of fuel only or various kinds. The present trend in the design of boilers for hand firing is, however, to design them to be adaptable for ready conversion to an efficient operation with oil, gas or stoker firing after the boiler has been installed.

In recent years, many boilers have been developed for oil firing exclusively, some for use with various makes or types of oil burners and some for use with one burner only which is shipped with the boiler and controls as a complete unit. There has been no common pattern of design followed in these boilers.

Boilers designed for gas firing exclusively have assumed a well defined individuality. They are usually of cast-iron sectional construction with a number of independent burners placed beneath the section. (See Chapter 10.)

Very few boilers have been designed for stoker firing exclusively.

Boilers for Special Applications

One of these is known as the *magazine feed boiler* developed for the burning of small sizes of anthracite and coke and has a large fuel carrying capacity which results in longer firing periods than would be the case with the standard types burning coal of buckwheat size. Special attention must be given to proper chimney sizes and connections when installing in order to insure adequate draft.

Boilers for *hot water supply* are classified as direct, if the water heated passes through the boiler, and as indirect, if the water heated does not come in contact with the water or steam in the boiler.

Direct heaters are built to operate at the pressures found in city supply mains and are tested at pressures from 200 to 300 lb per square inch. The life of direct heaters depends almost entirely on the scale-forming properties of the water supplied. If water temperatures are maintained below 140 F the life of the heater will be much longer than if higher temperatures are used, owing to decreased scale formation and minimized corrosion below 140 F. Direct water heaters in some cases are designed to burn refuse and garbage.

Indirect heaters generally consist of steam boilers in connection with

heat exchangers of the coil or tube types which transmit the heat from the steam to the water. This type of installation has the following advantages:

1. The boiler operates at low pressure.
2. The boiler is protected from scale and corrosion.
3. The scale is formed in the heat exchanger in which the parts to which the scale is attached can be cleaned or replaced. The accumulation of scale does not affect efficiency although it will affect the capacity of the heat exchanger.
4. Discoloration of water may be prevented if the water supply comes in contact with only non-ferrous metal.

Where a steam or a forced circulation hot water heating system is installed, the domestic hot water may be heated by an indirect heater attached to the boiler. For most satisfactory performance in the steam system, this heater is placed just below the water line of the boiler. In a forced circulation hot water system, it should be located as high as possible with respect to the boiler.

BOILER DESIGN CONSIDERATIONS

Furnace Design

Good efficiency and proper boiler performance are dependent on correct furnace design. There must be sufficient volume for burning the particular fuel which is used, and means to obtain a thorough mixing of air and gases at a high temperature and at a velocity low enough to permit complete combustion of all the volatiles. For hand fired boilers, the furnace volume should be large enough to hold sufficient fuel for reasonably long firing periods. (See Chapters 8 and 10.)

Heating Surface

Boiler heating surface is that portion of the surface of the heat transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other side. Heating surface on which the fire shines is known as *direct* or radiant surface and that in contact with hot gases only, as *indirect* or convection surface. The amount of heating surface, its distribution and the temperatures on either side thereof influence the capacity of any boiler.

Direct heating surface is more valuable than indirect per square foot because it is subjected to a higher temperature and also, in the case of solid fuel, because it is in position to receive the full radiant energy of the fuel bed.

The effectiveness of the heating surface depends on its cleanliness, its location in the boiler, and the shape of the gas passages. The area of the gas passages must not be so small as to cause excessive resistance to the flow of gases where natural draft is employed. Inserting baffles so that the heating surface is arranged in series with respect to the gas flow increases boiler efficiency and reduces stack temperature, but increases the draft loss through the boiler.

Heat Transfer Rate

Practical heat transfer rates in heating boilers expressed in Btu absorbed per square foot of surface per hour will average about 3300 for hand fired boilers and 4000 for mechanically fired boilers when operating at *design load*. When mechanically fired boilers are operating at *maximum load* as defined in this chapter under heading *Selection of Boilers*, these values will run between 5000 and 6000. Boilers operating under favorable

conditions at these heat transfer rates will give exit gas temperatures that are considered consistent with good practice, although there are boilers which have high efficiencies and also operate at higher transmission rates.

TESTING AND RATING CODES

The Society has adopted four solid fuel testing codes, a solid fuel rating code and an oil fuel testing code.

A.S.H.V.E. Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers—Codes 1 and 2—(Revision of June, 1929)[1], are intended to provide a method for conducting and reporting tests to determine heat efficiency and performance characteristics.

A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers—Code No. 3—(Edition of 1929)[1] is intended for use with A.S.H.V.E. Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers[2]. The object of this test code is to specify the tests to be conducted and to provide a method for conducting and reporting tests to determine the efficiencies and performance of the boiler.

The A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel[3], (Adopted June, 1932), is intended to provide a standard method for conducting and reporting tests to determine the heating efficiency and performance characteristics when oil fuel is used with steam heating boilers.

In 1938 the Society adopted a Standard Code for Testing Stoker-Fired Steam Heating Boilers[4], (Adopted June, 1938), which is intended to provide a test method for determining the efficiency and performance characteristics of any stoker and boiler combination burning any type of solid fuel such as anthracite or bituminous coal.

The *Steel Heating Boiler Institute* has adopted a method for the rating of low pressure boilers based on their physical characteristics and expressed in square feet of steam or water radiation or in Btu per hour as given in Table 1. The detailed requirements of this code were outlined in Chapter 13 of THE GUIDE 1939.

The *Institute of Boiler and Radiator Manufacturers* has adopted a method of rating cast-iron heating boilers based upon performance obtained under tests. This code[5] now applies to sectional boilers having a grate width of 41 in. or less. Eventually the Institute intends to include all sizes of cast-iron boilers in this program of testing and rating.

The *American Gas Association* has adopted a method of rating gas designed boilers based upon performance under tests. This is described in Approval Requirements for Central Heating Gas Appliances.

The *Heating, Piping and Air Conditioning Contractors National Association* has adopted a method of rating boilers based on their physical characteristics for those boilers that are not rated in accordance with the *S.H.B.I.* or $I=B=R$ Codes. Ratings are expressed on a Net Load basis in square feet of steam radiation.

[1]See A S H.V E. TRANSACTIONS, Vol. 35, 1929, pp. 322 and 332.
[2]See A.S H.V.E TRANSACTIONS, Vol. 36, 1930, p. 42.
[3]See A.S H.V E TRANSACTIONS, Vol. 37, 1931, p. 23.
[4]See A S H V E. TRANSACTIONS, Vol. 44, 1938, p. 366.
[5]$I=B=R$ Testing and Rating Code for Low Pressure Heating Boilers (*Institute of Boiler and Radiator Manufacturers*)

TABLE 1. STEEL HEATING BOILER STANDARD RATINGS[a]

Hand-Fired Rating						Mechanically-Fired Rating				
Catalog			Net Load	Heating Surface Sq Ft	Grate Area Sq Ft	Catalog			Net Load	Furnace Volume, Oil, Gas or Bituminous Coal Cu Ft
Steam Radiation Sq Ft	Water Radiation Sq Ft	Btu per Hour in Thousands	Steam Radiation Sq Ft			Steam Radiation Sq Ft	Water Radiation Sq Ft	Btu per Hour in Thousands	Steam Radiation Sq Ft	
1,800	2,880	432	1,389	129	7.9	2,190	3,500	525	1,695	15.7
2,200	3,520	528	1,702	158	8.9	2,680	4,280	643	2,089	19.2
2,600	4,160	624	2,020	186	9.7	3,160	5,050	758	2,461	22.6
3,000	4,800	720	2,335	215	10.5	3,650	5,840	876	2,853	26.1
3,500	5,600	840	2,732	250	11.4	4,250	6,800	1,020	3,335	30.4
4,000	6,400	960	3,135	286	12.2	4,860	7,770	1,166	3,830	34.8
4,500	7,200	1,080	3,540	322	13.4	5,470	8,750	1,312	4,330	39.1
5,000	8,000	1,200	3,945	358	14.5	6,080	9,720	1,459	4,834	43.5
6,000	9,600	1,440	4,770	429	16.4	7,290	11,660	1,749	5,850	52.1
7,000	11,200	1,680	5,608	500	18.1	8,500	13,600	2,040	6,885	60.8
8,500	13,600	2,040	6,885	608	20.5	10,330	16,520	2,479	8,490	73.8
10,000	16,000	2,400	8,197	715	22.5	12,150	19,440	2,916	10,125	86.8
12,500	20,000	3,000	10,417	893	25.6	15,180	24,280	3,643	12,650	108.5
15,000	24,000	3,600	12,500	1,072	28.4	18,220	29,150	4,372	15,183	130.2
17,500	28,000	4,200	14,584	1,250	30.9	21,250	34,000	5,100	17,708	151.8
20,000	32,000	4,800	16,667	1,429	33.2	24,290	38,860	5,829	20,242	173.5
25,000	40,000	6,000	20,834	1,786	37.4	30,360	48,570	7,286	25,300	216.9
30,000	48,000	7,200	25,000	2,143	41.2	36,430	58,280	8,743	30,359	260.3
35,000	56,000	8,400	29,167	2,500	44.7	42,500	68,000	10,200	35,417	303.6

[a]Adopted by the *Steel Heating Boiler Institute* in cooperation with the *Bureau of Standards, United States Department of Commerce Simplified Practice Recommendations R 157-35.*

BOILER EFFICIENCY

The term *efficiency* as used for guarantees of boiler performance is usually construed as follows:

1. *Solid Fuels.* The efficiency of the *boiler* alone is the ratio of the heat absorbed by the water and steam in the boiler per pound of combustible burned on the grate to the calorific value of 1 lb of combustible as fired. The *combined* efficiency of *boiler, furnace and grate* is the ratio of the heat absorbed by the water and steam in the boiler per pound of fuel as fired to the calorific value of 1 lb of fuel as fired.

2. *Liquid and Gaseous Fuels.* The *combined* efficiency of boiler, furnace and burner is the ratio of the heat absorbed by the water and steam in the boiler per pound or cubic foot of fuel to the calorific value of 1 lb or cubic foot of fuel respectively.

The following efficiencies apply to current designs of boilers operated under favorable conditions at their gross output ratings. Some older boilers designed primarily for hand firing may have lower efficiencies when automatically fired.

Anthracite, hand fired	60 to 75 per cent
Bituminous Coal, hand fired	50 to 65 per cent
Stoker fired	60 to 75 per cent
Oil and Gas fired	70 to 80 per cent

Higher efficiencies for hand fired bituminous coal may be obtained by careful firing of either a regular or a smokeless boiler.

RATING OF BOILERS

In referring to boiler rating it is necessary to know the basis on which the rating has been established in order to understand the exact meaning of the term. The following example will illustrate the meaning of three ratings which might be established for the same boiler.

Assume that an installation has the following loads determined in accordance with the section *Selection of Boilers:*

Net Load	1000 sq ft of steam radiation
Piping Tax	200 sq ft of steam radiation
Design Load	1200 sq ft of steam radiation
Pickup Allowance	240 sq ft of steam radiation
Maximum or Gross Load	1440 sq ft of steam radiation

A boiler that is just large enough to carry this system might be said to have a ***net load rating*** of 1000 sq ft, a ***design load*** rating of 1200 sq ft, or a ***gross load*** rating of 1440 sq ft, depending on the basis on which the boiler is rated.

On a ***net load*** basis the above boiler would be rated 1000 sq ft of steam radiation and would have sufficient excess capacity to supply the ***normal piping*** and ***pickup load***. $I = B = R$ ratings, *S.H.B.I.* Net Ratings, and Net Load Ratings of the *Heating, Piping and Air Conditioning Contractors National Association* are established on this basis.

On a ***design load*** basis the boiler would be rated 1200 sq ft of steam radiation and would have sufficient excess capacity to supply the ***pickup load***. It would be of adequate size for a system in which the ***sum*** of the ***net load*** and the ***piping heat loss*** did not exceed 1200 sq ft of steam radiation. The *S.H.B.I.* Ratings shown in the first and seventh columns of Table 1 (not to be confused with *S.H.B.I.* Net Rating) are established on a ***design load*** basis.

On a ***gross output*** basis of rating the boiler would be rated 1440 sq ft of steam radiation and would be of adequate size for a system in which the sum of the ***net load***, ***piping load***, and ***pick-up load*** did not exceed 1440 sq ft of steam radiation. Gross $I = B = R$ Output and *A.G.A.* Ratings are established on a ***gross output*** basis.

In the determination of boiler ratings, the ***Gross Output*** is the quantity of heat available at the boiler nozzle with the boiler normally insulated and when operating under limitations stipulated in the code or method by which the boiler is rated. The boiler may be capable of producing a greater nozzle output but in doing so would exceed some of these limitations.

SELECTION OF BOILERS

Factors Involved in Boiler Selection

The *Maximum Load* or *Gross Load* on the boiler is the sum of the four following items.

The *Design Load* is the sum of items 1, 2 and 3.

The *Net Load* is the sum of items 1 and 2.

1. ***Radiation Load.*** The estimated heat emission in Btu per hour of the connected radiation (direct, indirect, or forced convection coils) to be installed.

The connected radiation is determined by calculating the heat losses for each room in accordance with data given in Chapters 4, 5, and 6. The sum of the calculated heat losses for all the rooms represents the total required heat emission of the connected radiation expressed in Btu per hour. As practically all boilers are now rated on a Btu basis, it is unnecessary to convert the radiation load to equivalent square feet of equivalent radiation.

2. ***Hot Water Supply Load.*** The estimated maximum heat in Btu per hour required to heat water for domestic use.

When the hot water supply is heated by the building heating boiler, this load must be taken into consideration in sizing the boiler. A common practice is to add 240 Btu per hour to the radiation load for each gallon of storage tank capacity. For more specific information see Chapter 46.

3. *Piping Tax.* The estimated heat emission in Btu per hour of the piping connecting the radiation and other apparatus to the boiler.

As the heating industry as a whole is not entirely agreed upon piping tax allowances for different sizes of installations it is better to compute the heat emission from both bare and covered pipe surface in accordance with data in Chapter 17. In average house heating systems, it is common practice to consider the piping tax to be equal to 25 per cent of the Net Load. In determining Net $I=B=R$ Ratings from Gross $I=B=R$ Output, the piping factor allowed varies from 30 per cent for small boilers to 12 per cent for larger boilers.

4. *Warming-Up or Pick-Up Allowance.* The estimated increase in the normal load in Btu per hour caused by the heating up of the cold system.

The warming-up allowance represents the load due to heating the boiler and contents to operating temperature, and heating up cold radiation and piping. The factors to be used for determining the allowance to be made should be selected from Table 2.

TABLE 2. WARMING-UP ALLOWANCES FOR HAND-FIRED LOW-PRESSURE STEAM AND HOT WATER HEATING BOILERS[a,b,c]

DESIGN LOAD (REPRESENTING SUMMATION OF ITEMS 1, 2, AND 3)		PERCENTAGE CAPACITY TO ADD FOR WARMING-UP[c]
Btu per Hour	Equivalent Square Feet of Radiation[d]	
Up to 100,000	Up to 420	65
100,000 to 200,000	420 to 840	60
200,000 to 600,000	840 to 2500	55
600,000 to 1,200,000	2500 to 5000	50
1,200,000 to 1,800,000	5000 to 7500	45
Above 1,800,000	Above 7500	40

[a]This table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings, except that the second column has been added for convenience in interpreting the design load in terms of equivalent square feet of radiation.

[b]See also Time Analysis in Starting Heating Apparatus, by Ralph C. Taggart (A.S.H.V.E. TRANSACTIONS, Vol. 19, 1913, p. 292); Report of A.S.H.V.E. Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 35); Selecting the Right Size Heating Boiler, by Sabin Crocker (*Heating, Piping and Air Conditioning*, March, 1932).

[c]This table refers to hand-fired, solid fuel boilers. A factor of 20 per cent over design load is adequate when automatically-fired fuels are used.

[d]240 Btu per square foot.

Other items to be considered in boiler selection are:

(*a*) Efficiency with hard or soft coal, gas, or oil firing, as the case may be.

(*b*) Grate area with hand-fired coal, or fuel burning rate with stokers, oil, or gas.

(*c*) Combustion space in the furnace.

(*d*) Type of heat liberation, whether continuous or intermittent, or a combination of both.

(*e*) Convenience in firing and cleaning.

(*f*) Adaptability to changes in fuel and kind of attention.

(*g*) Height of water line.

(*h*) Miscellaneous items such as draft available, possibility of future extension, possibility of break-down, and head room in the boiler room.

(*i*) The most economical size of boiler is usually one that is just the right size for the load. Either larger or smaller boilers may be less economical.

Selection of Cast-Iron Boilers

Net load ratings of cast-iron boilers are usually available from manufacturers' catalogs. They may also be obtained conveniently from published tables of $I=B=R$ ratings[6], or from recommendations of the *Heating, Piping and Air Conditioning Contractors National Association*[7]

[6]I=B=R Ratings for Cast-Iron Boilers (*Institute of Boiler and Radiator Manufacturers*).

[7]Engineering Standards, Part II, Net Square Feet Radiation Loads in 70 Deg Fahr., Recommended for Low Pressure Heating Boilers, 1943 (*Heating, Piping and Air Conditioning Contractors National Association*.)

and can be used in selection of boilers, unless the heating system contains an unusual amount of bare pipe, or the nature of the connected load is such that the normal allowances for pipe loss and pick-up do not apply. In such a case, the selection must be based on the gross output.

Selection of Steel Heating Boilers

Catalog ratings in accordance with the previously mentioned *Steel Heating Boiler Institute* code are intended to correspond with the estimated *design load*. When the heat emission of the piping is not known, the net load to be considered for the boiler may be determined from Table 1. The difference between design load and net load represents an amount which is considered normal for piping loss of the ordinary heating system.

Boilers with less than 128 sq ft of heating surface are classified as residence size. An insulated residence boiler for oil or gas, not convertible, may carry a *net load* expressed in square feet of steam radiation of not more than 17 times the square feet of heating surface in the boiler, provided the boiler manufacturer guarantees it to be capable of operating at a *maximum output* of not less than 150 per cent of net load rating with over-all efficiency of not less than 75 per cent with at least two different makes of each type of standard commercial burner recommended by the boiler manufacturer. If the heat loss from the piping system exceeds 20 per cent of the installed radiation, the excess is to be considered as a part of the *net load*.

Selection Based on Heating Surface and Grate Area

Where neither the *net load* nor *gross output* ratings based upon performance tests are available, a good general rule for conventionally designed boilers is to provide 1 sq ft of boiler heating surface for each 14 sq ft of equivalent radiation (240 Btu per square foot) represented by the design load. This is equivalent to allowing 10 sq ft of boiler heating surface per boiler horsepower. In this case it is assumed that the *maximum load* including the warming-up allowance will be provided for by operating the boiler in excess of the *design load*, that is, in excess of the 100 per cent rating on a boiler-horsepower basis. *S.H.B.I.* ratings for hand firing are based on 10 sq ft of heating surface per boiler horsepower.

Due to the wide variation which may be encountered in manufacturers' ratings for boilers of approximately the same capacity, it is advisable to check the grate area required for heating boilers burning solid fuel by means of the following formula:

$$G = \frac{H}{C \times F \times E} \tag{1}$$

where

G = grate area, square feet.
H = required *gross output* of the boiler, Btu per hour (see Selection of Boilers).
C = desirable combustion rate for fuel selected, pounds of dry coal per square foot of grate per hour (see Table 3).
F = calorific value of fuel, Btu per pound.
E = efficiency of boiler, usually taken as 0.60.

Example 1. Determine the grate area for a required *gross output* of the boiler of 500,000 Btu per hour, a combustion rate of 6 lb per hour, a calorific value of 13,000 Btu per pound, and an efficiency of 60 per cent.

$$G = \frac{500{,}000}{6 \times 13{,}000 \times 0.60} = 10.7 \text{ sq ft}$$

3. *Piping Tax.* The estimated heat emission in Btu per hour of the piping connecting the radiation and other apparatus to the boiler.

As the heating industry as a whole is not entirely agreed upon piping tax allowances for different sizes of installations it is better to compute the heat emission from both bare and covered pipe surface in accordance with data in Chapter 17. In average house heating systems, it is common practice to consider the piping tax to be equal to 25 per cent of the Net Load In determining Net $I=B=R$ Ratings from Gross $I=B=R$ Output, the piping factor allowed varies from 30 per cent for small boilers to 12 per cent for larger boilers

4. *Warming-Up or Pick-Up Allowance.* The estimated increase in the normal load in Btu per hour caused by the heating up of the cold system.

The warming-up allowance represents the load due to heating the boiler and contents to operating temperature, and heating up cold radiation and piping. The factors to be used for determining the allowance to be made should be selected from Table 2

TABLE 2. WARMING-UP ALLOWANCES FOR HAND-FIRED LOW-PRESSURE STEAM AND HOT WATER HEATING BOILERS[a,b,c]

Design Load (Representing Summation of Items 1, 2, and 3)		Percentage Capacity to Add for Warming-Up[c]
Btu per Hour	Equivalent Square Feet of Radiation[d]	
Up to 100,000	Up to 420	65
100,000 to 200,000	420 to 840	60
200,000 to 600,000	840 to 2500	55
600,000 to 1,200,000	2500 to 5000	50
1,200,000 to 1,800,000	5000 to 7500	45
Above 1,800,000	Above 7500	40

[a]This table is taken from the A.S H.V.E Code of Minimum Requirements for the Heating and Ventilation of Buildings, except that the second column has been added for convenience in interpreting the design load in terms of equivalent square feet of radiation

[b]See also Time Analysis in Starting Heating Apparatus, by Ralph C Taggart (A S H.V E. TRANSACTIONS, Vol. 19, 1913, p. 292); Report of A.S H V.E. Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers (A.S H V.E. TRANSACTIONS, Vol. 36, 1930, p 35); Selecting the Right Size Heating Boiler, by Sabin Crocker (*Heating, Piping and Air Conditioning*, March, 1932).

[c]This table refers to hand-fired, solid fuel boilers A factor of 20 per cent over design load is adequate when automatically-fired fuels are used.

[d]240 Btu per square foot

Other items to be considered in boiler selection are:

(*a*) Efficiency with hard or soft coal, gas, or oil firing, as the case may be.

(*b*) Grate area with hand-fired coal, or fuel burning rate with stokers, oil, or gas.

(*c*) Combustion space in the furnace.

(*d*) Type of heat liberation, whether continuous or intermittent, or a combination of both.

(*e*) Convenience in firing and cleaning.

(*f*) Adaptability to changes in fuel and kind of attention.

(*g*) Height of water line.

(*h*) Miscellaneous items such as draft available, possibility of future extension, possibility of break-down, and head room in the boiler room.

(*i*) The most economical size of boiler is usually one that is just the right size for the load. Either larger or smaller boilers may be less economical.

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and can be used in selection of boilers, unless the heating system contains an unusual amount of bare pipe, or the nature of the connected load is such that the normal allowances for pipe loss and pick-up do not apply. In such a case, the selection must be based on the gross output.

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$$G = \frac{H}{C \times F \times E} \tag{1}$$

where

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$$G = \frac{500{,}000}{6 \times 13{,}000 \times 0.60} = 10.7 \text{ sq ft}$$

TABLE 3. PRACTICAL COMBUSTION RATES FOR COAL-FIRED HEATING BOILERS OPERATING AT MAXIMUM LOAD ON NATURAL DRAFT OF FROM 1/8 IN. TO 1/2 IN. WATER[a]

KIND OF COAL	SQ FT GRATE	LB OF COAL PER SQ FT GRATE PER HOUR
No. 1 Buckwheat Anthracite	Up to 4	3
	5 to 9	3½
	10 to 14	4
	15 to 19	4½
	20 to 25	5
Anthracite Pea	Up to 9	5
	10 to 19	5½
	20 to 25	6
Anthracite Nut and Larger	Up to 4	8
	5 to 9	9
	10 to 14	10
	15 to 19	11
	20 to 25	13
Bituminous	Up to 4	9.5
	5 to 14	12
	15 and above	15 5

[a]Steel boilers usually have higher combustion rates for grate areas exceeding 15 sq ft than those indicated in this table.

The boiler selected should have a grate area not less than that determined by Formula 1. With small boilers, where it is desired to provide sufficient coal capacity for approximately an eight-hour firing period plus a 20 per cent reserve for igniting a new charge, more grate area may be required depending upon the depth of the fuel pot.

Selection of Gas-Fired Boilers

After determining the *net load* for the installation, gas designed boilers can usually be selected from manufacturers' tables of *net load ratings* which are based on piping and pickup allowances varying from 56 per cent for boilers of 200 sq ft and less to 35 per cent for boilers of 4000 sq ft and larger. If the piping and pickup load or other factors create an unusual load, a boiler should be selected which has an *A.G.A. output rating* equal to the maximum output required. Detailed recommendations for selection of gas designed boilers are given in the *A.G.A.* publication, Comfort Heating[8].

SPACE LIMITATIONS

Boiler rooms should, if possible, be situated at a central point with respect to the building and should be designed for a maximum of natural light. The space in front of the boilers should be sufficient for firing, stoking, ash removal and cleaning or renewal of flue tubes, and should be at least 3 ft greater than the length of the tubes.

A space of at least 3 ft should be allowed on at least one side of every boiler for convenience of erection and for accessibility to the various dampers, cleanouts and trimmings. The space at the rear of the boiler

[8]Comfort Heating, 1938, pp. 35 to 39 (*American Gas Association*).

should be ample for the chimney connection and for cleanouts. With large boilers the rear clearance should be at least 3 ft in width.

The boiler room height should be sufficient for the location of boiler accessories and for proper installation of piping. In general the ceiling height for small steam boilers should be at least 3 ft above the normal boiler water line. With vapor heating, especially, the height above the boiler water line is of vital importance.

When steel boilers are used, space should be provided for the removal and replacement of tubes.

CONNECTIONS AND FITTINGS

Steam outlet connections should be the full size of the manufacturers' tappings in order to keep the velocity of flow through the outlet reasonably low and to avoid fluctuation of the water line and undue entrainment of moisture, and should extend vertically to the maximum height available above the boiler. A steam velocity in boiler outlets not exceeding 25 to 30 fps at maximum load is recommended unless data are available to show that a higher velocity is satisfactory.

Particular attention should be given to *fitting connections* to secure conformity with the *A.S.M.E.* Boiler Construction Code for Low Pressure Heating Boilers. Attention is called in particular to pressure gage piping, water gage connections and safety valve capacity.

Steam gages should be fitted with a water seal and a shut-off, consisting of a cock with either a tee or lever handle which is parallel to the pipe when the cock is open. Connections to *steam gage siphons* shall be of non-ferrous metal when smaller than 1 in. pipe size and longer than 5 ft between siphon and point of connection of pipe to boiler, and also when smaller than ½ in. pipe size and shorter than 5 ft.

Each steam or vapor boiler must have at least one *water gage* glass and two or more *gage cocks* located within the range of the visible length of the glass. The water gage fittings or gage cocks may be directly connected to the boiler, if so located by the manufacturer, or may be mounted on a separate water column. No connections, except for combustion regulators, drains or steam gages, should be placed on the pipes connecting the water column and the boiler. If the water column or gage glass is connected to the boiler by pipe and fittings, a cross, tee or equivalent, in which a cleanout plug or a drain valve and piping may be attached, should be placed in the water connection at every right-angle turn to facilitate cleaning. The water line in steam boilers should be carried at the level specified by the boiler manufacturer.

Safety valves must be capable of discharging all the steam that can be generated by the boiler without allowing the pressure to rise more than [illegible] lb above the maximum allowable working pressure of the boiler. This should be borne in mind particularly in the case of boilers equipped with mechanical stokers or oil burners where the amount of grate area has little significance as to the steam generating capacity of the boiler.

Where a *return header* is used on a cast-iron sectional boiler to distribute the returns to both rear tappings, it is advisable to provide full size plugged tees instead of elbows where the branch connections enter the return tappings. This facilitates cleaning sludge from the bottom of the boiler sections through the large plugged openings. An equivalent cleanout plug should be provided in the case of a single return connection.

arranged that the entire system may be drained of water by opening the drain cock. In the case of two or more boilers separate blow-off connections must be provided for each boiler on the boiler side of the stop valve on the main return connection.

Water service connections must be provided for both steam and water boilers, for refilling and for the addition of make-up water to boilers. This connection is usually of galvanized steel pipe, and is made to the return main near the boiler or boilers.

For further data on pipe connections for steam and hot water heating systems, see Chapters 14 and 15 and the *A.S.M.E.* Boiler Construction Code for Low Pressure Heating Boilers.

Smoke Breeching and Chimney Connections. The breeching or smoke pipe from the boiler outlet to the chimney should be air-tight and as short and direct as possible, preference being given to long radius and 45-deg instead of 90-deg bends. The breeching entering a brick chimney should not project beyond the flue lining and where practicable it should be grouted from the inside of the chimney. A thimble or sleeve usually is provided where the breeching enters a brick chimney.

Where a battery of boilers is connected into a breeching each boiler should be provided with a tight damper. The breeching for a battery of boilers should not be reduced in size as it goes to the more remote boilers. Good connections made to a good chimney will usually result in a rapid response by the boilers to demands for heat.

ERECTION, OPERATION, AND MAINTENANCE

The directions of the boiler manufacturer should always be read before the assembly or installation of any boiler is started, even though the contractor may be familiar with the boiler. All joints requiring boiler putty or cement which cannot be reached after assembly is complete must be finished as the assembly progresses.

Five precautions that should be taken in all installations to prevent damage to the boiler are:

1. There should be provided proper and convenient drainage connections for use if the boiler is not in operation during freezing weather.
2. Strains on the boiler due to movement of piping during expansion should be prevented by suitable anchoring of piping and by proper provision for pipe expansion and contraction.
3. Direct impingement of too intense local heat upon any part of the boiler surface, as with oil burners, should be avoided by protecting the surface with firebrick or other refractory material.
4. Condensation in steam systems must flow back to the boiler as rapidly and uniformly as possible. Return connections should prevent the water from backing out of the boiler.
5. Automatic boiler feeders and low water cut-off devices which shut off the source of heat if the water in the boiler falls below a safe level are recommended for mechanically fired boilers.

Boiler Troubles

A complaint regarding boiler operation generally will be found to be due to one of the following:

1. *The boiler fails to deliver enough heat.* The cause of this condition may be: (*a*) poor draft; (*b*) poor fuel; (*c*) inferior attention or firing; (*d*) boiler too small; (*e*) improper piping; (*f*) improper arrangement of sections; (*g*) heating surfaces covered with soot; (*h*) insufficient radiation installed; and (*i*) with mechanical firing, fuel burning equipment too small.

2. *The water line is unsteady.* The cause of this condition may be: (*a*) grease and dirt in boiler; (*b*) water column connected to a very active section and, therefore, not showing actual water level in boiler; and (*c*) boiler operating at excessive output.

3. *Water disappears from gage glass.* This may be caused by: (*a*) priming due to grease and dirt in boiler; (*b*) too great pressure difference between supply and return piping preventing return of condensation; (*c*) valve closed in return line; (*d*) connection of bottom of water column into a very active section or thin waterway; and (*e*) improper connections between boilers in battery permitting boiler with excess pressure to push returning condensation into boiler with lower pressure.

4. *Water is carried over into steam main.* This may be caused by: (*a*) grease and dirt in boiler; (*b*) insufficient steam dome or too small steam liberating area, (*c*) outlet connections of too small area; (*d*) excessive rate of output; and (*e*) water level carried higher than specified.

5. *Boiler is slow in response to operation of dampers.* This may be due to· (*a*) poor draft resulting from air leaks into chimney or breeching; (*b*) inferior fuel; (*c*) inferior attention; (*d*) accumulation of clinker on grate; and (*e*) boiler too small for the load.

6. *Boiler requires too frequent cleaning of flues.* This may be due to: (*a*) poor draft; (*b*) smoky combustion; (*c*) too low a rate of combustion; and (*d*) too much excess air in firebox causing chilling of gases.

7. *Boiler smokes through fire door.* This may be due to: (*a*) defective draft in chimney or incorrect setting of dampers; (*b*) air leaks into boiler or breeching; (*c*) gas outlet from firebox plugged with fuel; (*d*) dirty or clogged flues; and (*e*) improper reduction in breeching size.

8. *Low carbon dioxide.* This may be due on oil burning boilers to: (*a*) improper adjustment of the burner; (*b*) leakage through the boiler setting; (*c*) improper fire caused by a fouled nozzle; or (*d*) to an insufficient quantity of oil being burned.

Cleaning Boilers

All boilers are provided with flue clean-out openings through which the heating surface can be reached by means of brushes or scrapers. Flues of solid fuel boilers should be cleaned often to keep the surfaces free of soot or ash. Gas boiler flues and burners should be cleaned at least once a year. Oil burning boiler flues should be examined periodically to determine when cleaning is necessary.

The grease used to lubricate the cutting tools during erection of new piping systems serves as a carrier for sand and dirt, with the result that a scum of fine particles and grease accumulates on the surface of the water in all new boilers, while heavier particles may settle to the bottom of the boiler and form sludge. These impurities have a tendency to cause foaming, preventing the generation of steam and causing an unsteady water line.

This unavoidable accumulation of oil and grease should be removed by blowing off the boiler as follows: If not already provided, install a surface blow connection of at least 1¼ in. nominal pipe size with outlet extended to within 18 in. of the floor or to sewer, inserting a valve in line close to boiler. Bring the water line to center of outlet, raise steam pressure and while fire is burning briskly open valve in blow-off line. When pressure recedes, close valve and repeat process adding water at intervals to maintain proper level. As a final operation bring the pressure in the boiler to about 10 lb, close blow-off, draw the fire or stop burner, and open drain valve. After boiler has cooled partly, fill and flush out several times before filling it to proper water level for normal service. The use of soda, or any alkali, vinegar or any acid is not recommended for cleaning heating boilers because of the difficulty of complete removal and the possibility of subsequent injury, after the cleaning process has been completed.

Insoluble compounds have been developed which are effective, but special instructions on the proper cleaning compound and directions for

its use in a boiler, as given by the boiler manufacturer, should be carefully followed.

Care of Idle Heating Boilers

Heating boilers are often seriously damaged during summer months due chiefly to corrosion resulting from the combination of sulphur from the fuel with the moisture in the cellar air. At the end of the heating season the following precautions should be taken:

1. All heating surfaces should be cleaned thoroughly of soot, ash and residue, and the heating surfaces of steel boilers should be given a coating of lubricating oil on the fire side.

2. All machined surfaces should be coated with oil or grease.

3. Connections to the chimney should be cleaned and in case of small boilers the pipe should be placed in a dry place after cleaning.

4. If there is much moisture in the boiler room, it is desirable to drain the boiler to prevent atmospheric condensation on the heating surfaces of the boiler when they are below the dew-point temperature. Due to the hazard that some one may inadvertently build a fire in a dry boiler, however, it is safer to keep the boiler filled with water, particularly in residential installations. Air can be excluded from a steam boiler by raising the water level into the steam outlets. A hot water system usually is left filled to the expansion tank.

5. The grates and ashpit should be cleaned.

6. Clean and repack the gage glass if necessary.

7. Remove any rust or other deposit from exposed surfaces by scraping with a wire brush or sandpaper. After boiler is thoroughly cleaned, apply a coat of preservative paint where required to external parts normally painted.

8. Inspect all accessories of the boiler carefully to see that they are in good working order. In this connection, oil all door hinges, damper bearings and regulator parts.

Radiators and Convectors

Heat Emission of Radiators and Convectors, Types of Radiators, Convectors, Radiator and Convector Ratings, Effect of Operating Conditions, Heating Effect, Heating Up the Radiator and Convector, Enclosed Radiators

THE accepted terms for heating units are: (1) *radiators*, for direct surface heating units, either exposed, enclosed, or shielded, which emit a large percentage of their heat by radiation; and (2) *convectors*, for heating units having a large percentage of extended fin surface and which emit heat principally by convection. Convectors are dependent upon enclosures to provide the circulation by gravity of large volumes of air.

HEAT EMISSION OF RADIATORS AND CONVECTORS

Most heating units emit heat by *radiation* and *convection*. The resultant heat from these processes depends upon whether or not the heating unit is exposed or enclosed and upon the contour and surface characteristics of the material in the units.

An exposed radiator emits roughly half of its heat by radiation, the amount depending upon the size and number of sections. When the radiator is enclosed or shielded, the proportion of radiation is further reduced. The balance of the emission is by conduction to the air in contact with the heating surface, and the resulting circulation of the air warms by convection.

A convector emits practically all of its heat by conduction to the air surrounding it and this heated air is in turn transmitted by convection to the rooms or spaces to be warmed, the heat emitted by radiation being negligible.

The output of a radiator can be measured only by the heat it emits. The old standard of comparison used to be square feet of *actual surface*, but since the advance in radiator design and proportions, the surface area alone is not a true index of output. (The engineering unit of output is the *Mbh* or 1000 Btu per hour.) However, during the period of transition from the old to the new, radiators may be referred to in terms of *equivalent square feet*. For steam service this is based on an emission of 240 Btu per hour per square foot and for hot water service 150 Btu.

TYPES OF RADIATORS

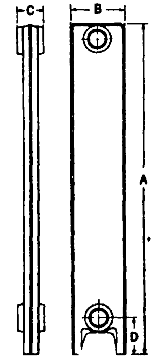
Present day radiators may be classified as tubular, wall, or window type and are generally made of cast-iron. Only the small-tube type of tubular radiators with a spacing of 1¾ in. per section are now available, the large-tube type which had a spacing of 2½ in. per section having been discontinued. Small-tube radiators occupy less space and are particularly suited for installation in recesses.

After a complete study of the demand for various sizes of radiators, the *Institute of Boiler and Radiator Manufacturers*, in cooperation with the Division of Simplified Practice, *National Bureau of Standards*, established Simplified Practice Recommendation R174-43 for small-tube cast-iron radiators. Table 1 shows the size and dimensions of small-tube cast-iron radiators which are being manufactured at the present time.

Wall radiators are now rated in terms of equivalent square feet, the same as small-tube radiators. Tests have shown that the heat emitted

TABLE 1. SMALL-TUBE CAST-IRON RADIATORS

Number of Tubes per Section	Catalog Rating per Section[a]	Section Dimensions				
		A Height[c]	*B* Width		*C* Spacing[b]	*D* Leg Height[c]
			Minimum	Maximum		
	Sq Ft	In.	In.	In.	In.	In.
3[d]	1.6	25	3¼	3½	1¾	2½
4[d]	1.6	19	4 7/16	4 13/16	1¾	2½
	1.8	22	4 7/16	4 13/16	1¾	2½
	2.0	25	4 7/16	4 13/16	1¾	2½
5[d]	2.1	22	5⅝	6 5/16	1¾	2½
	2.4	25	5⅝	6 5/16	1¾	2½
6[d]	1.6	14	6 13/16	8	1¾	2½
	2.3	19	6 13/16	8	1¾	2½
	3.0	25	6 13/16	8	1¾	2½
	3.7	32	6 13/16	8	1¾	2½



[a]The square foot of equivalent direct steam radiation is defined as the ability to emit 240 Btu per hour, with steam at 215 F, in air of 70 F. These ratings apply only to installed radiators exposed in a normal manner; not to radiators installed behind enclosures, grilles, etc. (See A S H.V E. Code for Testing Radiators adopted January, 1927)

[b]Length equals number of sections times 1¾ in.

[c]Over-all height and leg height, as produced by some manufacturers, are one inch (1 in) greater than shown in Columns *A* and *D*. Radiators may be furnished without legs Where greater than standard leg heights are required this dimension shall be 4½ in.

[d]Or equal.

from a wall-type radiator may be reduced from 5 to 10 per cent if the radiator is placed near the ceiling with the bars horizontal and in an air temperature exceeding 70 F. When radiators are placed near the ceiling, there is usually such a large difference in the temperature between the floor level and the ceiling that it becomes difficult to heat the living zone of the room satisfactorily.

Pipe coils are assemblies of standard pipe or tubing (1 in. to 2 in.) which are used as radiators. In older practice these coils were commonly used in factory buildings, but now wall-type radiators are most frequently used for this service. When coils are used, the miter type assembly is to be preferred as it best cares for expansion in the pipe. Cast manifolds or headers, known as branch tees, are available for this construction.

The heat emission of pipe coils placed vertically on a wall with the pipes horizontal is given in Table 2. This has been developed from avail-

TABLE 2. HEAT EMISSION OF PIPE COILS PLACED VERTICALLY ON A WALL (PIPES HORIZONTAL) CONTAINING STEAM AT 215 F AND SURROUNDED WITH AIR AT 70 F

Btu per linear foot of coil per hour (not linear feet of pipe)

Size of Pipe	1 In.	1¼ In.	1½ In.
Single row	132	162	185
Two	252	312	348
Four	440	545	616
Six	567	702	793
Eight	651	796	907
Ten	732	907	1020
Twelve	812	1005	1135

able data and does not represent definite results of tests. For such coils the heat emission varies as the height of the coil. The heat emission of each pipe of ceiling coils, placed horizontally, is about 126 Btu, 156 Btu, and 175 Btu per linear foot of pipe, respectively, for 1-in., 1¼-in., and 1½-in. coils.

CONVECTORS

Cast-iron radiators may be concealed in a cabinet or other enclosure for appearance. In such cases a greater percentage of heat is conveyed to the room by convection thereby resulting in a form of gravity convector. A typical recessed convector is shown in Fig. 1. The heating element consisting of a large percentage of fin surface is usually shallow in depth and placed low in the enclosure in order to produce maximum chimney effect in the enclosure. The air enters the enclosure near the floor line just below the heating element, is moderately heated in passing through

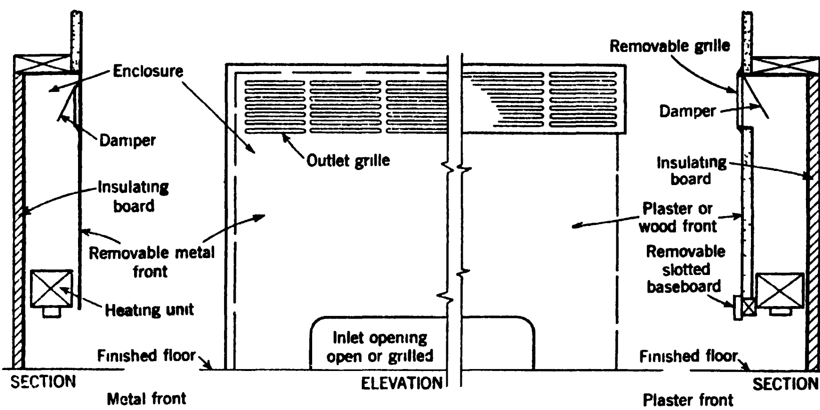


FIG. 1. TYPICAL RECESSED CONVECTOR

the core and delivered to the room through an opening near the top of enclosure. Since the air can only enter the enclosure at the floor line, the cooler air in the room, which always lies at this level, is constantly being withdrawn and replaced by the warmer air. This air movement accomplishes the desired reduction in temperature differentials and assures maximum comfort in the living zone.

Concealed heaters or convectors are generally available as completely built-in units. The enclosing cabinet should be designed with suitable air inlet and outlet grilles to give the heating element its best performance. Tables of capacities are catalogued for various lengths, depths and heights, and combinations are available in several styles for installations, such as the wall-hung type, free-standing floor type, recess type set flush with wall or offset, and the completely concealed type. Most of these types may be arranged with a top outlet grille in a plane parallel with the floor, although the front outlet is practically standard. In cases where enclosures are to be used but are not furnished by the heater manufacturer, it is important that the proportions of the cabinet and the grilles be so designed that they will not impair the performance of the assembled convector. It is desirable that the enclosure or housing for the convector fit as snugly as possible

so that the air to be heated must pass through the convector and cannot be by-passed in the enclosure.

The output of a convector, for any given length and depth, is a function of the height. Published ratings are generally given in terms of equivalent square feet, corrected for heating effect. However, an extended surface heating unit is entirely different structurally and physically from a direct radiator and, since it has no area measurement corresponding to the heating surface of a radiator, many engineers believe that the performance of convectors should be stated in Btu. For steam convectors, as for radiators, 240 Btu per hour may be taken as an equivalent square foot of radiation. When more than one heating unit is used, one mounted above the other in the same cabinet, the output of the upper unit or units will be materially less than that of the bottom unit.

RADIATOR AND CONVECTOR RATINGS

A standard method of testing radiators was adopted by the A.S.H.V.E. in 1927[1]. This Code provides for a standard test room, the temperature of which is to be maintained at 70 F, measured in the center of the room at an elevation of 5 ft above the floor. The steam temperature in the radiator is to be 215 F, which corresponds to 15.6 lb per square inch absolute. The weight of condensate per hour, under these standard conditions, multiplied by the difference in the enthalpy of the steam entering the radiator and that of the condensate leaving the radiator, gives the radiator output in Btu per hour. This output divided by 240 gives the steam rating of the radiator in square feet.

Similar test methods for convectors are the A.S.H.V.E. Codes for Testing and Rating Concealed Gravity Type Radiation[2], (Steam Code 1932 and Hot Water Code 1933). These Codes recognize a different type of test booth, and the air temperature used is that of the air entering the convector casing instead of the temperature in the center of the room. The entering air temperature for standard test conditions is 65 F. For hot water the standard test conditions call for a mean temperature of the water in the convector of 170 F.

The *Convector Manufacturers Association* has adopted the A.S.H.V.E. standard in the formulation of its ratings and has compiled a tentative standard of heating effect allowances for various enclosure heights to be included in the ratings by its members.

All published ratings bearing the title *C.M.C.* Ratings (*Convector Manufacturers Certified Ratings*) indicate that the convectors have been tested in accordance with the A.S.H.V.E. Code by an impartial and disinterested laboratory and that the ratings have been approved by the Standardization Committee of the *Convector Manufacturers Association*.

Effect of Operating Conditions

The heat output of a radiator is proportional to the 1.3 power of the temperature difference between the air in the room at the 60 in. level and the heating medium in the radiator. The heat output of a convector is proportional to the 1.5 power of the temperature difference between the air entering the convector and the heating medium, steam or hot water,

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[1]A.S.H.V.E. Code for Testing Radiators (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p 18).

[2]A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam), (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 367); (Hot Water), (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p 237) (See also A.S.H.V.E. TRANSACTIONS, Vol 41, 1935, p. 38, and Vol. 42, 1936, p. 29)

within the convector[3]. For hot water the arithmetical average between entering and leaving water temperatures is used. These laws may be expressed as correction factors to change from output under standard rating-test conditions to output under other operating conditions. Such factors are given in Table 3.

TABLE 3. CORRECTION FACTORS FOR DIRECT CAST-IRON RADIATORS AND CONVECTORS[a]

Steam Press. Approx.		Heating Medium Temp F Steam or Water	Factors for Direct Cast-Iron Radiators							Factors for Convectors						
			Room Temperature F							Inlet Air Temperature F						
Gage Vacuum In. Hg.	Abs. Lb per Sq In.		80	75	70	65	60	55	50	80	75	70	65	60	55	50
22.4	3.7	150	2.58	2.36	2.17	2.00	1.86	1.73	1.62	3.14	2.83	2.57	2.35	2.15	1.98	1.84
20.3	4.7	160	2.17	2.00	1.86	1.73	1.62	1.52	1.44	2.57	2.35	2.15	1.98	1.84	1.71	1.59
17.7	6.0	170	1.86	1.73	1.62	1.52	1.44	1.35	1.28	2.15	1.98	1.84	1.71	1.59	1.49	1.40
14.6	7.5	180	1.62	1.52	1.44	1.35	1.28	1.21	1.15	1.84	1.71	1.59	1.49	1.40	1.32	1.24
10.9	9.3	190	1.44	1.35	1.28	1.21	1.15	1.10	1.05	1.59	1.49	1.40	1.32	1.24	1.17	1.11
6.5	11.5	200	1.28	1.21	1.15	1.10	1.05	1.00	0.96	1.40	1.32	1.24	1.17	1.11	1.05	1.00
Lb per Sq In																
1	15.6	215	1.10	1.05	1.00	0.96	0.92	0.88	0.85	1.17	1.11	1.05	1.00	0.95	0.91	0.87
6	21	230	0.96	0.92	0.88	0.85	0.81	0.78	0.76	1.00	0.95	0.91	0.87	0.83	0.79	0.76
15	30	250	0.81	0.78	0.76	0.73	0.70	0.68	0.66	0.83	0.79	0.76	0.73	0.70	0.68	0.65
27	42	270	0.70	0.68	0.66	0.64	0.62	0.60	0.58	0.70	0.68	0.65	0.63	0.60	0.58	0.56
52	67	300	0.58	0.57	0.55	0.53	0.52	0.51	0.49	0.56	0.54	0.53	0.51	0.49	0.48	0.47

[a]To determine the size of a radiator or a convector for a given space, divide the heat loss in Btu per hour by 240 and multiply the result by the proper factor from the above table.

To determine the heating capacity of a radiator or a convector under conditions other than the basic ones with the heating medium at a temperature of 215 F, and the room temperature at 70 F in the case of a radiator, and the inlet air temperature at 65 F in the case of a convector, divide the heating capacities at the basic conditions by the proper factor from the above table.

When it is desired to change the output under any test conditions to the corresponding output under standard Code test conditions, the reciprocal form of correction factor may be derived. The equations for steam units are:

For radiators: For convectors:

$$C_s = \left(\frac{215 - 70}{t_s - t_r}\right)^{1.3} \qquad C_s = \left(\frac{215 - 65}{t_s - t_i}\right)^{1.5} \tag{3}$$

The output under standard conditions will be:

$$H_s = C_s H_t \tag{4}$$

where

C_s = correction factor.
t_s = steam temperature during test, degrees Fahrenheit.
t_r = room temperature during test, degrees Fahrenheit.
t_i = inlet air temperature during test, degrees Fahrenheit.
H_s = heat emission rating under standard conditions, Btu per hour.
H_t = heat output under test conditions, Btu per hour.

The relation between the size of the radiator or convector and the size of the test room will affect the results obtained in a capacity-rating test[4]. The height and location of the radiator and the insulation of the test room are other important factors that are not specifically regulated by the Code.

[3]A.S.H.V.E. RESEARCH REPORT No. 998—Factors Affecting the Heat Output of Convectors, by A. P Kratz, M. K. Fahnestock, and E. L. Broderick (A.S.H.V.E TRANSACTIONS, Vol. 40, 1934, p. 443).

[4]Factors Influencing the Heat Output of Radiators, by A C. Davis, W M. Sawdon and David Dropkin (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1942, p. 185) and Cornell University, *Engineering Experiment Station Bulletin* No. 29, April, 1942.

For a radiator, the finish coat of paint affects the heat output. Oil paints of any color will give about the same results as unpainted black or rusty surfaces, but an aluminum or a bronze paint will reduce the heat emitted by radiation. The net effect may be a reduction of 10 per cent or more in the total heat output of the radiator[5,6,7].

Radiator enclosures and convector casings affect the heat distribution within the room as well as the total amount of heat supplied by the steam or hot water[8].

Heating Effect

For several years the term *heating effect* has been used to designate the relation between the *useful output* of a radiator, in the comfort zone of a room, and the total input as measured by steam condensation or water temperatures[9,10]. The application of such a heating effect factor is a recog-

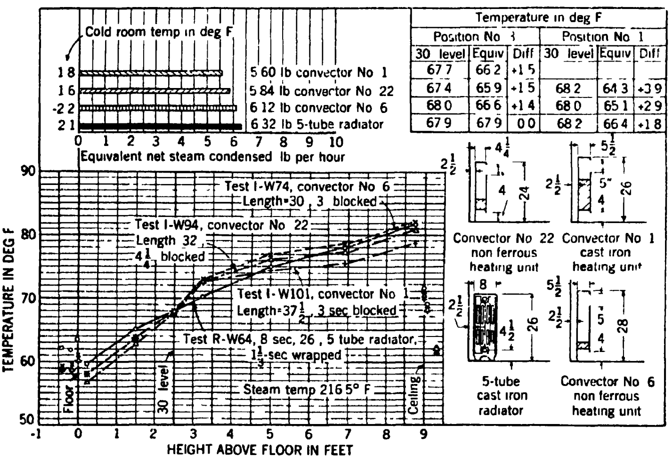

Fig. 2. Temperature Gradients and Equivalent Temperatures for Radiator and Convectors with Common 30 in. Level Temperature

nition that some radiators and convectors use less steam than others for producing equal comfort heating results in the room.

No standard method for evaluating the heating effect of radiators and convectors and correlating it with comfort has yet been accepted. One method, with test data[11] on radiators and convectors, and making use of

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[5]Heat Emission from Radiators, by K F Rubert (Cornell University, *Engineering Experiment Station Bulletin* No 24, 1937)

[6]Comparative Tests of Radiator Finishes, by W H Severns (A S.H V E Transactions, Vol 33, 1927, p. 41).

[7]Heat Loss from Direct Radiation, by J R Allen (A S H V E Transactions, Vol 26, 1920, p 11)

[8]Heat Output of Concealed Radiators, by E A Allcut (University of Toronto, *School of Engineering Research, Bulletin* No 140, 1933)

[9]The Heating Effect of Radiators, by Charles Brabbeé (A S H V E Transactions, Vol 33, 1927, p. 33).

[10]University of Illinois, *Engineering Experiment Station Bulletins* Nos 192 and 223, and Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A C Willard, A. P. Kratz, M. K. Fahnestock and S. Konzo (A S H V E Transactions, Vol 35, 1929, p 77)

[11]A.S.H V.E. Research Report No 962—The Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperature, by A. C. Willard, A. P. Kratz and M. K. Fahnestock (A S H V E. Transactions, Vol 39, 1933, p. 303).

the eupatheoscope for evaluating the environment produced, has been suggested by the University of Illinois. The principle underlying the eupatheoscope involves the measurement of the heat loss from a sizable body by radiation and convection, when the surface is maintained at some constant temperature. Through the use of this instrument and its calibration curve, non-uniform environments may be referred to uniform environments in which the air and all surrounding surfaces are at the same temperature. The temperatures of the uniform environments are referred to as equivalent temperatures.

Data given in Fig. 2 show that while the air temperature at the 30-in. level is the same for the three convectors and the one large-tube cast-iron radiator, in position No. 3 in the test room, the equivalent temperature is 1.5 F lower than the air temperature in the case of the three convectors,

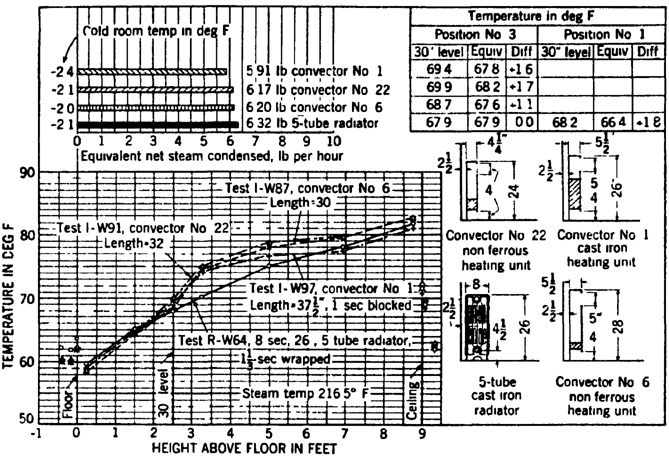


FIG. 3. TEMPERATURE GRADIENTS AND EQUIVALENT TEMPERATURES FOR RADIATOR AND CONVECTORS WITH COMMON EQUIVALENT TEMPERATURE

and the same as the air temperature in the case of the radiator. The difference between the minimum and the maximum amount of heat required to maintain the common air temperature at the 30-in. level is of the order of 13 per cent.

In Fig. 3 are shown the results of tests made with the same three convectors and the one large-tube cast-iron radiator, so adjusted in size that each gave approximately the same equivalent temperature in the No. 3 position in the test room. The difference between the minimum and the maximum amount of heat required to maintain the common equivalent temperature is of the order of 7 per cent.

The Kata thermometer[12], the thermo-integrator[13,14], and the globe[15]

[12]The Kata Thermometer—Its Value and Defects, by W. J. McConnell and C. P. Yagloglou. (Reprint No. 953 from *U. S Public Health Service Report*, pp 2293-2337, September 5, 1924).

[13]The Thermo-Integrator—A New Instrument for the Observation of Thermal Interchanges, by C.-E. A. Winslow and Leonard Greenburg (A S.H V.E. TRANSACTIONS, Vol. 41, 1935, p. 149)

[14]The Calibration of the Thermo-Integrator, by C -E A Winslow, A. P Gagge, Leonard Greenburg, I. M. Moriyama and E J Rodee (*The American Journal of Hygiene*, Vol 22, No. 1, July, 1935, pp. 137-156).

[15]The Globe Thermometer in Studies of Heating and Ventilation, by T. Bedford and C. G. Warner. (*The Journal of Hygiene*, Vol 34, No. 4).

thermometer are other instruments which have been used to measure the influence of air temperature, air movement and radiation in an environment.

The following statements applying to the use of radiators are based on experience and test results:

1. The heating effect of a radiator cannot be judged solely by the amount of steam condensed within the radiator.

2. Smaller floor-to-ceiling temperature differentials can be maintained with long, low, thin, direct radiators, than is possible with high, direct radiators.

3. The larger portion of the floor-to-ceiling temperature differential in a room of average ceiling height heated with direct radiators occurs between the floor and the breathing level.

4. The comfort level (approximately 2 ft-6 in. above floor) is below the breathing line level (approximately 5 ft-0 in. above floor), and temperatures taken at the breathing line may not be indicative of the actual heating effect of a radiator in the room. The comfort-indicating temperature should be taken below the breathing line level.

5. High column radiators placed at the sides of window openings do not produce as comfortable heating effects as long, low, direct radiators placed beneath window openings.

HEATING UP THE RADIATOR AND CONVECTOR

The maximum condensation occurs in a heating unit when the steam s first turned on. Tests[16] on an old-style column-type cast-iron radiator ndicated that in the first 10 min the condensation rate reached a peak of).95 lb per square foot of radiator per hour and 10 to 15 min later lowered o a rate of 0.24 lb. In practice the rate of steam supply to the heating ınit, while heating up, is frequently retarded by controlled elimination of ir through air valves or traps. Automatic control valves may also retard he supply of steam. Vacuum types of air venting valves may be used o reduce the length of the venting periods.

ENCLOSED RADIATORS

The general effect of an enclosure placed about a direct radiator is to estrict the air flow, diminish the radiation and, when properly designed, mprove the heating effect. Investigations[17] indicate that in the design f the enclosure three things should be considered:

1. There should be better distribution of the heat below the breathing line level to roduce greater heating comfort and lowered ceiling temperatures.

2. The lessened steam consumption may not materially change the radiator heating erformance.

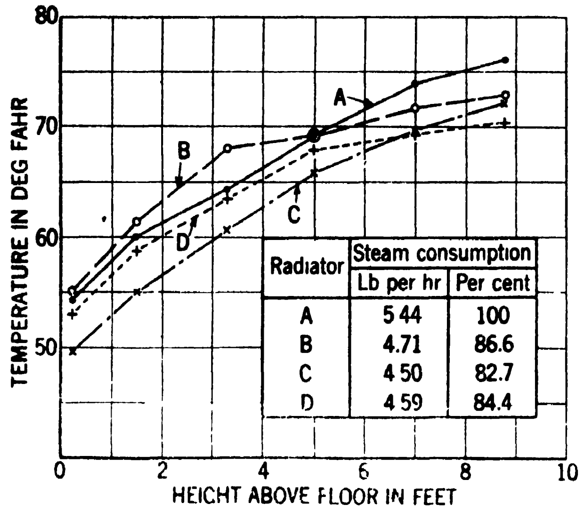
3. The enclosed radiator may inadequately heat the space.

A comparison between a bare or exposed radiator (A) and the same adiator with a well-designed enclosure (B), with a poorly-designed nclosure (C), and with a cloth cover (D) will illustrate the relative

[16] A S H.V.E RESEARCH REPORT No 1067—The Cooling and Heating Rates of a Room with Different ypes of Steam Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock and E L Broderick A S H.V E. TRANSACTIONS, Vol 43, 1937, p 389).

[17] Loc. Cit. Note 10.

heating effects. In Fig. 4 the curve (*B*) reveals that the enclosed radiator used less steam than the exposed radiator, but gave a satisfactory heating performance. A well-designed shield placed over a radiator gives about the same heating effect. Curve (*C*) shows the unsatisfactory effects produced by improperly-designed enclosures. Curve (*D*) shows that the



Radiator	Steam consumption	
	Lb per hr	Per cent
A	5 44	100
B	4.71	86.6
C	4 50	82.7
D	4 59	84.4

Fig. 4. Steam Consumption of Exposed and Concealed Radiators

effect of a cloth cover extending downward 6 in. from the top of the radiator was to make the performance unsatisfactory and inadequate.

Some commercial enclosures and shields for use on direct radiators are equipped with water pans for the purpose of adding moisture to the air in the room. Tests[18] show that an average evaporative rate of about 0.235 lb per square foot of water surface per hour may be obtained from such pans, when the radiator is steam hot and the relative humidity in the room is between 25 and 40 per cent. This source of supply of moisture alone is not adequate to maintain a relative humidity above 25 per cent on a zero day.

[18]University of Illinois, *Engineering Experiment Station Bulletin* No. 230, p 20.

CHAPTER 14

Steam Heating Systems and Piping

Classification, Piping for Steam Heating Systems, Steam Flow, Pipe Sizes, Indirect Heating Units, Types of Heating Systems, High Pressure Steam Systems, Boiler Connections, Condensation Return Pumps, Vacuum Pumps, Traps, Control Valves, Connections to Heating Units

STEAM heating systems may be classified as *gravity* or *mechanical* according to the method of returning the condensation to the boiler. They may also be classified according to any one of, or combination of the following features: the pipe arrangement, the accessories used, the method of expelling or removing the air from the system, the type of control used, and the pressure or vacuum conditions obtained in operation

GRAVITY AND MECHANICAL RETURN

In *gravity systems* the condensate is returned by gravity due to the static head of water in the return pipes or mains. The elevation of the boiler water line must be sufficiently below the lowest heating unit, steam pipe or dry return pipe to permit the return by gravity. The *water line difference* forming the *static head* must be sufficient to overcome the maximum pressure drop in the system, including the pressure drop due to the condensing effect of the radiation. When radiator and drip traps are used, as in two-pipe vapor systems, the static pressure must also exceed the operating pressure of the boiler. The pressure drop caused by condensing rate of the radiation is especially important during those portions of the operating periods where changing pressure conditions prevail, as for example, when the system is being initially filled with steam. In systems where the condensate is wasted to the sewer, no water line difference is required. However, the waste of condensate may introduce conditions which warrant the use of an appropriate mechanical return system. Whenever the conditions of a heating system are such that the returns cannot gravitate to the boiler, they must be returned by some mechanical means.

In *mechanical systems* the condensate flows to a receiver by gravity and is then forced into the boiler against its pressure. In all instances the preferable practice is to provide for gravity flow even where a vacuum pump is used. The lowest parts of the supply side of the system must be kept sufficiently above the water line of the receiver to insure adequate drainage of water from the system.

There are three general types of mechanical return devices in common use, namely, (1) the mechanical return trap, (2) the condensation return pump, and (3) the vacuum return line pump.

PIPING FOR STEAM HEATING SYSTEMS

The functions of the piping system are the distribution of the steam, the return of the condensate and, in systems where no local air vents are provided, the removal of the air. The distribution of the steam should be rapid, uniform and without noise, and the release of air should be facilitated as much as possible, as an air bound system will not heat readily nor properly. In designing the piping arrangement it is desirable to

maintain equivalent resistances in the supply and return piping to and from a radiator. Arranging the piping so the total distance from the boiler to the radiation is the same as the return piping distance from the heating unit back to the boiler tends to obtain such a result. The condensation which occurs in steam piping as well as in radiators must be drained to prevent impeding the ready flow of the steam and air. The effect of back pressure in the returns and excessive revaporization, such as occurs where condensation is released from pressures considerably higher than the vacuum or pressure in the return, must be avoided.

It is important that steam piping systems distribute steam not only at full *design* load but during excess and partial loads. Usually the average winter steam demand is less than half of the demand at the *design* outside temperature. Moreover, in rapidly warming up a system even in moderate weather, the load on the steam main and returns may exceed the

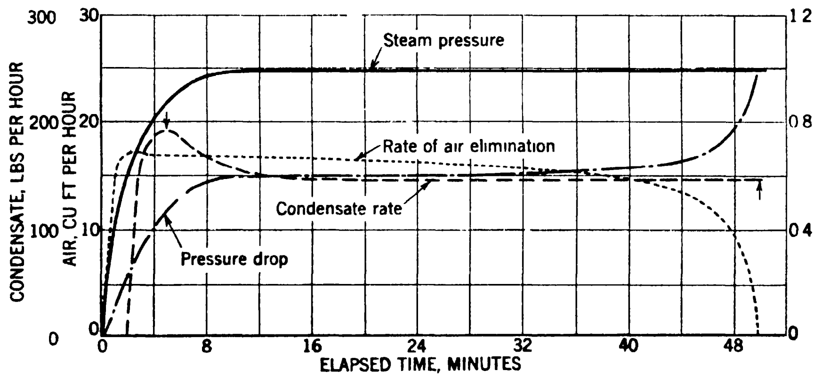


FIG 1 RELATION BETWEEN ELAPSED TIME, STEAM PRESSURE, CONDENSATE AND AIR ELIMINATION RATES

maximum operating load for severe weather due to the necessity of raising the temperature of the metal in the system to the steam temperature and the building to the *design* indoor temperature. Investigations of the return of condensation have revealed that as high as 143 per cent of the design condensation rate may exist under conditions of actual operation.

The piping design of a heating system is greatly influenced by its operating characteristics. Heating systems do not operate under constant conditions as they are continually changing due to variation in load. As the system is being filled with steam the pressures existing in various locations may be different from those which exist for appreciable periods at other locations although at equilibrium conditions the pressures are approximately the same. In designing piping it is of especial importance to arrange the system to preclude trouble caused by such pressure differences. The systems which readily release the air permit uniform pressures to be attained in much shorter time intervals than those which are sluggish. Results are given in Fig. 1 from investigations[1] to determine the rate of condensate and air return from a two-pipe gravity heating system. Variations in the steam pressure during the warming up period

[1]A.S H.V.E. RESEARCH REPORT No. 954—Condensate and Air Return in Steam Heating Systems, by F. C. Houghten and J. L. Blackshaw (A.S H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 199).

when the rate of air elimination and condensation is high are clearly indicated in these curves.

It is evident that the condensation flow during the initial warming-up period reaches a peak which is greater than the constant condensation rate which is eventually reached when the pressure becomes uniform. Moreover, the peak condensation rate is obtained when the system steam pressure is lower than that existing during a period of constant condensing rate. It will also be noted that the peak rate of air elimination does not coincide with the higher condensing rate.

STEAM FLOW

The rate of flow of dry steam or steam with a small amount of water flowing in the same direction is in accordance with the general laws of gas flow and is a function of the length and diameter of the pipe, the density of the steam, and the pressure drop through the pipe. This relationship has been established by Babcock in the formula given at the top of Table 1. In Columns 1, 2, 3, and 4 of this table, the numerical values of the factors for different pressure losses, pipe diameters, steam densities and lengths of pipe have been worked out in convenient form so that the steam flowing in any pipe may be calculated by multiplying together the proper factors in each column as shown in the example at the bottom of the table.

PIPE SIZES

The determination of pipe sizes for a given load in steam heating depends on the following principal factors:

1. The initial pressure and the total pressure drop which may be allowed between the source of supply and the end of the return system.

2. The maximum velocity of steam allowable for quiet and dependable operation of the system, taking into consideration the direction of condensate flow.

3. The equivalent length of the run from the boiler or source of steam supply to the farthest heating unit.

Initial Pressure and Pressure Drop

Theoretically there are several factors to be considered, such as initial pressure and pressure required at the end of the line, but it is most important that (1) the total pressure drop does not exceed the initial pressure of the system and in actual practice should never exceed one-half of the initial pressure; (2) the pressure drop is not so great as to cause excessive velocities; (3) there is a constant initial pressure, except on systems specially designed for varying initial pressures, such as the sub-atmospheric which normally operate under controlled partial vacua, the orifice, and the vapor systems which at times operate under such partial vacua as may be obtained due to the condition of the fire; and (4) the equivalent head due to pressure drop does not exceed the difference in level, for gravity return systems, between the lowest point on the steam main, the heating units, or the dry return, and the boiler water line.

All systems should be designed for a low initial pressure and a reasonably small pressure drop for two reasons: *first*, the present tendency in steam heating unmistakably points toward a constant lowering of pressures even to those below atmospheric; *second*, a system designed in this manner will operate under higher pressures without difficulty. When a system designed for a relatively high initial pressure and a relatively high pressure drop is operated at a lower pressure, it is likely to be noisy and have poor circulation.

Table 1. Flow of Steam in Pipes

P = loss in pressure in pounds.
D = inside diameter of pipe in inches.
L = length of pipe in feet.
d = weight of 1 cu ft of steam.
W = pounds of steam per hour.

$$W = 5220\sqrt{\frac{PdD^5}{\left(1+\frac{3.6}{D}\right)L}}$$

$$P = 0.0000000367\left(1+\frac{3.6}{D}\right)\frac{W^2L}{dD^5}$$

Pressure Loss in Ounces	Col 1	Pipe Size		Internal Area of Pipe Sq Inches	Col 2	Steam Press. Psig	Col 3	Length of Pipe in Feet	Col 4
	$5220\sqrt{\frac{P}{100}}$	Nominal	Actual Internal Diameter		$\sqrt{\frac{D^5}{1+\frac{3.6}{D}}}$		$\sqrt{d}$		$\sqrt{\frac{100}{L}}$
0.25	65.28	1	1.049	0.864	0.536	−1.0[a]	0.187	20	2.240
0.50	92.28	1¼	1.380	1.496	1.178	−0.5[a]	0.190	40	1.580
1.00	130.5	1½	1.610	2.036	1.828	0.0	0.193	60	1.290
2	184.6	2	2.067	3.356	3.710	0.3	0.195	80	1.120
3	226.0	2½	2.469	4.788	6.109	1.3	0.201	100	1.000
4	261.0	3	3.068	7.393	11.183	2.3	0.207	120	0.912
5	291.8	3½	3.548	9.887	16.705	5.3	0.223	140	0.841
6	319.7	4	4.026	12.730	23.631	10.3	0.248	160	0.793
7	345.3	4½	4.506	15.947	32.134	15.3	0.270	180	0.741
8	369.1	5	5.047	20.006	43.719	20.3	0.290	200	0.710
10	412.7	6	6.065	28.886	71.762	30.3	0.326	250	0.632
12	452.0	7	7.023	38.743	106.278	40.3	0.358	300	0.578
14	488.3	8	7.981	50.027	149.382	50.3	0.388	350	0.538
16	522.0	9	8.941	62.786	201.833	60.3	0.415	400	0.500
20	583.6	10	10.020	78.854	272.592	75.3	0.452	450	0.477
24	639.3	12	12.000	113.098	437.503	100.3	0.507	500	0.447
28	690.5	14	13.250	137.880	566.693	125.3	0.557	600	0.407
32	738.2	16	15.250	182.655	816.872	150.3	0.603	700	0.378
40	825.4					175.3	0.645	800	0.354
48	904.1					200.3	0.685	900	0.333
80	1167.2							1000	0.316
160	1650.7							1200	0.289
320	2334.5							1500	0.258
480	2859.1							2000	0.224

Column 1 × 2 × 3 × 4 = lb of steam per hour that will flow through a straight pipe for a given condition.

Example 1: 1 oz drop − 2 in. pipe − 1.3 lb press. − 100 ft equivalent length:

130.5 × 3.710 × 0.201 × 1 = 97.2 lb per hour.
97.2 × 4[b] = 388.8 sq ft equivalent radiation.

Table 1 does not allow for entrained water in low-pressure steam, condensation in covered pipe and roughness in commercial pipe as found in practice.

[a] Pounds per square inch gage = 2.04 in. Vacuum, Mercury Column.

[b] The factor 4 is the approximate equivalent in square feet of steam radiation of 1 lb of steam per hour.

The total pressure drop should never exceed one-half of the initial pressure when condensate is flowing in the same direction as the steam. Where the condensate must flow counter to the steam, the governing factor is the velocity permissible without interfering with the condensate flow. A.S.H.V.E. Research Laboratory experiments limit this to the capacities given in Table 2 for horizontal pipes at varying grades.

Maximum Velocity

The capacity of a steam pipe in any part of a steam system depends upon the quantity of condensation present, the direction in which the condensate is flowing, and the pressure drop in the pipe. Where the quantity of condensate is limited and is flowing in the same direction as the steam, only the pressure drop need be considered. When the condensate must flow against the steam, even in limited quantity, the velocity of the steam must not exceed limits above which the disturbance between the steam and the counter-flowing water may produce objectionable sounds, such as water hammer, or may result in the retention of water in certain parts of the system until the steam flow is reduced sufficiently to permit the water to pass. The velocity at which such disturbances take place is a function of (1) the pipe size, whether the pipe runs horizontally or vertically, (2) the pitch of the pipe if it runs horizontally, (3) the quantity of condensate flowing against the steam, and (4) freedom of the piping from water pockets which under certain conditions act as a restriction in pipe size.

TABLE 2. COMPARATIVE CAPACITY OF STEAM LINES AT VARIOUS PITCHES FOR STEAM AND CONDENSATE FLOWING IN OPPOSITE DIRECTIONS[a]

Pitch of Pipe in Inches per 10 Ft

PITCH OF PIPE	¼ IN		½ IN.		1 IN.		1½ IN.		2 IN		3 IN		4 IN.		5 IN	
Pipe Size Inches	Sq Ft Rad Based on 240 Btu	Max Vel.	Sq Ft Rad Based on 240 Btu	Max Vel.	Sq Ft Rad Based on 240 Btu	Max Vel.	Sq Ft Rad Based on 240 Btu	Max Vel.	Sq Ft Rad Based on 240 Btu	Max Vel	Sq Ft Rad Based on 240 Btu	Max Vel.	Sq Ft Rad Based on 240 Btu	Max Vel	Sq Ft Rad. Based on 240 Btu	Max Vel
¾	25.0	12	30.3	14	37.3	18	40.4	19	42.5	20	46.1	21	47.5	22	49.3	23
1	45.8	12	52.6	15	63.0	17	70.0	20	75.2	22	83.0	23	87.9	25	90.2	26
1¼	104.9	18	117.2	20	133.0	23	144.5	25	154.0	27	165.0	28	172.6	29	178.2	31
1½	142.6	18	159.0	21	181.0	23	196.5	25	209.3	27	224.0	28	234.8	30	242.6	31
2	236.0	19	263.5	20	299.5	23	325.5	25	346.5	27	371.5	28	388.4	29	401.1	30

[a]Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory

Reaming Important

Three factors of uncertainty always exist in determining the capacity of any steam pipe. The first is variation in manufacture, which apparently cannot be avoided. The second is the care used in reaming the ends of the pipe after cutting. The effect of both of these factors increases as the pipe size decreases. According to A.S.H.V.E. Research Laboratory tests, either of these factors may affect the capacity of a 1-in. pipe as much as 20 per cent. The third factor is the uniformity in grading the pipe line. All of the capacity tables given in this chapter include a factor of safety. However, the factor of safety referred to does not cover abnormal defects or constrictions *nor does it cover pipe not properly reamed.*

Equivalent Length of Run

All tables for the flow of steam in pipes, based on pressure drop, must allow for the friction offered by the pipe as well as for the additional

resistance of the fittings and valves. These resistances generally are stated in terms of straight pipe; in other words, a certain fitting will produce a drop in pressure equivalent to so many feet of straight run of the same size of pipe. Table 3 gives the number of feet of straight pipe usually allowed for the more common types of fittings and valves. In all pipe sizing tables in this chapter the *length of run* refers to the *equivalent length of run* as distinguished from the *actual length* of pipe in feet. The length of run is not usually known at the outset; hence it is necessary to assume some pipe size at the start. Such an assumption frequently is considerably in error and a more common and practical method is to

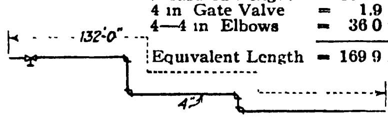
TABLE 3. LENGTH IN FEET OF PIPE TO BE ADDED TO ACTUAL LENGTH OF RUN—OWING TO FITTINGS—TO OBTAIN EQUIVALENT LENGTH

Size of Pipe Inches	Length in Feet to be Added to Run				
	Standard Elbow	Side Outlet Tee	Gate Valve[a]	Globe Valve[a]	Angle Valve[a]
½	1.3	3	0.3	14	7
¾	1.8	4	0.4	18	10
1	2.2	5	0.5	23	12
1¼	3.0	6	0.6	29	15
1½	3.5	7	0.8	34	18
2	4.3	8	1.0	46	22
2½	5.0	11	1.1	54	27
3	6.5	13	1.4	66	34
3½	8	15	1.6	80	40
4	9	18	1.9	92	45
5	11	22	2.2	112	56
6	13	27	2.8	136	67
8	17	35	3.7	180	92
10	21	45	4.6	230	112
12	27	53	5.5	270	132
14	30	63	6.4	310	152

[a]Valve in full open position

Example of length in feet of pipe to be added to actual length of run.

Measured Length	=	132.0 ft
4 in Gate Valve	=	1.9 ft
4—4 in Elbows	=	36.0 ft
Equivalent Length	=	169.9 ft



assume the length of run and to check this assumption after the pipes are sized. For this purpose the length of run usually is taken as double the actual length of pipe.

TABLES FOR PIPE SIZING[2]

One factor determining the size of a steam pipe and its allowable limit of capacity is the direction of the flow of condensate, whether against or with the steam.

Tables 4 and 5 are based on the actual inside diameters of the pipe and the condensation of ¼ lb (4 oz) of steam per square foot of equivalent

[2]Pipe size tables in this chapter have been compiled in simplified and condensed form for the convenience of the user; at the same time all of the information contained in previous editions of THE GUIDE has been retained Values of pressure drops, formerly expressed in ounces, are now expressed in fractions of a pound.

direct radiation[3] (*abbreviated EDR*) per hour. The drops indicated are drops in pressure per 100 ft of equivalent length of run. The pipe is assumed to be well reamed and without unusual or noticeable defects.

Table 4 may be used for sizing piping for steam heating systems by pre-determining the allowable or desired pressure drop per 100 equivalent feet of run and reading from the column for that particular pressure drop. This applies to all steam mains on both one-pipe and two-pipe systems, vapor systems, and vacuum systems. Columns *B* to *G*, inclusive, are used

Table 4. Steam Pipe Capacities

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter *A* through *L*)

This table is based on pipe size data developed through the research investigations of the American Society of Heating and Ventilating Engineers.

	Capacities of Steam Mains and Risers								Special Capacities for One-Pipe Systems Only		
	Direction of Condensation Flow in Pipe Line										
	With the Steam in One-Pipe and Two-Pipe Systems						Against the Steam Two-Pipe Only				
Pipe Size In	1/32 lb or 1/2 Oz Drop	1/24 lb or 2/3 Oz Drop	1/16 lb or 1 Oz Drop	1/8 lb or 2 Oz Drop	1/4 lb or 4 Oz Drop	1/2 lb or 8 Oz Drop	Vertical	Horizontal	Supply Risers Up-Feed	Radiator Valves and Vertical Connections	Radiator and Riser Run-outs
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*[a]	*I*[c]	*J*[b]	*K*	*L*[c]
3/4	....	....	30	....	....	....	30	....	25	....	....
1	39	46	56	79	111	157	56	34	45	28	28
1 1/4	87	100	122	173	245	346	122	75	98	62	62
1 1/2	134	155	190	269	380	538	190	108	152	93	93
2	273	315	386	546	771	1,091	386	195	288	169	169
2 1/2	449	518	635	898	1,270	1,800	635	395	464	....	260
3	822	948	1,160	1,650	2,330	3,290	1,130	700	....	....	475
3 1/2	1,230	1,420	1,740	2,460	3,470	4,910	1,550	1,150	1,140	....	745
4	1,740	2,010	2,460	3,480	4,910	6,950	2,040	1,700	1,520	....	1,110
5	3,210	3,710	4,550	6,430	9,090	12,900	....	3,150	....	....	2,180
6	5,280	6,100	7,460	10,550	14,900	21,100	....	....	....	....	....
8	11,000	12,700	15,500	21,970	31,070	43,900	....	....	....	....	....
10	20,000	23,100	28,300	40,100	56,700	80,200	....	....	....	....	....
12	32,000	37,100	45,500	64,300	91,000	129,000	....	....	....	....	....
16	61,000	69,700	84,800	121,000	170,000	242,000	....	....	....	....	....
	All Horizontal Mains and Down-Feed Risers						Up-Feed Risers	Mains and Undripped Run-outs	Up-Feed Risers	Radiator Connections	Run-outs Not Dripped

Note.—All drops shown are in pounds per 100 ft of equivalent run—based on pipe properly reamed.

[a]Do not use Column *H* for drops of 1/24 or 1/32 lb; substitute Column *C* or Column *B* as required.

[b]Do not use Column *J* for drop of 1/32 lb except on sizes 3 in. and over; below 3 in. substitute Column *B*.

[c]On radiator runouts over 8 ft long increase one pipe size over that shown in Table 6.

where the steam and condensation flow in the same direction, while Columns *H* and *I* are for cases where the steam and condensation flow in opposite directions, as in risers and runouts that are not dripped. Columns *J*, *K*, and *L* are for one-pipe systems and cover riser, radiator valve and vertical connection sizes, and radiator and runout sizes, all of which are based on the critical velocities of the steam to permit the counter flow of condensation without noise.

[3]As steam system design has materially changed in recent years so that 240 Btu no longer expresses the heat of condensation from a square foot of radiator surface per hour, and as present day heating units have different characteristics from older forms of radiation, it is the purpose of The Guide to gradually eliminate the empirical expression *square foot of equivalent direct radiation*, EDR, and to substitute a logical unit based on the Btu. The new terms to express the equivalent of 1000 Btu (Mb), and 1000 Btu per hour (Mbh), have been approved by the A.S.H.V.E.

Table 5. Return Pipe Capacities

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter *M* through *EE*)

This table is based on pipe size data developed through the research investigations of the American Society of Heating and Ventilating Engineers.

Pipe Size Inches	Capacity of Return Mains and Risers																	
	Mains																	
	1/32 Lb or 1/2 Oz Drop per 100 Ft			1/24 Lb or 2/3 Oz Drop per 100 Ft			1/16 Lb or 1 Oz Drop per 100 Ft			1/8 Lb or 2 Oz Drop per 100 Ft			1/4 Lb or 4 Oz Drop per 100 Ft			1/2 Lb or 8 Oz Drop per 100 Ft		
	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac	Wet	Dry	Vac.	Wet	Dry	Vac
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*
3/4	......	......	......	......	......	326	......	......	400	......	......	568	......	......	800	......	......	1,130
1	500	248	......	580	285	570	700	320	700	1,000	412	994	1,400	460	1,400	......	......	1,980
1 1/4	850	520	......	990	595	976	1,200	670	1,200	1,700	868	1,700	2,400	962	2,400	......	......	3,390
1 1/2	1,350	822	......	1,570	943	1,550	1,900	1,060	1,900	2,700	1,360	2,700	3,800	1,510	3,800	......	......	5,370
2	2,800	1,880	......	3,240	2,140	3,260	4,000	2,300	4,000	5,600	2,960	5,680	8,000	3,300	8,000	......	......	11,300
2 1/2	4,700	3,040	......	5,300	3,470	5,450	6,700	3,800	6,700	9,400	4,900	9,510	13,400	5,450	13,400	......	......	18,900
3	7,500	5,840	......	8,500	6,250	8,710	10,700	7,000	10,700	15,000	9,000	15,200	21,400	10,000	21,400	......	......	30,200
3 1/2	11,000	7,880	......	13,200	8,800	13,000	16,000	10,000	16,000	22,000	12,900	22,700	32,000	14,300	32,000	......	......	45,200
4	15,500	11,700	......	18,300	13,400	18,000	22,000	15,000	22,000	31,000	19,300	31,200	44,000	21,500	44,000	......	......	62,190
5	......	......	......	......	......	31,500	......	......	38,700	......	......	54,900	......	......	77,400	......	......	109,000
6	......	......	......	......	......	50,450	......	......	62,000	......	......	88,000	......	......	124,000	......	......	175,000
	Risers																	
3/4	......	190	......	......	190	570	......	190	700	......	190	994	......	190	1,400	......	......	1,980
1	......	450	......	......	450	976	......	450	1,200	......	450	1,700	......	450	2,400	......	......	3,390
1 1/4	......	990	......	......	990	1,550	......	990	1,900	......	990	2,700	......	990	3,800	......	......	5,370
1 1/2	......	1,500	......	......	1,500	3,260	......	1,500	4,000	......	1,500	5,680	......	1,500	8,000	......	......	11,300
2	......	3,000	......	......	3,000	5,450	......	3,000	6,700	......	3,000	9,510	......	3,000	13,400	......	......	18,900
2 1/2	......	......	......	......	......	8,710	......	......	10,700	......	......	15,900	......	......	21,400	......	......	30,200
3	......	......	......	......	......	13,000	......	......	16,000	......	......	22,700	......	......	32,000	......	......	45,200
3 1/2	......	......	......	......	......	17,900	......	......	22,000	......	......	31,200	......	......	44,000	......	......	62,200
4	......	......	......	......	......	31,500	......	......	38,700	......	......	54,900	......	......	77,400	......	......	109,000
5	......	......	......	......	......	50,500	......	......	62,000	......	......	88,000	......	......	124,000	......	......	175,000

Return piping may be sized with the aid of Table 5 where pipe capacities for wet, dry, and vacuum return lines are shown for the pressure drops per 100 ft corresponding to the drops in Table 4. *It is customary to use the same pressure drop on both the steam and return sides of a system.*

Example 2. What pressure drop should be used for the steam piping of a system if the measured length of the longest run is 500 ft and the initial pressure is not to be over 2-lb gage?

Solution. It will be assumed, if the measured length of the longest run is 500 ft, that when the allowance for fittings is added the equivalent length of run will not exceed 1,000 ft. Then, with the pressure drop not over one half of the initial pressure, the drop could be 1 lb or less. With a pressure drop of 1 lb and a length of run of 1,000 ft, the drop per 100 ft would be $\frac{1}{10}$ lb, while if the total drop were $\frac{1}{2}$ lb, the drop per 100 ft would be $\frac{1}{20}$ lb. In the first instance the pipe could be sized according to Column *D* for $\frac{1}{16}$ lb per 100 ft, and in the second case, the pipe could be sized according to Column *C* for $\frac{1}{24}$ lb. On completion of the sizing, the drop could be checked by taking the longest line and actually calculating the equivalent length of run from the pipe sizes determined. If the calculated drop is less than that assumed, the pipe size is all right; if it is more, it is probable that there are an unusual number of fittings involved, and either the lines must be straightened or the column for the next lower drop must be used and the lines resized. Ordinarily resizing will be unnecessary.

PIPE SIZING FOR INDIRECT HEATING UNITS

Pipe connections and mains for indirect heating units are sized in a manner similar to radiators, but the equivalent direct radiation must be ascertained for each row of heating unit stacks and then must be divided into the number of stacks constituting that row and into the number of connections to each stack.

$$EDR = \frac{Q \times 60 \times (t_1 - t_e)}{55.2 \times 240} = \frac{Q \times (t_1 - t_e)}{220.8} \quad (1)$$

where

EDR = equivalent direct radiation, square feet.

Q = volume of air, cubic feet per minute.

t_e = the temperature of the air entering the row of heating units under consideration, degrees Fahrenheit.

t_1 = the temperature of the air leaving the row of heating units under consideration, degrees Fahrenheit.

60 = the number of minutes in one hour.

55.2 = the number of cubic feet of air heated 1 F by 1 Btu

240 = Btu equivalent of 1 sq ft of EDR.

Example 3 Assume that a 3-row heating unit shown in this chapter in Fig. 38 is handling 50,000 cfm of air and that the rise in the first row is from 0 to 40 F, in the second row from 40 to 65 F, and in the third row from 65 to 80 F. What is the load in EDR on each supply and return connection?

Solution. For row 1,

$$R = \frac{50{,}000 \times (40 - 0)}{220.8} = 9058 \text{ sq ft.}$$

For row 2,

$$R = \frac{50{,}000 \times (65 - 40)}{220.8} = 5661 \text{ sq ft.}$$

For row 3,

$$R = \frac{50{,}000 \times (80 - 65)}{220.8} = 3397 \text{ sq ft.}$$

Each row of heating units consists of four stacks and each stack has two connections so that the load on each stack and each connection of the stack is as follows:

Row	Total Load (EDR)	Stack Load[a] (EDR)	Connection Load[b] (EDR)
1	9058	2265	2265 or 1132
2	5661	1415	1415 or 708
3	3397	849	849 or 425

[a]One quarter of total row load.
[b]One half of stack load if two steam connections are made; otherwise, same as stack load.

The pipe sizes would then be based on the length of the run and the pressure drop desired, as in the case of radiators. It generally is considered desirable to place the indirect heating units on a separate system and not on supply or return lines connected to the general heating system.

GRAVITY ONE-PIPE AIR-VENT SYSTEM

This system is the most common of all methods of steam heating, especially for small size installations, due largely to its low cost and simplicity.

The downward pitch of a one-pipe air-vent system is indicated in Fig. 2. Low points and ends of steam mains pitched down from the boiler should be dripped. All drips should be sealed below water line before connecting together. In the risers and radiator connections, steam and condensation flow in opposite directions. In long steam mains it flows in the same direction as the steam and is removed from the main through the drip. Short mains may be arranged for the condensate to flow in a direction opposite the steam by sizing them so the critical velocity is not exceeded. It is customary to drip the heel of each riser in buildings of several stories to avoid counter-flow of the steam and condensate in the riser branch. In buildings of one or two stories the condensate is returned

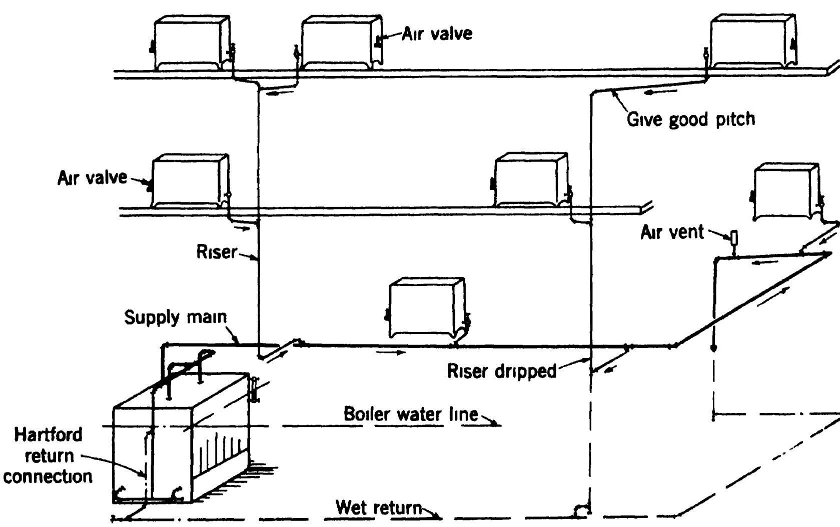


Fig. 2. Typical Up-Feed Gravity One-Pipe Air-Vent System

to the steam main instead of being dripped. Both types of risers are shown in Fig. 2, and riser connections are shown in Figs. 3 and 4. A typical overhead down-feed system is illustrated in Fig. 5. While wet return mains need not be pitched toward the boiler to maintain steam circulation, they should be pitched for drainage.

To improve steam circulation in one-pipe systems quick vent air valves should be provided at the ends and at intermediate points where the

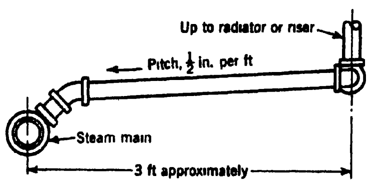


FIG. 3. TYPICAL STEAM RUNOUT WHERE RISERS ARE NOT DRIPPED

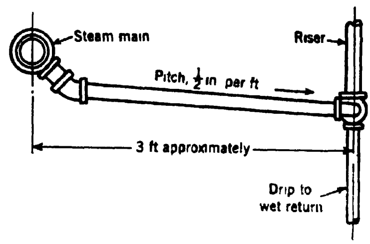


FIG. 4. TYPICAL STEAM RUNOUT WHERE RISERS ARE DRIPPED

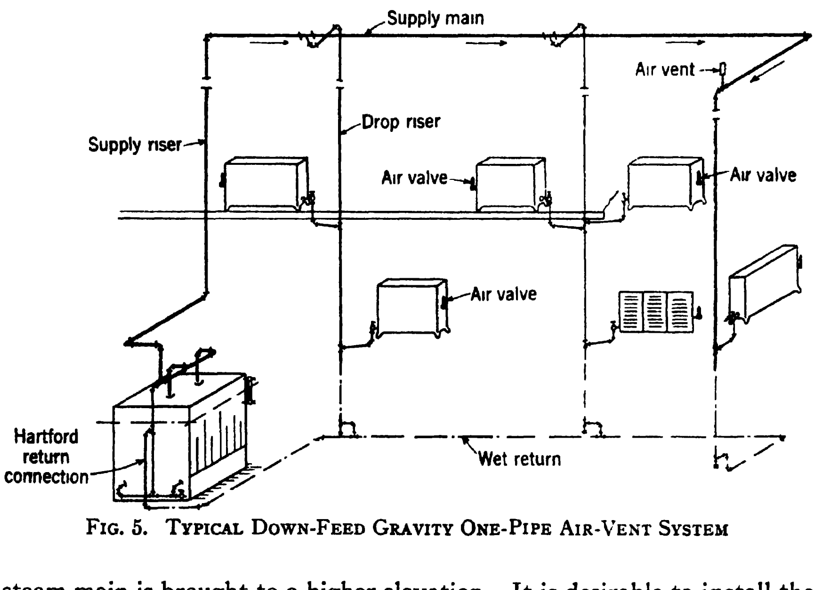


FIG. 5. TYPICAL DOWN-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

steam main is brought to a higher elevation. It is desirable to install the air-vent valves about a foot ahead of the drips, as indicated in Fig. 2, to prevent possible damage to their mechanisms by water.

The radiator valves may be the angle-globe, offset-corner pattern or gate type. Straight-globe and straight-corner type should not be used since the damming effect of the raised valve seat would interfere with the flow of condensation through the valve. Graduated valves cannot be used since the steam valves on this system must be fully open or fully closed to prevent the radiators filling with water and creating a dangerous water line condition. With a one-pipe system the heat cannot be modulated at the radiator, the steam being either *all on* or *all off*. Systems and

devices are available which make it possible to obtain a partial modulating effect from one-pipe heating systems.

It is important to keep the lowest points of the steam mains and heating units sufficiently above the water line of the boiler to prevent flooding. The minimum water line difference depends on the initial steam pressure and piping pressure drop plus a safety factor for heating up,

Referring to Fig. 6 it will be noted that the water in the wet return is a U-shaped container, with the boiler steam pressure on the top of the water at one end and the steam main pressure on the top of the water at the other end. The difference between these two pressures is the *pressure drop* in the system, *i.e.*, the friction and resistance to the flow of steam in passing from the boiler to the far end of the main and the pressure reduction in consequence of the condensation occurring in the system. The water in the far end will rise sufficiently to overcome this difference in order to balance the pressures, and it will rise far enough to produce a flow through the return pipe and overcome the resistance of check valves if installed.

If a one-pipe steam system is designed, for example, for a total pressure drop of $1\frac{1}{8}$ lb, and utilizes a Hartford return connection instead of a check valve on the return, the rise in the water level at the far end of the return due to the difference in steam pressure would be $\frac{1}{8}$ of 28 in. (28 in. head being equal to one pound per square inch), or $3\frac{1}{2}$ in. Adding 3 in. to overcome the resistance of the return main and 6 in. as a factor of safety

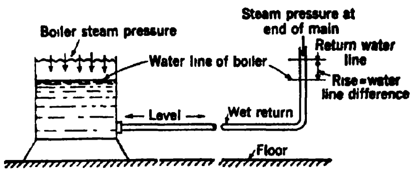


FIG. 6. DIFFERENCE IN STEAM PRESSURE ON WATER IN BOILER AND AT END OF STEAM MAIN

for heating up gives $12\frac{1}{2}$ in. as the distance the bottom of the lowest part of the steam main and all heating units must be above the boiler water line. The same system, however, installed and sized for a total pressure drop of $\frac{1}{2}$ lb, and with a check in the return, would require $\frac{1}{2}$ of 28 in., or 14 in. for the difference in steam pressure, 3 in. for the flow through the return, 4 in. to operate the check, and 6 in. for a factor of safety, making a total of 27 in. as the required distance. Higher pressure drops would increase the distance accordingly.

Gravity one-pipe air-vent systems in which the equivalent length of run does not exceed 200 ft should be sized by means of Tables 4 and 5 as follows:

1. *For the steam main and dripped runouts to risers* where the steam and condensate flow in the same direction, use $\frac{1}{16}$-lb drop (Column *D*).

2. *Where the riser runouts are not dripped* and the steam and condensation flow in opposite directions, and *also in the radiator runouts* where the same condition occurs, use Column *L*.

3. *For up-feed steam risers* carrying condensation back from the radiators, use Column *J*.

4. *For down-feed systems the main risers* of which do not carry any radiator condensation, use Column *H*.

5. *For the radiator valve size and the stub connection*, use Column *K*.

6. For the *dry return main*, use Column *U*.

7. For the *wet return main* use Column *T*.

On systems exceeding an equivalent length of 200 ft, it is suggested that the total drop be not over $\frac{1}{4}$ lb. The return piping sizes should correspond with the drop used on the steam side of the system. Thus, where $\frac{1}{64}$-lb drop is being used, the steam main and dripped runouts would be sized from

Column *C;* radiator runouts and undripped riser runouts from Column *L;* up-feed risers from Column *J*; the main riser on a down-feed system from Column *C* (it will be noted that if Column *H* is used the drop would exceed the limit of $\frac{1}{64}$ lb); the dry return from Column *R;* and the wet return from Column *Q*.

With a $\frac{1}{32}$-lb drop the sizing would be the same as for $\frac{1}{64}$ lb except that the steam main and dripped runouts would be sized from Column *B*, the main riser on a down-feed system from Column *B*, the dry return from Column *O*, and the wet return from Column *N*.

Notes on Gravity One-Pipe Air-Vent Systems

1. Pitch of mains should not be less than ¼ in. in 10 ft.

2. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. in 10 ft. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

TABLE 6. PIPE SIZES FOR ONE-PIPE UP-FEED SYSTEM SHOWN IN FIG 7

Part of System	Section of Pipe	Radiation Supplied EDR Sq Ft	Theoretical Pipe Size (Inches)	Practical Pipe Size (Inches)
Branches to radiators..	....	100	2	2
Branches to radiators..	....	50	1¼	1¼
Riser	*a* to *b*	200	2	2
Riser	*b* to *c*	300	2½	2½
Riser	*c* to *d*	400	2½	2½
Riser	*d* to *e*	500	3	3
Riser	*e* to *f*	600	3	3
Branch to riser	*f* to *g*	600	3½	3½
Supply main	*g* to *h*	600	3	3
Branch to supply main	*h* to *j*	600	2½	3
Dry return main	*f* to *k*	600	1¼	2
Wet return main	*k* to *m*	600	1	2
Wet return main	*m* to *n*	600	1	2
Wet return main	*n* to *p*	600	1	2

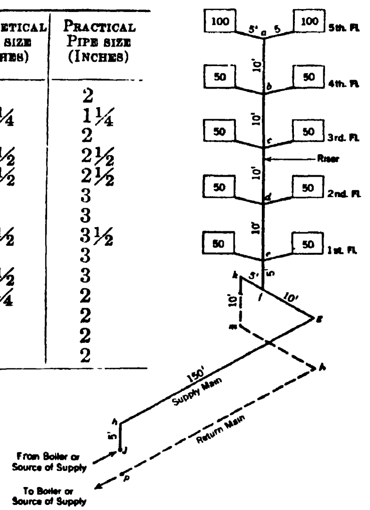


FIG. 7. RISER, SUPPLY MAIN AND RETURN MAIN OF ONE-PIPE SYSTEM

3. In general, it is not desirable to have a main less than 2 in The diameter of the far end of the supply main should not be less than half its diameter at its largest part.

4. Supply mains, branches to risers, or risers, should be dripped where necessary.

5. Where supply mains are decreased in size they should be dripped, or be provided with eccentric couplings, flush on bottom.

Example 4. Size the one-pipe gravity steam system shown in Fig. 7 assuming that this is all there is to the system or that the riser and run shown involve the longest run on the system.

Solution. The total length of run actually shown is 215 ft. If the equivalent length of run is taken at double this, it will amount to 430 ft, and with a total drop of ¼ lb the drop per 100 ft will be slightly less than $\frac{1}{16}$ lb. It would be well in this case to use $\frac{1}{64}$ lb, and this would result in the theoretical sizes indicated in Table 6. These theoretical sizes, however, should be modified by not using a wet return less than 2 in. while the main supply, *g-h*, if from the uptake of a boiler, should be made the full size of the main, or 3 in. Also the portion of the main *k-m* should be made 2 in. if the wet return is made 2 in.

ONE-PIPE VAPOR SYSTEM

The one-pipe vapor system operates under pressures at or near atmospheric and returns its condensation to the boiler by gravity. In this system the automatic air valves are of special design to permit the ready release of air and prevent its ready return after it is expelled. The steam radiator valves are a type which, when opened, give a free and unobstructed passageway for water. The piping is the same as for the one-pipe gravity system but sized so as to permit operation at a few ounces pressure.

TWO-PIPE VAPOR SYSTEM

A two-pipe up-feed vapor system using separate supply and return pipes is shown in Fig. 8. The radiators discharge their condensation

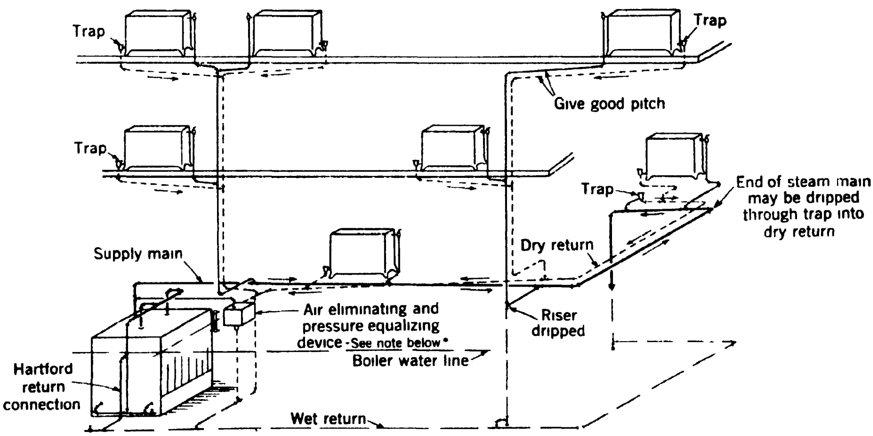


FIG. 8. TYPICAL UP-FEED SYSTEM WITH AUTOMATIC RETURN TRAP[a]

[a]Proper piping connections are essential with special appliances for pressure equalizing and air elimination.

through thermostatic traps to the dry return pipe. These systems operate at a few ounces pressure and above, but those with mechanical condensate return devices may operate at pressures upward of 10 lb. The simplest method of venting the system consists of a ¾-in. pipe with a check valve opening outward. Most systems employ various forms of vent valves, designed to allow the air to readily pass out of the system and to prevent its return. These systems permit control of the heat in the radiator by varying the opening of the graduated radiator valves. The boiler pressure is maintained at substantially constant pressure slightly above atmospheric pressure.

These systems may be classified as (1) *closed systems*, consisting of those which have a device to prevent the return of air after it has once been expelled from the system, and which can operate at both super and subatmospheric pressures for a period of four to eight hours depending upon the tightness of the system and rate of firing, and (2) *open systems*, comprising those which have the return line constantly open to the atmosphere without a check or other means to prevent the return of air. The

open systems are not so popular because they have the disadvantage of not holding heat when the rate of steam generation is diminishing.

Closed systems should preferably be equipped with an automatic return trap to prevent water from backing out of the boiler. In installing the return trap a check valve is inserted in the return main at a point near the boiler and a vertical pipe is run up into the bottom of the return trap, which is usually located with the bottom about 18 in. above the boiler water line. Some traps are constructed so that they will operate when they are installed with their bottom as close as 8 in. above the boiler water line. On the other side of this connection a second check valve is installed in the main return just before it enters the boiler. Fig. 9 shows a typical connection for an automatic return trap.

Down-Feed Two-Pipe Vapor System

In the down-feed two-pipe vapor system the steam is carried to the top of the building, the top of the vertical riser constituting the high point of

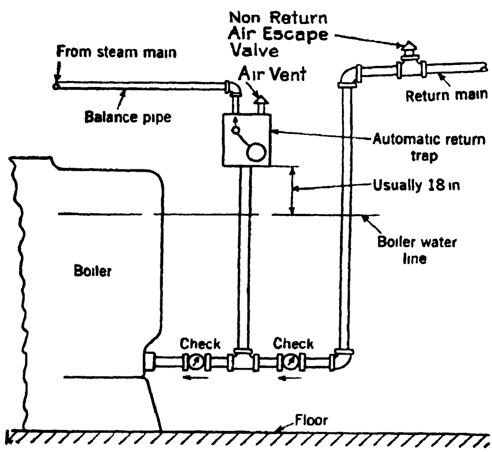


FIG. 9. TYPICAL CONNECTIONS FOR AUTOMATIC RETURN TRAP

the system, and the horizontal supply main is sloped down from this location to the far ends of each branch. The branches are taken off the main from the bottom or at a 45-deg angle downward, with the runouts sloped toward the drops. Thus each branch from the main forms a drip and no accumulation of water is carried down any one drop.

The steam drops are carried down through the building with suitable reductions as the various radiator connections are taken off until the lowest radiator runout is reached. If the drop is only two or three stories high, the portion feeding the bottom radiator should be increased one pipe size to provide for draining the riser, and if the drop is over three stories high it is well to increase the portion feeding the two lowest radiators one or two pipe sizes, especially if the two lowest radiators are small and the normal size of drop required is 1 in. or less. The bottom of each steam drop should terminate with a dirt pocket and be dripped as shown in Fig. 10. The returns on a down-feed vapor system are the same as on an up-feed system. The runouts to the radiators and the radiator connections of the down-feed system are the same as those for the up-feed system already described.

While many manufacturers of patented vapor heating accessories have their own schedules for pipe sizing, an inspection of these sizing tables indicates that in general as small a drop as possible is recommended. The reasons for this are: (1) to have the condensation return to the boiler by gravity, (2) to obtain a more uniform distribution of steam throughout the system, especially when it is desirable to carry a moderate or low fire, and (3) because with large variation in pressure the value of graduated valves on radiators is destroyed.

For small vapor systems where the equivalent length of run does not exceed 200 ft, it is recommended that the main and any runouts to risers that may be dripped should be sized from Column *D*, Table 4, while riser runouts not dripped and radiator runouts should employ Column *I*. The up-feed steam risers should be taken from Column *H*. On the returns, the risers should be sized from Table 5, Column *U*, (lower portion) and the mains from Column *U* (upper portion). It should again be noted that

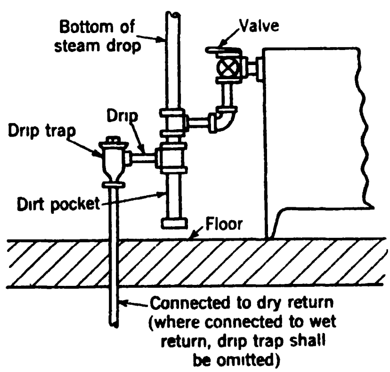


FIG. 10. DETAIL OF DRIP CONNECTIONS AT BOTTOM OF DOWN-FEED STEAM DROP

the pressure drop in the steam side of the system is kept the same as on the return side except where the flow in the riser is concerned.

On a down-feed system the main vertical riser should be sized from Column *H*, but the down-feed risers can be taken from Column *D* although it so happens that the values in Columns *D* and *H* correspond. This will not hold true in larger systems.

For vapor systems over 200 ft of equivalent length, the drop should not exceed 1/8 lb to 1/4 lb, if possible. Thus, for a 400 ft equivalent run the drop per 100 ft should be not over 1/8 lb divided by 4, or 1/32 lb. In this case the steam mains would be sized from Column *B*; the radiator and undripped riser runouts from Column *I*; the risers from Column *B*, because Column *H* gives a drop in excess of 1/32 lb. On a down-feed system, Column *B* would have to be used for both the main riser and the smaller risers feeding the radiators in order not to increase the drop over 1/32 lb. The return risers would be sized from the lower portion of Column *O* and the dry return main from the upper portion of the same column, while any wet returns would be sized from Column *N*. The same pressure drop is applied on both the steam and the return sides of the system.

Notes on Vapor Systems

1. Pitch of mains should not be less than 1/4 in. in 10 ft.
2. Pitch of horizontal runouts to risers and radiators should not be less than 1/2 in.

in 10 ft. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

3. In general it is not desirable to have a supply main smaller than 2 in.

4. When necessary, supply main, supply risers, or branches to supply risers should be dripped separately into a wet return, or may be connected into the dry return through a thermostatic drip trap.

VACUUM SYSTEMS

In the vacuum system, a vacuum is maintained in the return line practically at all times. The pump is usually controlled by a vacuum regulator which operates the pump to maintain the vacuum within limits and operates in response to a pressure difference between the atmosphere

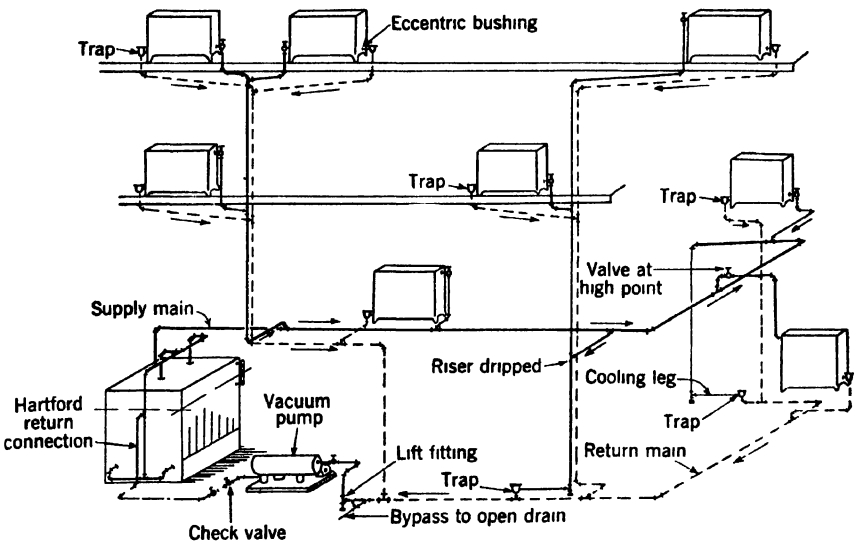


Fig. 11. Typical Up-Feed Vacuum Pump System

and the return to control the vacuum in the return main. The source of steam supply may be a low pressure boiler as shown in Fig. 11, or a high pressure line through a pressure reducing valve. The piping and other details are the same as for the vapor systems.

The return risers are connected in the basement into a common return main which slopes downward toward the vacuum pump. The vacuum pump withdraws the air and water from the system, separates the air from the water and expells it to atmosphere and pumps the water back to the boiler, or other receiver, which may be a feed-water heater or hot well. It is essential that no connection be made from the supply side to the return side at any point except through a trap. The desirable practice demands a return flowing to the vacuum pump by an uninterrupted downward slope. In some instances local conditions make it necessary to drop the return below the level of the vacuum pump inlet, before the pump can be reached. In such an event one of the advantages of the vacuum system is the ability to raise the condensate to a considerable height by the suction of the vacuum pump by means of a lift connection or fitting inserted in the return. The height the condensate can be raised depends on the amount of vacuum maintained. It is preferable to limit lift con-

nections to a single lift at the vacuum pump. A still more preferable arrangement is the use of an accumulator tank, or receiver tank, with a float control for the pump, at the low point of the return main located adjacent to the vacuum pump.

When the vertical lift is considerable, several lift fittings should be used in steps as shown in Fig. 12. This permits a given lift to be secured with a somewhat lower vacuum than where the vertical distance is served by a single lift. Where several lifts are present in a given system at different locations, the lifting cannot occur until the entire system is filled with steam. A lift connection for location close to the pump, where the size may be above the commercial stock sizes, is shown in Fig. 13. It is

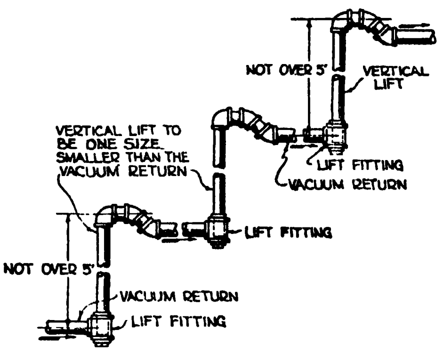


FIG. 12. METHOD OF MAKING LIFTS ON VACUUM SYSTEMS WHEN DISTANCE IS OVER 5 FT

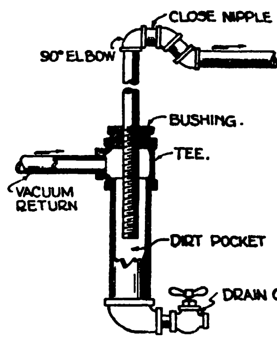


FIG. 13. DETAIL OF MAIN RETURN LIFT AT VACUUM PUMP

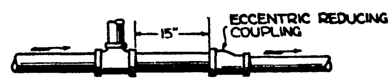


FIG. 14. METHOD OF CHANGING SIZE OF STEAM MAIN WHEN RUNOUTS ARE TAKEN FROM TOP

desirable that means be provided for manually draining the low point of the lift fittings to eliminate danger of freezing.

Down-Feed Vacuum System

The piping arrangement for the down-feed vacuum system is similar on the supply side to the down-feed vapor system in that it has similar runouts, radiator valves, drips on the bottom of the steam drops, and enlargement of the drops for the lower radiator connections. The return side of the system is exactly the same as the up-feed system except that the steam riser drips at the bottom are connected into the return line through thermostatic traps. It is preferable to take the runouts for the risers from the bottom or at a 45-deg angle down from the steam main so that they may serve as steam main drips. When this is done it is practical to run the steam main level if a runout is located at every change in pipe size, or if eccentric fittings are used (Fig. 14). A slight pitch in the steam main, however, should be used when possible. An overhead vacuum down-feed system is shown diagrammatically in Fig. 15.

Vacuum, atmospheric, sub-atmospheric and orifice systems are usually employed in large installations and have total drops varying from ¼ to

½ lb. Systems where the maximum equivalent length does not exceed 200 ft preferably employ the smaller pressure drop while systems over 200 ft equivalent length of run more frequently are designed for the higher drop, owing to the relatively greater saving in pipe sizes. For example, a system with 1200 ft longest equivalent length of run would employ a drop per 100 ft of ½ lb divided by 12, or $^1/_{24}$ lb. In this case the steam main would be sized from Column *C*, Table 4, and the risers also from Column *C* (Column *H* could be used as far as critical velocity is concerned but the drop would exceed the limit of $^1/_{24}$ lb). Riser runouts, if dripped, would use Column *C* but if undripped would use Column *I*; radiator runouts, Column *I*; return risers, lower part of Column *S*, Table 5; return runouts to radiators, one pipe size larger than the radiator trap connections

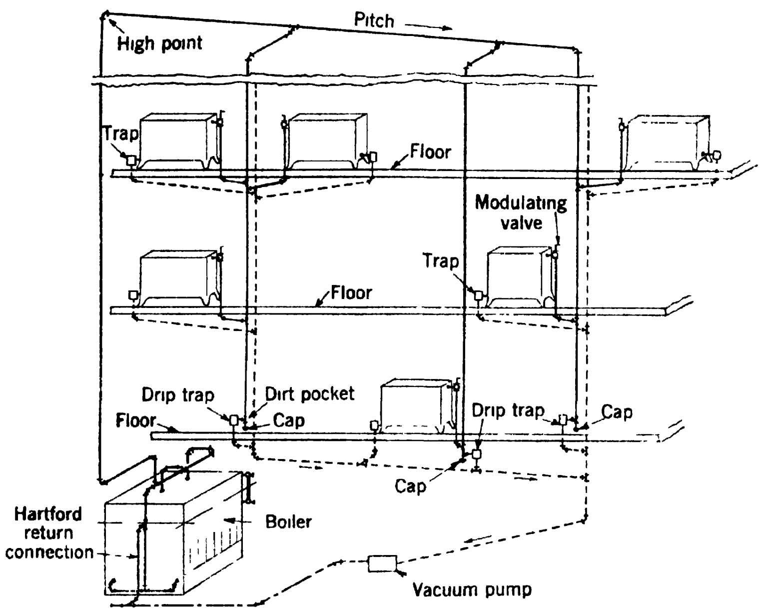


FIG. 15. TYPICAL DOWN-FEED VACUUM SYSTEM

Notes on Vacuum Systems

1. It is not generally considered good practice to exceed ⅛ lb drop per 100 ft of equivalent run nor to exceed 1 lb total pressure drop in any system.

2. Pitch of mains should not be less than ¼ in in 10 ft.

3. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. in 10 ft. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table

4. In general it is not considered desirable to have a supply main smaller than 2 in.

5. When necessary, the supply main, supply riser, or branch to a supply riser should be dripped separately through a trap into the vacuum return. A connection should not be made between the steam and return sides of a vacuum system without interposing a trap to prevent the steam from entering the return line.

6. Lifts should be avoided if possible, but when they cannot be eliminated they should be made in the manner described in this chapter.

7. No lifts can be used in orifice and atmospheric systems. In sub-atmospheric systems the lift must be at the vacuum pump.

SUB-ATMOSPHERIC SYSTEMS

Sub-atmospheric systems are similar to vacuum systems but, in contrast, provide control of building temperature by variation of the heat output from the radiators. The radiator heat emission is controlled by varying the pressure, temperature and volume of steam in circulation. These systems differ from the ordinary vacuum system in that they maintain a controllable partial vacuum on both the supply and return sides of the system, instead of only on the return side. In the vacuum system, steam pressure above that of the atmosphere exists in the supply mains and radiators practically at all times. In the sub-atmospheric system, atmospheric pressure or higher exists in the steam supply piping and radiators only during severe weather. Under average winter temperature the steam is under partial vacuum which in mild weather may reach as high as 25 in. Hg, after which further reduction in heat output is obtained by restricting the quantity of steam.

The rate of steam supply is controlled by a valve in the steam main or by thermostatically controlling the rate of steam production in the boiler. The control valve may be of the automatic *modulating* or *floating* type governed thermostatically from selected control points in the building, or it may be a special pressure reducing valve which will maintain the desired sub-atmospheric pressures by continuous flow into the heating main. All radiator supply valves have incorporated adjustable orifices or are equipped with regulating orifice plates. The sizes of orifices used are larger than for orifice systems because for equal radiator sizes the volume flowing is larger. These orifices are omitted on some systems, depending upon the type of control. Radiator traps and drips are designed to operate at any pressure from 15 lb gage to 26 in. Hg. A vacuum pump capable of operating at high vacuum is preferable to promote accuracy in the distribution of steam throughout the system, particularly in mild weather. This vacuum is partially self-induced by the condensation of the steam in the system under conditions of restricted supply for reduction of the radiator heat emission.

The returns must grade downward constantly and uninterruptedly from the radiator return outlets to the inlet of the receiver of the vacuum pump. One radical difference between this and the ordinary vacuum system is that no lifts should be made in the return line, except at the vacuum pump. The receivers are placed at a lower level than the pump and equipped with float control so the pump may operate as a return pump under night conditions. The system may be operated in the same manner as the ordinary vacuum system when desired.

Steam for heating domestic hot water should be taken from the boiler header back of the control valve so that pressures sufficiently high for heating the water may be maintained on the heater. The sub-atmospheric method of heating can be used for the heating coils of ventilating and air conditioning systems. The flexible control of heat output secured by this method materially reduces the required size of by-pass around the heaters. Some applications of sub-atmospheric systems are proprietary.

ORIFICE SYSTEMS

Orifice systems of steam heating may have piping arrangements identical with vacuum systems. Some of these omit the radiator thermostatic traps but use thermostatic or combination float and thermostatic traps on all drip points. A return condensation pump with receiver vented to atmosphere, a return line vacuum pump, or a return trap, is generally

used to return the condensation to the boiler or place of similar disposition, such as a feed-water heater or hot well. The heat emission from the radiators is controlled by varying the pressure differential maintained.

The principle on which these systems operate is based on the fact that the steam flow through an orifice will vary when the ratio of the absolute pressures on the two sides of the orifice exceeds 58 per cent. If the absolute pressure on the outlet side is less than 58 per cent of the absolute pressure on the inlet side, no further increase in flow will be obtained as a result of the increased pressure difference. If an orifice is so designed in size as to exactly fill a radiator with 2 lb gage on one side and ¼ lb gage on the other, the absolute pressure relation is:

$$\frac{14.7 + 0.25}{14.7 + 2.0} = 0.90 \text{ or } 90 \text{ per cent.}$$

Should the steam pressure be dropped to ¼ lb on the supply pipe, the pressure on each side of the orifice would be balanced and no steam flow would take place. From this it will be apparent that if an orifice of a given diameter will fill a given radiator with steam when there is a given pressure on the main, reducing this steam main pressure will permit filling various desired portions of the radiator down to the point where the main pressure equals the back pressure in the radiator provided the supply pipe pressures may be controlled sufficiently close. If orifices are designed on a similar basis for a given system and proportioned to the heating capacity of the radiators they serve, all radiators will heat proportionately to the steam pressure. The range of pressure variation is limited by the permissible noise level of the steam flowing under the pressure difference required for maximum heat output. The control of the steam supply is obtained by a valve placed in the steam main, which maintains a determined pressure by varying the vacuum in the return lines, or by varying the pressure in the supply lines and the vacuum in the returns. The valves are frequently manually set from a remote location, guided by temperature indicating stations in the building; or thermostatically controlled from a thermostat on the roof, which automatically measures the differential of outside and inside temperatures. Since the range through which the pressures may be varied is usually from 0 to 4.0 lb gage, the control should be capable of maintaining close regulation to maintain the desired space temperatures, particularly in mild weather.

A recommended orifice schedule is shown in Table 7. Some systems use orifices not only in radiator inlets but also at different points in the steam supply piping for the purpose of balancing the system to a greater extent. In this manner the difference between the initial and terminal pressure in the steam main may be compensated to a great extent. For example, if the initial pressure was 3 lb gage and the pressure at the end of the main was 2 lb, an orifice could be used in each branch for the purpose of obtaining a more uniform pressure throughout the system. Such a provision may be particularly useful in this system for branches close to the boiler where the drop in the main has not yet been produced. Some orifice systems are proprietary.

HIGH PRESSURE STEAM SYSTEMS

Many of the recent installations of heating systems for large industrial type buildings have been designed for the use of high pressure steam, that is, without the use of pressure reducing valves. Such systems usually involve the use of unit heaters or large built-up fan units with blast

heating coils. Pressures on these systems vary from 30 to 150 psi. Temperatures are controlled by a modulating or throttling type thermostatic valve controlled by the air temperature in the fan outlet.

Tables 8 to 11 may be used for the sizing of steam and return piping for systems of 30 and 150 psi pressure at various pressure drops. These tables are based on Babcock's formula, and have been used as the basis of design for a number of years. Capacities at other pressures may be computed by means of Table 1.

The steam lines can be sized for greater pressure drops than the return piping. For a system using steam at 30 psi pressure, the total pressure drop can be 5 to 10 lb, and for 150-psi systems 25 to 50 lb.

It has been observed that the maximum pressure in the returns of a

TABLE 7. ORIFICE CAPACITIES

Capacity Expressed in Square Feet Equivalent Direct Radiation

This table is based on data from actual tests[a]

Orifice Diameter 64ths of an Inch	6 In Hg Differential	5 In Hg Differential	4 In Hg Differential	2 In Hg Differential	1 In Hg Differential
7	18–23	16–21	15–19	10–13	
8	23–29	21–27	19–25	13–17	8–11
9	29–36	27–33	25–30	17–21	11–14
10	36–44	33–40	30–37	21–26	14–17
11	44–52	40–48	37–44	26–31	17–20
12	52–62	48–57	44–51	31–37	20–24
13	62–72	57–66	51–59	37–43	24–28
14	72–83	66–76	59–67	43–49	28–32
15	83–94	76–86	67–76	49–56	32–37
16	94–106	86–97	76–86	56–64	37–42
17	106–119	97–109	86–97	64–72	42–47
18	119–133	109–122	97–108	72–80	47–52
19	133–148	122–135	108–120	80–88	52–58
20	148–163	135–149	120–133	88–98	58–64
21	163–179	149–164	133–145	98–107	64–71

Note—The radiator orifice plates recommended in this table are made of brass stampings 0.023 in. thick, cup-shaped to be inserted in radiator valve unions.

[a] Flow of steam through Orifices into Radiators, S. S Sanford and C. B. Sprenger (A.S.H.V.E TRANSACTIONS, Vol. 37, 1931, p. 371).

30-psi system is about 5 lb, and that of a 150-psi system is about 20 lb. The pressure in the return mains is of course caused by the discharge of traps or by leaky traps and by flashing due to the lower pressure in the return line. The total pressure drop in the returns can be approximately 2 lb for the 30-psi returns, and about 10 lb for the 150-psi returns. The usual practice in the sizing of high pressure returns has been to size on the basis of ½ lb per 100 ft of pipe for 30-psi systems, and 1 lb per 100 ft for 150-psi systems. This is an average figure which corresponds generally to several of the previously published tables for the design of high pressure return piping.

The returns generally discharge to a vented receiver. It is, of course, necessary to provide for the elimination of air from high pressure systems, the same as in low pressure systems. If possible, it is desirable to design the system so as to condense some of the steam escaping through the vents by passing it through a heat exchanger.

When high pressure steam is being supplied and lower steam pressures are required for use in heating, domestic hot water, utility services, etc.,

Table 8. Steam Pipe Capacities for 30 psi Steam Systems

Capacity Expressed in Pounds per Hour

(Steam and Condensate Flowing in Same Direction)

Pipe Size Inches	Drop in Pressure—Pounds per 100 Ft in Length					
	⅛	¼	½	¾	1	2
¾	15	22	31	38	45	63
1	31	46	63	77	89	125
1¼	69	100	141	172	199	281
1½	107	154	219	267	309	437
2	217	313	444	543	627	886
2½	358	516	730	924	1,033	1,460
3	651	940	1,330	1,628	1,880	2,660
3½	979	1,414	2,000	2,447	2,825	4,000
4	1,386	2,000	2,830	3,464	4,000	5,660
5	2,560	3,642	5,225	6,402	7,390	10,460
6	4,210	6,030	8,590	10,420	12,140	17,180
8	8,750	12,640	17,860	21,865	25,250	35,100
10	16,250	23,450	33,200	40,625	46,900	66,350
12	25,640	36,930	52,320	64,050	74,000	104,500

one or more pressure reducing valves, or pressure regulators, as they are sometimes called, are required.

These are used in two classes of service, one where the steam must be shut off tight to prevent the low pressure building up at time of no load, and the other where the low pressure lines will condense enough steam to offset normal leaking through the valve. In the latter case, double seated valves may be used in a manner that reduces the work required of the diaphragm in closing the valve and consequently the size of the diaphragm. These valves also control the low pressures more closely under conditions of varying high pressures.

Valves that shut off all steam are called *dead end* type. They are single seated, and some of them have pilot operation that provides close control of the reduced pressure. If a thermostatically controlled valve is installed

Table 9. Steam Pipe Capacities for 150 psi Steam Systems

Capacity Expressed in Pounds per Hour

(Steam and Condensate Flowing in Same Direction)

Pipe Size Inches	Drop in Pressure—Pounds per 100 Ft in Length					
	⅛	¼	½	¾	1	2
¾	29	41	58	82	116	184
1	58	82	117	165	233	369
1¼	130	185	262	370	523	827
1½	203	287	407	575	813	1,290
2	412	583	825	1,167	1,650	2,600
2½	683	959	1,359	1,920	2,710	4,290
3	1,237	1,750	2,476	3,500	4,940	7,820
3½	1,855	2,626	3,715	5,250	7,420	11,740
4	2,625	3,718	5,260	7,430	10,500	16,620
5	4,858	6,875	9,725	13,750	19,450	30,750
6	7,960	11,275	15,950	22,550	31,850	50,400
8	16,590	23,475	33,200	46,950	66,400	105,000
10	30,820	43,430	61,700	87,250	123,400	195,000
12	48,600	68,750	97,250	137,600	194,400	307,400

TABLE 10. RETURN PIPE CAPACITIES FOR 30 PSI STEAM SYSTEMS

Capacity Expressed in Pounds per Hour

Pipe Size Inches	Drop in Pressure—Pounds per 100 Ft in Length				
	1/8	1/4	1/2	3/4	1
3/4	115	170	245	308	365
1	230	340	490	615	730
1 1/4	485	710	1,025	1,285	1,530
1 1/2	790	1,155	1,670	2,100	2,500
2	1,575	2,355	3,400	4,300	5,050
2 1/2	2,650	3,900	5,600	7,100	8,400
3	4,850	7,100	10,250	12,850	15,300
3 1/2	7,200	10,550	15,250	19,150	22,750
4	10,200	15,000	21,600	27,000	32,250
5	19,000	27,750	40,250	55,500	60,000
6	31,000	45,500	65,500	83,000	98,000

after, and near, a reducing valve in such a manner as to cut off the passage of steam, the dead end type should be used.

It is common practice when the initial steam pressure is 100 lb or higher to install two-stage reduction. If the radiation served is cast-iron, the *A.S.M.E.* code requires two reducing valves when the inlet pressure exceeds 50 psi. This makes a quieter condition of steam flow, as it is apparent that with one reduction, as for example, from 150 to 2 lb, there is a smaller opening with greater velocity across the reducing valve, and consequently, more noise. A two-stage reduction also introduces a source of safety, since if one reducing valve were to build up its discharge pressure, this excess pressure would not be so great as the case might be in a one-stage reduction.

If an installation requires single seated valves, and the pilot type cannot be used, it is necessary to use two-stage reduction, as single seated valves require sufficient diaphragm area to overcome the unbalanced pressure underneath the single valve. In many cases the large diameter of diaphragm required would make it impractical in construction. With a two-stage reduction the diaphragm diameter required would be reduced. If a one-stage reduction is desired, it is necessary to use a pilot controlled pressure reducing valve, where low pressures are to be maintained closely.

TABLE 11. RETURN PIPE CAPACITIES FOR 150 PSI STEAM SYSTEMS

Capacity Expressed in Pounds per Hour

Pipe Size Inches	Drop in Pressure—Pounds per 100 Ft in Length				
	1/8	1/4	1/2	3/4	1
3/4	156	232	360	560	890
1	313	462	690	1,120	1,780
1 1/4	650	960	1,500	2,330	3,700
1 1/2	1,070	1,580	2,460	3,800	6,100
2	2,160	3,300	4,950	7,700	12,300
2 1/2	3,600	5,350	8,200	12,800	20,400
3	6,500	9,600	15,000	23,300	37,200
3 1/2	9,600	14,400	22,300	34,500	55,000
4	13,700	20,500	31,600	49,200	78,500
5	25,600	38,100	58,500	91,500	146,000
6	42,000	62,500	96,000	150,000	238,000

In making a two-stage reduction, allowance should be made by increasing the pipe size for expansion of steam on the low pressure side of the valve. This also allows steam flow to be at a more nearly uniform velocity. Separating the valves by a distance up to 20 ft is recommended to reduce excessive hunting action of the first valve.

When the reduced pressure is approximately 15 lb or lower, the weight and lever diaphragm valve gives the best results with minimum maintenance. Above 15 lb, spring loaded diaphragm valves should be used, because of the extra weights required on weight and lever type. Equalizing line connections should be made not too close to the valve, and into the bottom of the reduced pressure steam main, to allow maximum condensation to exist in this equalizing line, or the connection is made into the top of the main and a water accumulator used to reduce the variation of the head of water on the diaphragm.

Care should be exercised in selecting the size of a reducing valve. The safest method is to consult the manufacturer. It is essential that sizes of piping to and from the reducing valve be such that they will pass the desired amount of steam with the maximum velocity desired. A common error is to make the size of the reducing valve the same size as that of the service, or outlet pipe size. Generally, this will make the reducing valve oversized, and bring about wire-drawing of valve and seat, due to small lift of the valve seat.

On installations where the steam requirements are relatively large and variable in mild weather or reduced demand periods, wire-drawing may occur. To overcome this condition, two reducing valves are installed in parallel, with the sizes selected on a 70 and 30 per cent proportion of maximum flow. For example, if 50,000 lb of steam per hour are required, the size of one valve is on the basis of $0.7 \times 50{,}000$ lb, or 35,000 lb, and the other on the basis of $0.3 \times 50{,}000$ lb, or 15,000 lb. During the mild or reduced demand periods, steam will flow through the smaller valve only. During the remainder of the season, the larger valve is set to control at whatever low pressure is desired, and the smaller one at a somewhat lower pressure. Thus, when steam flow is not at its maximum, the smaller valve is shut, and automatically opens when the maximum steam demand occurs, since this maximum demand of steam creates a slight pressure drop in the service line.

The installation of reducing valves in pipe lines requires detailed planning. They should be installed to give ease of access for inspection and repair, and wherever possible with diaphragm downward, except in cases of pilot operated valves.

There should be a by-pass around each reducing valve of size equal to one-half the size of reducing valve. The globe valve in by-pass line should be of a better type of construction, and must shut off absolutely tight. A steam pressure gage, graduated up to the initial pressure should be installed on the low pressure side. Safety valves located on the low pressure side should be set 5 lb higher than the final pressure but may be 10 lb higher than the reduced pressure if this reduced pressure is that of the first stage reduction of a double reduction. Strainers are sometimes installed on the inlet to the reducing valve but are not required before a second-stage reduction. If a two-stage reduction is made, it is well to install a pressure gage immediately before the reducing valve of the second-stage reduction also. In sizes 3 in. and above, it is advisable to install a drip trap between the two reducing valves.

BOILER CONNECTIONS

Steam

Cast-iron, sectional heating boilers usually have several outlets in the top. Two or more outlets should be used whenever possible to reduce the velocity of the steam in the vertical uptakes from the boiler and thus to prevent water being carried over into the steam main.

Return

Cast-iron boilers are generally provided with return tappings on both sides, while steel boilers are generally equipped with only one return tapping. Where two tappings are provided, both should be used to effect proper circulation through the boiler. The return connection should include either a Hartford return connection or a check valve to prevent the accidental loss of boiler water to the returns with consequent danger

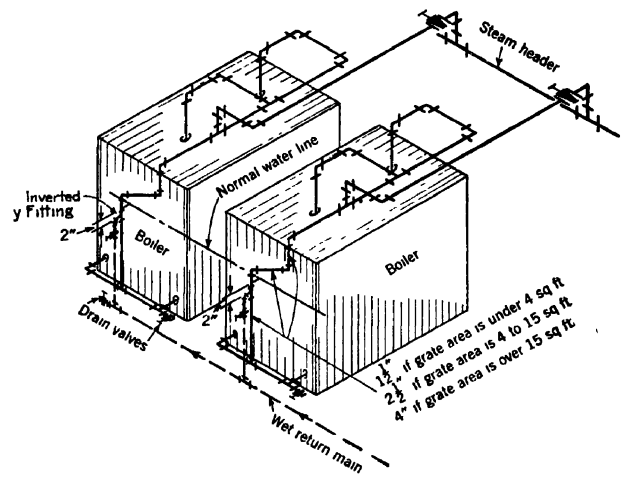


FIG. 16. THE HARTFORD RETURN CONNECTION

of boiler damage. The Hartford return connection is to be preferred over the check valve because the latter is apt to stick or not close tightly and, furthermore, because the check valve offers additional resistance to the condensate coming back to the boiler, which in gravity systems would raise the water line in the far end of the wet return several inches.

In order to prevent the boiler from losing its water under any circumstances, the use of the Hartford return connection is recommended. This connection for a one- or two-boiler installation is shown in Fig. 16. The essential features of construction of a Hartford return connection are: (1) a direct connection (made without valves) between the steam side of the boiler and the return side of the boiler, and (2) a close nipple, or preferably an inverted Y-fitting connection about 2 in. below the normal boiler water line from the return main to the boiler steam and return pressure balance connection. Equalizing pipe connections between the steam and return are given in Fig. 16, based on grate areas, but in no case shall this pipe size be less than the main return piping from the system.

Sizing Boiler Connections

Little information is available on the sizing of boiler runouts and steam headers. Although some engineers prefer an enlarged steam header to

serve as additional steam storage space, there ordinarily is no sudden demand for steam in a steam heating system except during the heating-up period, at which time a large steam header is a disadvantage rather than an advantage. The boiler header may be sized by first computing the maximum load that must be carried by any portion of the header under any conceivable method of operation, and then applying the same schedule of pipe sizing to the header as is used on the steam mains for the building. The horizontal runouts from the boiler, or boilers, may be sized by calculating the heaviest load that will be placed on the boiler at any time, and sizing the runout on the same basis as the building mains. The difference in size between the vertical uptakes from the boiler, which should be of same size as the boiler outlet tapping, and the horizontal main or runout is compensated for by the use of reducing ells.

Return connections to boilers in gravity systems are made the same size as the return main itself. Where the return is split and connected to two tappings on the same boiler, both connections are made the full size of the return line. Where two or more boilers are in use, the return to each may be sized to carry the full amount of return for the maximum load which that boiler will be required to carry. Where two boilers are used, one of them being a spare, the full size of the return main would be carried to each boiler, but if three boilers are installed, with one spare, the return line to each boiler would require only half of the capacity of the entire system or, if the boiler capacity were more than one-half the entire system load, the return would be sized on the basis of the maximum boiler capacity. As the return piping around the boiler is usually small and short, it should not be sized to the minimum.

With returns pumped from a vacuum or receiver return pump, the size of the line may be calculated from the water rate on the pump discharge when it is operating, and the line sized for a very small pressure drop. The relative boiler loads should be considered, as in the case of gravity return connections. Boiler header and piping sizes should be based on the total load.

CONDENSATION RETURN PUMPS

Condensation return pumps are used for gravity systems when the local conditions do not permit the condensation to return to the boiler under the existing static head. The return of the condensate permits the water to repeatedly go through the cycle of vaporization, with subsequent condensation and return to the boiler. During such repeated cycles any incrustants or other substances in solution are precipitated and the water de-activated to a considerable extent so that corrosion of a serious nature is seldom ever encountered where the condensate is repeatedly used. Serious corrosion is more frequently found in systems where the condensation is not repeatedly used but is wasted and fresh make-up water is continually being introduced.

The most generally accepted condensation pump unit for low pressure heating systems consists of a motor-driven centrifugal pump with receiver and automatic float control. Other types in use include rotary, screw and reciprocating pumps with steam turbine or motor drive, and direct-acting steam reciprocating pumps.

The receiver capacities of these automatic units should be sized so as not to cause too great a fluctuation of the boiler water line if fed directly to the boiler and at the same time not so small as to cause too frequent operation of the unit. The usual unit provides storage capacity between

stops in the receiver of approximately 1.5 times the amount of condensate returned per minute and the pump generally has a delivery rate of 3 to 4 times the normal flow. This relation of receiver and pump size to heating system condensing capacity takes account of the peak condensation rate.

A typical installation of a motor driven automatic condensation unit is illustrated in Fig. 17.

VACUUM HEATING PUMPS

On vacuum systems, where the returns are under a vacuum, and sub-atmospheric systems, where the supply piping, radiation and the returns are under a vacuum, it is necessary to use a vacuum pump to discharge the

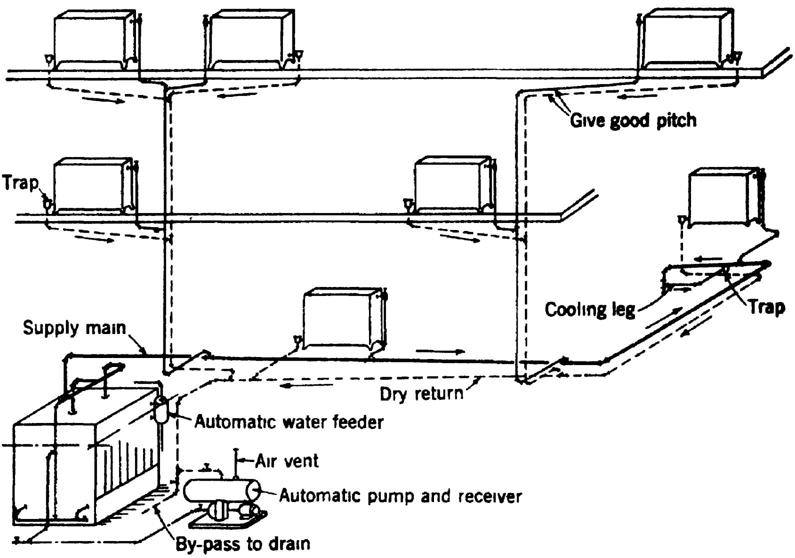


FIG. 17. TYPICAL INSTALLATION USING CONDENSATION PUMP

air and non-condensable gases to atmosphere and to dispose of the condensation. Direct-acting steam-driven reciprocating vacuum pumps are sometimes used where high pressure steam is available or where the exhaust steam from the pump can be utilized. In general, however, these have been replaced by the automatic motor-driven return line heating pump especially developed for this service. Steam turbine drive is also frequently used where steam at suitable pressures is available, the steam being used afterward for building heating. The usual vacuum pump unit consists of a compact assembly of exhausting unit for withdrawing the air-vapor mixture and discharging the air to atmosphere and a water removal unit which discharges the condensate to the boiler. They are furnished complete with receiver, separating tank and automatic controls mounted as an integrated unit on one base. There are also special steam turbine driven units which are operated by passing the steam to be used in heating the building through the turbine with only a 2 to 3 lb drop across the turbine required for its operation. Under special conditions such as installations where it is necessary to return the condensate to a high pressure boiler, auxiliary water pumps may be supplied. In some instances separate air and water pumps may be used.

For rating purposes[4] vacuum pumps are classified as ***low vacuum*** and ***high vacuum***. Low vacuum pumps are those rated for maintaining 5½ in. Hg vacuum on the system, and high vacuum pumps are those rated to maintain vacuums above 5½ in.

Manufacturers of vacuum pumps specify that the standard capacity of pumps shall be 0.3 to 0.5 cfm of air removal and 0.5 gpm of water per 1000 *EDR* served. This capacity is at 5½ in. of vacuum and with condensate at 160 F. The larger air capacity is for smaller systems and the smaller capacity for the larger systems.

Some manufacturers, however, specify more air capacity than standard where higher vacuums are desired and where air leakage is suspected.

The vacuum that can be maintained on a system depends upon the relationship of the air leakage rate into the system to the operating air capacity of the hydraulic evacuator when operating at any given return

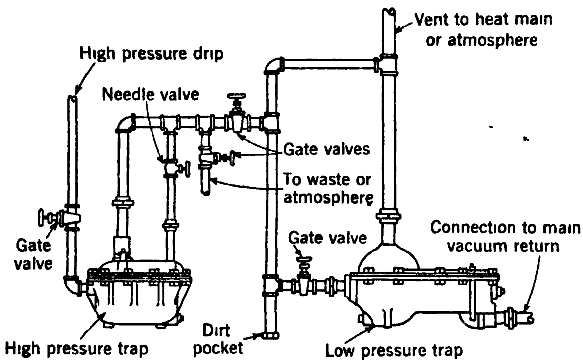


FIG. 18. METHOD OF DISCHARGING HIGH-PRESSURE APPARATUS INTO LOW-PRESSURE HEATING MAINS AND VACUUM RETURN MAINS THROUGH A LOW PRESSURE TRAP

line temperature. The hotter the returns, the lower will be the possible vacuum for a given air leakage rate into the system. It is particularly essential on high vacuum installations to see that the entire system is tight in order to reduce the amount of inward air leakage and, furthermore, to see that relatively higher temperature steam is prevented from entering the vacuum return lines through leaky traps, high pressure drips, etc. It is for this reason that the condensate from equipment using steam at high pressures should not be connected directly to a vacuum return line but should drain to a receiver through a high pressure trap. The receiver should have an equalizing connection to a low pressure steam main and drain through a low pressure trap to the vacuum return main as indicated in Fig. 18.

Vacuum Pump Controls

In the ordinary vacuum system, the vacuum pump is controlled by a vacuum regulator which cuts in when the vacuum drops to the lowest point desired and cuts out when it has been increased to the highest point, these points being varied to suit the particular system or operating conditions. In addition to this vacuum control, a float control is included

[4]A.S H V.E Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps, (A S H.V.E TRANSACTIONS, Vol 40, 1934, p. 33).

which will automatically start the pump whenever sufficient condensation accumulates in the receiver, regardless of the vacuum on the system. A selector switch is usually provided to allow operation at night as a condensation pump only, also to give manual or continuous operation when desired.

There are several variations in the control of the vacuum maintained on the system by the pump. In some sub-atmospheric systems where orifices are used, the vacuum pump control maintains a pressure difference between the supply and the return piping, which is held within relatively close limits. There are other sub-atmospheric systems which utilize special temperature-pressure actuated controls for maintaining the desired conditions in the return lines. Where various zones are connected to the same return main, the return vacuum must be controlled to meet the requirements of the zone operating at the lowest steam supply pressure.

Piston Displacement Vacuum Pumps

Piston displacement return vacuum heating pumps may be either electric or steam driven. Their piston speed in feet per minute should not exceed 20 times the square root of the number of inches in their stroke. They are usually supplied with an air separating tank, open to atmosphere, placed on the discharge side of the pump and at an elevation sufficiently high to allow gravity flow of the condensate to the boiler. If the boiler pressure is too high for such gravity feed then an additional steam pump for feeding the boiler is desirable. The extra pump is sometimes avoided by using a closed separating tank with a float controlled vent. In both arrangements, the air taken from the system must be discharged against the full discharge pressure of the vacuum pump. In the case of high or medium pressure boilers, it is better to use the atmospheric separator and the second pump.

In figuring the required displacement for such pumps, a value of from 6 to 10 times the volumetric flow of condensation is used for average vacuums and systems.

TRAPS

Traps are generally classified as to function as (*a*) separating traps, (*b*) return, lifting or vacuum traps, and (*c*) air traps. Separating traps may be either float operated, thermostatically operated, or float and thermostatically operated. Return traps for low pressure service are referred to later as *alternating receivers* in this chapter. Return traps may also operate to receive condensate under a vacuum and return it to atmosphere or a higher pressure. Air traps are generally float operated.

Separating traps are used to release water of condensation but to retain steam. The thermostatic, and float and thermostatic types release both condensate and air but retain steam. Separating traps are used for draining condensate from radiators, indirect air heaters, steam piping systems, kitchen equipment, laundry equipment, hospital equipment, drying equipment and many other kinds of apparatus. Air traps release air but retain water. Devices known as air vents are, in principle, traps which allow the passage of air but prevent the passage of either water or steam.

Return traps are used for returning condensate either by gravity, by steam pressure, or by both, to a boiler or other point of disposal, and for lifting condensate from a lower to a higher elevation, or for handling condensate from a lower to a higher pressure.

The fundamental principle upon which the operation of practically all traps depends is that the pressure within the trap at the time of discharge shall be equal to, or slightly in excess of, the pressure against which the trap must discharge, including the friction head, velocity head and static head on the discharge side of the trap.

Traps may also be classified according to the principle of operating device which supplies the power to cause them to function as (1) float, (2) bucket, (3) thermostatic, (4) float and thermostatic, (5) impulse, or (6) tilting traps.

Float Traps. A discharge valve is operated by the rise and fall of a float due to the change of water level in the trap. When the trap is empty the float is in its lowest position, and the discharge valve is closed. A gage glass indicates the height of water in the chamber.

Unless float traps are well made and proportioned there is danger of considerable steam leakage through the discharge valve due to unequal expansion of the valve and seat and the sticking of moving parts. The discharge from a float trap is usually continuous since the height of the float, and consequently the area of the outlet, is proportional to the amount of water present.

Bucket Traps. Bucket traps are of two types, the upright and inverted, and although they are both of the open float construction, their operating principle is entirely different. In the *upright bucket* trap, the water of condensation enters the trap and fills the space between the bucket and the walls of the trap. This causes the bucket to float and forces the valve against its seat, the valve and its stem usually being fastened to the bucket. When the water rises above the edges of the bucket it flows into it and causes it to sink, thereby withdrawing the valve from its seat. This permits the steam pressure acting on the surface of the water in the bucket to force the water to a discharge opening. When the bucket is emptied it rises and closes the valve and another cycle begins. The discharge from this type of trap is intermittent.

In the *inverted bucket* trap, steam floats the inverted submerged bucket and closes the valve. Water entering the trap fills the bucket, which sinks and through compound leverage opens the valve, and the trap discharges. It is impossible to install a water gage glass on an inverted bucket trap, but if visual inspection is necessary, a gage glass can be placed on the line leading to the trap. No air relief cocks can be used, but they are unnecessary, as the elimination of air is automatically taken care of by air passing through the vent in the top of the inverted bucket regardless of temperature.

Thermostatic Traps. Thermostatic traps are of two types, those in which the discharge valve is operated by the relative expansion of metals, and those in which the action of a volatile liquid is utilized for this purpose. Thermostatic traps of large capacity for draining blast coils or very large radiators are called *blast* traps.

Float and thermostatic traps have both a thermostatic element to release air and a float element to release the water.

Impulse traps operate with a moving valve actuated by a control cylinder. When the trap is handling condensate, the pressure required to lift the valve is greater than the reduced pressure in the control cylinder and consequently the valve opens allowing a free discharge of condensate. As the remaining condensate approaches steam temperature, flashing

results, flow through the valve orifice is choked and the pressure builds up in the control chamber closing the valve.

Automatic Return Traps

In the general heating plant, where thermostatic traps are installed on the heating units, it becomes necessary to provide a means for returning the water of condensation to the boiler, if a condensation or vacuum pump is not used. When the return main can be kept sufficiently high above the boiler water line for all operating conditions, the water of condensation will flow back by gravity, and no mechanical device is required. But actually this does not work out in practice. It follows, therefore, that a direct-return trap is needed for the handling of the condensation even though it may not be called into action except under some operating condition where the pressure differential exceeds the static head provided. The installation of a direct-return trap assures safety for such systems, and guarantees the operation of the plant under varying conditions.

Automatic return traps, sometimes called alternating receivers, may be of the counter-balanced, tilting type, or spring actuated. These consist of a small receiver with an internal float, and when the condensate will not flow into the boiler under pressure, it will feed into the receiver of the trap, and in so doing, raise or tilt the float or mechanism which actuates a steam valve automatically. This admits steam to the receiver, at boiler pressure, and the equalizing of the pressures which follows allows the water to flow into the boiler.

Tilting Traps. With this type of trap, water enters a bowl and rises until its weight overbalances that of a counter-weight, and the bowl sinks to the bottom. As the bowl sinks, a valve is opened thus admitting live steam pressure on the surface of the water and the trap then discharges. After the water is discharged, the counter-weight sinks and raises the bowl, which in turn closes the valve and the cycle begins again. Tilting traps are necessarily intermittent in operation. They are not ordinarily equipped with glass water gages, as the action of the trap shows when it is filling or emptying. The air relief of tilting traps is taken care of by the valves of the trap.

DRIPS

A steam main in any type of steam heating system may be dropped to a lower level without dripping if the pitch is downward with the direction of steam flow. Any steam main in any heating system can be elevated if dripped. Fig. 19 shows a connection where the steam main is raised and the drain is to a wet return. If the elevation of the low point is above a dry return it may be drained through a trap to the dry return in two-pipe vapor, vacuum and sub-atmospheric systems. Horizontal steam pipes may also be run over obstructions without a change in level if a small pipe is carried below the obstruction to care for the condensation (Fig. 20). Horizontal return pipes may be carried past doorways and other obstructions by using the scheme illustrated in Fig. 21. It will be noted that the large pipe, in this case, runs below the obstruction and the smaller one over it.

Branches from steam mains in one-pipe gravity steam systems should use the *preferred connection* shown in Fig. 22, but where radiator condensation does not flow back into the main the *acceptable* method shown in the same figure may be used. This acceptable method has the advantage of

giving a perfect swing joint when connected to the vertical riser or radiator connection, whereas the preferred connection does not give this swing without distorting the angle of the pipe. Runouts from the steam main are usually made about 5 ft long to provide flexibility for movement in the main.

Offsets in steam and return piping should preferably be made with 90-deg ells but occasionally fittings of other angles are used, and in such

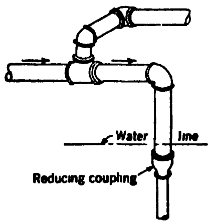


FIG. 19. DRIPPING MAIN WHERE IT RISES TO HIGHER LEVEL

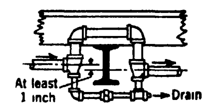


FIG. 20. LOOPING MAIN AROUND BEAM

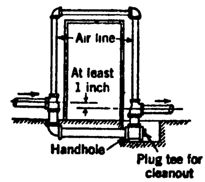


FIG. 21. LOOPING DRY RETURN MAIN AROUND OPENING

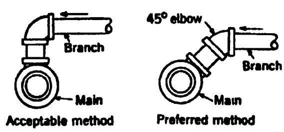
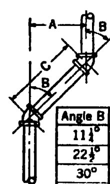


FIG. 22. METHODS OF TAKING BRANCH FROM MAIN



Angle B	Constant
11¼°	5 126
22½°	2 613
30°	2 000
45°	1 414
60°	1 155

To find length C-multiply A by constant for angle B

FIG. 23. CONSTANTS FOR DETERMINING LENGTH OFFSET PIPE

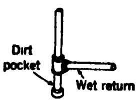


FIG. 24. DIRT POCKET CONNECTION

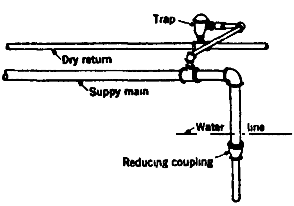


FIG. 25. DRIPPING END OF MAIN INTO WET RETURN

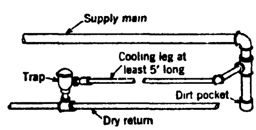


FIG. 26. DRIPPING END OF MAIN INTO DRY RETURN

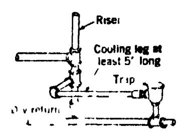


FIG. 27 DRIPPING HEEL OF RISER INTO DRY RETURN

cases the length of the diagonal offset will be found as shown in Fig. 23.

Dirt pockets, desirable on all systems employing thermostatic traps, should be so located as to protect the traps from scale and muck which will interfere with their operation. Dirt pockets are usually made 8 in. to 12 in. deep and serve as receivers for foreign matter which otherwise would be carried into the trap. They are constructed as shown in Fig. 24.

On vapor systems where the end of the steam main is dripped down into the wet return, the air venting at the end of the main is accomplished by an air vent passing through a thermostatic trap into the dry return

line as shown in Fig. 25. On vacuum systems the ends of the steam mains are dripped and vented into the return through drip traps opening into the return line. The same method may be used in atmospheric systems. A float type trap is preferable to a thermostatic trap for dripping steam mains and large risers. If thermostatic traps are used, a cooling leg (Fig. 26) should always be provided. The cooling leg is for cooling the condensation sufficiently before it reaches the trap so the trap will not be held shut by too high a temperature. On down-feed systems of atmospheric, vapor, and vacuum types, the bottoms of the steam risers are dripped in the manner shown in Fig. 27. On large systems it is desirable to install a gate valve in the cooling leg ahead of the trap.

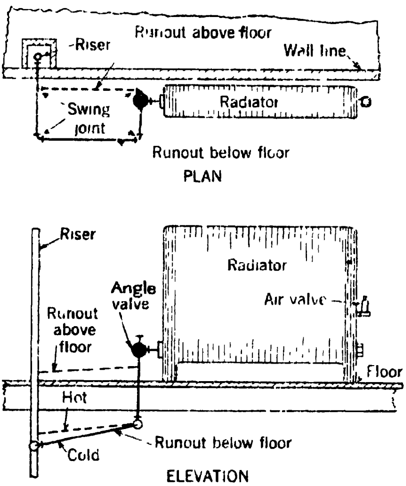


FIG. 28. ONE-PIPE RADIATOR CONNECTIONS

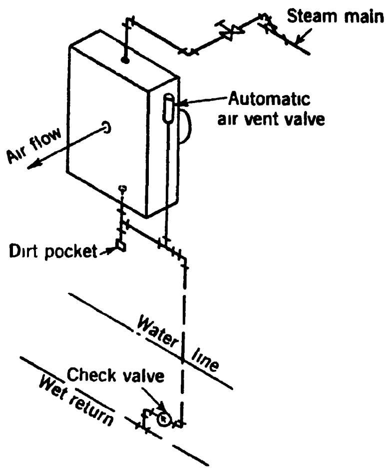


FIG. 29. UNIT HEATER CONNECTED TO ONE-PIPE AIR-VENT SYSTEM

CONTROL VALVES

Gate valves are recommended in all cases where service demands that the valve be either entirely open or entirely closed, but they should never be used for throttling. Angle globe valves and straight globe valves should be used for throttling, as done on by-passes around pressure reducing valves or on by-passes around traps.

CONNECTIONS TO HEATING UNITS

Riser, radiator and convector connections must not only be properly pitched at the time they are installed but must be arranged so that the pitch will be maintained under the strains of expansion and contraction. These connections may be made by swing joints which permit the expansion or contraction to occur under heating and cooling without bending of pipes. To take care of expansion in long risers, either expansion joints of commercial construction or pipe swing joints are used. Anchoring of pipes between expansion joints is desirable.

Two satisfactory methods of making runouts for one-pipe systems for either the up-feed or the down-feed type are shown in Fig. 28. Where the vertical distance is limited and the runouts must run above the floor the radiator may be set on pedestals or of the high leg type. A method

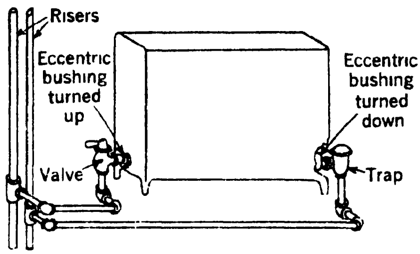


FIG. 30. TYPICAL CONNECTIONS FOR TWO-PIPE SYSTEM

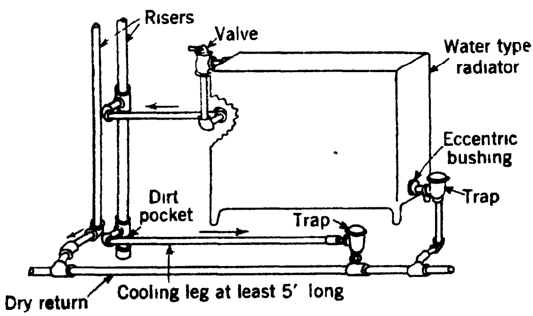


FIG. 31. TOP AND BOTTOM OPPOSITE END RADIATOR CONNECTIONS

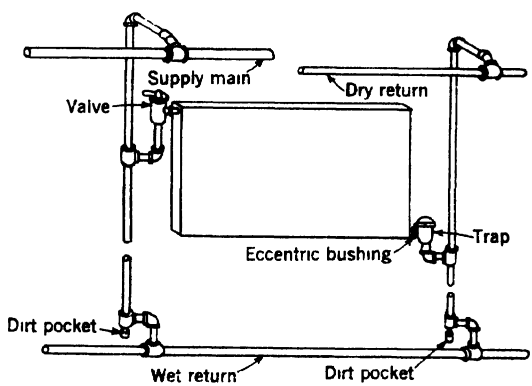


FIG. 32. CONNECTIONS TO RADIATOR HUNG ON WALL

of connecting a unit heater to a one-pipe steam heating system is illustrated in Fig. 29.

Typical two-pipe radiator or convector connections are shown in Figs. 30, 31 and 32. While the top is the preferred location for the control valve, it may be located at the bottom. Short radiators may be top supply and bottom return on same end. With convectors the control

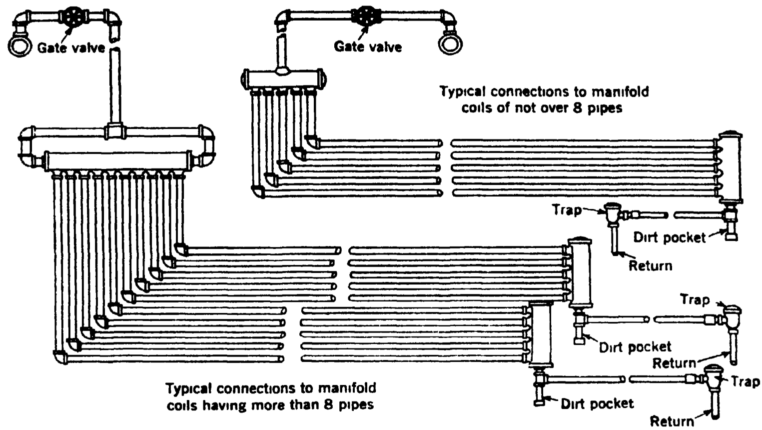


FIG. 33 TYPICAL PIPE COIL CONNECTIONS

valve is sometimes omitted and a damper in outlet grille used for heat control. The typical method of connecting pipe coils is shown in Fig. 33 and is suitable for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

Typical pipe connections for indirect radiators and tempering or heating stacks are shown in Figs. 34, 35, 36, 37 and 38.

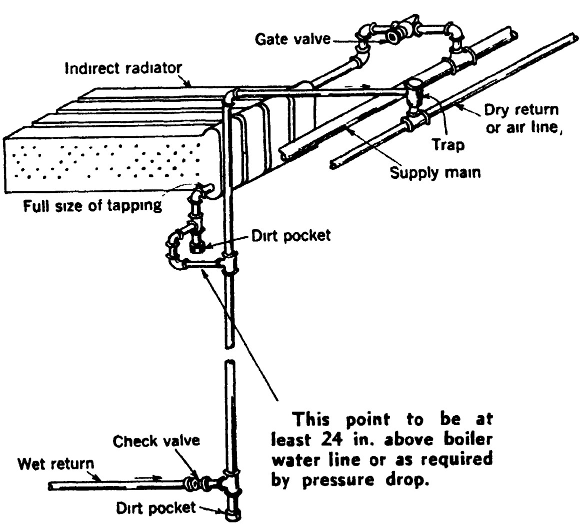


FIG. 34. TYPICAL PIPING CONNECTIONS TO CONCEALED HEATING UNITS WITH WET RETURNS

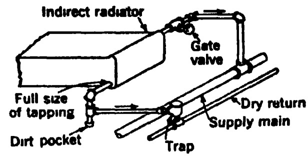


Fig. 35. Piping Connections to Indirect Radiators with Dry Returns

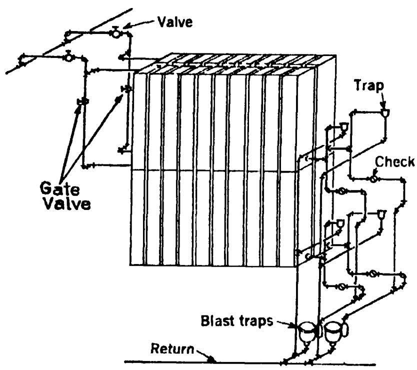


Fig. 36. Supply and Return Connections for Heating Units of Central Fan Systems

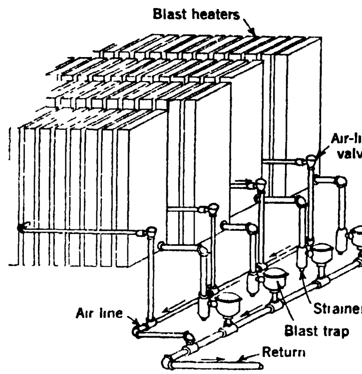


Fig. 37. Typical Connections to Central Fan System Heating Units Exceeding 12 Sections

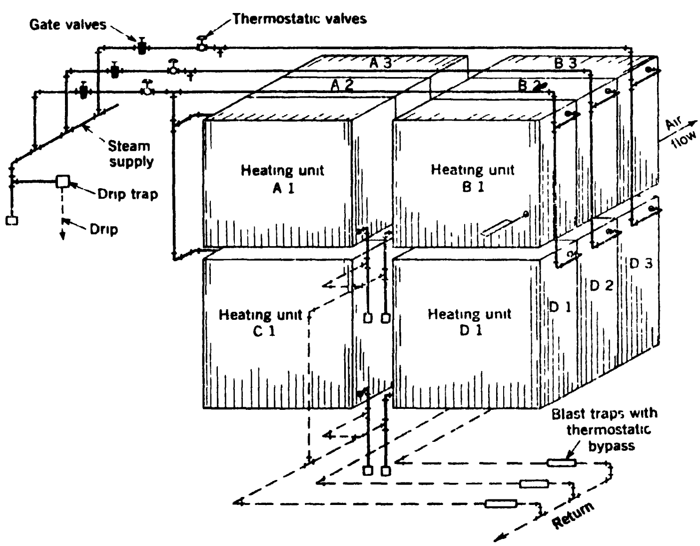


Fig. 38. Typical Piping for Atmospheric and Vacuum Systems with Thermostatic Control (Central Fan System)

Where a building is served by a vacuum system or a sub-atmospheric system the stacks should be piped in the usual manner and traps of large capacity, preferably of the combination float and thermostatic type, should be used. In the orifice and *closed* two-pipe systems, traps should be used on the returns so that a pressure above that of the atmosphere may be secured on the heaters.

Each stack should have a separate steam and return connection and trap. Wide stacks are more evenly heated if divided and supplied with two steam connections, one at each end, and a return connection for each steam connection. For stacks of large capacity it is sometimes desirable to run a separate steam main direct from the boiler to the stacks.

CHAPTER 15

Hot Water Heating Systems and Piping

One- and Two-Pipe Systems, Selecting Pipe Sizes, Forced Circulation, Gravity Circulation, Expansion Tanks, Installation Details, Examples of Piping Design

HOT water heating systems may be divided into two general classes, the gravity systems in which circulation is caused by the difference in density of the water in the supply and return risers, and the forced circulation systems in which circulation is caused by a pump. Water is supplied at temperatures from 150 to 250 F, and the higher temperatures are generally used with the forced system.

Four principal elements of a hot water system may be recognized, and these are discussed in the various chapters as follows:

1. The boiler or heat exchanger in which the water is heated is similar to the steam boiler, and is discussed in Chapter 12.

2. The radiators, convectors, pipe coils or panels that deliver the heat to the spaces to be heated are covered in Chapters 13 and 45.

3. The design of the piping system through which the water flows from the boiler to the radiators and back to the boiler is considered in this chapter.

4. The control system by which the temperature in the heated spaces is regulated according to varying requirements is covered in Chapter 33

SYSTEMS OF PIPING

There are two general systems of piping used for either gravity or forced hot water systems:

(a) Two-pipe system.
(b) One-pipe system.

With either of these piping systems the distributing mains may be located in the basement with up-feed to the radiators and risers, or the supply main may be located in the attic with the return main located in the basement. For radiators located on the basement floor the mains may be run at the ceiling, as one of the advantages of a forced hot water heating system is that the returns need not be below the radiators as required with a steam system. In some one-pipe systems there is one supply main in the basement with separate flow and return riser connections to the radiators.

In the two-pipe system there are separate supply and return pipes throughout so that the radiators are connected in parallel, resulting in the same water temperature in all radiators. With the one-pipe system part of the water flows through more than one radiator, so that the water temperature toward the end of the main is not as high as near the boiler. However, with the one-pipe system, by maintaining a rapid circulation and small difference in temperature between the water leaving and returning to the boiler or other heat generator, the variation in the radiator water temperature is reduced.

The two-pipe system for larger buildings should, if possible, be arranged for reversed return. The direct and reversed return systems are shown in Figs. 1 and 2. With the reversed return system, the length of the water circuit for any one radiator is the same as for any other radiator and, therefore, the friction and temperature losses to all radiators should be nearly the same.

In some cases the reversed return system involves no more piping than the direct return system. In the case of large buildings, it is often advisable to zone the piping.

Mechanical Circulators

Circulating pumps are usually of the centrifugal type. The capacity of the pump is figured from the Mbh (1000 Btu per hour) required for heating and the drop in temperature selected. For example, for 100 Mbh and 20 F drop a pump having a capacity of 5000 lb water per hour or 10 gpm should be used. The resistance head is based on the system as designed. In large systems the economical size of pump may be determined by comparing the cost of power for operation, with the annual charges on the capital cost of the piping system, as larger pipe sizes mean less pump power. Velocities through piping in excess of 4 fps are likely to cause disturbing noises in buildings other than factories. In large systems the pumps are run continuously while in small ones they are run either continuously or intermittently depending on the type of automatic temperature control selected. Small circulating pumps are usually driven by direct-connected electric motors. Under certain conditions a valved by-pass should be provided and the piping so designed that in

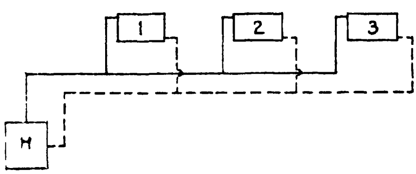


FIG 1. A DIRECT RETURN SYSTEM

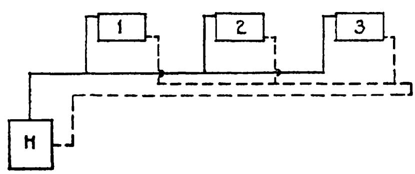


FIG. 2. A REVERSED RETURN SYSTEM

case of breakdown of the pump or failure of electric current there will be sufficient gravity circulation to keep the building reasonably warm. In large buildings or groups of buildings, it is often advisable to have two pumps, each of about 70 per cent of the total capacity, to take care of breakdown service. During mild weather, variations in water temperature may be utilized to balance the required heat loss. In the larger systems steam turbines are sometimes used to drive the pumps, the exhaust steam being used for heating the water, and in such buildings as hospitals this may be the most economical method.

As the average pump used for water circulation is not over 60 per cent efficient, the cost of power on a large installation should be computed and comparisons made between the savings in capital cost of piping and the annual cost of power.

FORCED CIRCULATION PIPE SIZES

The pressure heads available in forced circulation systems are much greater than those in gravity circulation systems, consequently, higher velocities may be used in designing the system, with the result that smaller pipes may be selected and the first cost of the installation reduced. As the pipe sizes of a heating system are reduced, the necessary increase in the velocity of the water increases the friction losses and thus the cost of operation and the initial cost of the circulating equipment. The increased velocity of a forced circulation system offers a number of advantages, such as a much shorter heating-up period and a more flexible

Table 1. Heat-carrying Capacity of Standard Black Pipes with Temperature Drop of 20 F[a]

Nominal Pipe Sizes 3/8 in. to 12 in., and Friction Heads 4 to 800 milinches per foot (A = Capacity, Mbh. B = Velocity, inches per second) (One milinch equals 0.001 in.)

Milinch Friction Loss per Foot of Pipe		Nominal Pipe Size, Inches															
		3/8	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	5	6	8	10	12
4	*A*	0.75	1.35	2.85	5.4	11.3	17.0	33.0	53.1	95	141	197	363	596	1250	2320	3730
	B	1.5	1.7	2.1	2.4	2.9	3.2	3.8	4.3	5.0	5.5	6.0	7.0	7.9	9.6	11	12
6	*A*	0.9	1.7	3.6	6.75	14.0	21.2	41.3	66.4	119	176	248	456	748	1570	2920	4690
	B	1.8	2.1	2.6	3.0	3.6	4.0	4.7	5.3	6.2	6.9	7.5	8.8	10	12	14	16
8	*A*	1.05	2.0	4.2	7.9	16.4	24.8	48.4	77.9	140	207	291	535	879	1850	3440	5520
	B	2.1	2.5	3.0	3.5	4.2	4.7	5.6	6.3	7.3	8.0	8.8	10	12	14	17	19
10	*A*	1.2	2.2	4.7	8.9	18.6	28.0	54.7	88.1	158	234	329	605	997	2100	3910	6270
	B	2.4	2.8	3.4	4.0	4.8	5.3	6.3	7.1	8.2	9.1	9.9	12	13	16	19	27
12	*A*	1.35	2.45	5.2	9.8	20.5	31.0	60.4	97.4	175	259	364	671	1100	2320	4330	6950
	B	2.7	3.1	3.7	4.4	5.3	5.9	6.9	7.8	9.1	10	11	13	15	18	21	24
14	*A*	1.45	2.65	5.65	10.7	22.3	33.7	65.8	106	190	282	397	731	1200	2530	4730	7590
	B	2.9	3.4	4.1	4.8	5.7	6.4	7.6	8.5	9.9	11	12	14	16	20	23	26
16	*A*	1.55	2.85	6.05	11.5	24.0	36.3	70.8	114	205	303	428	787	1300	2730	5100	8190
	B	3.1	3.6	4.4	5.1	6.2	6.9	8.1	9.7	11	12	13	15	17	21	25	28
20	*A*	1.75	3.25	6.85	13.0	27.1	41.0	80.0	129	232	344	484	892	1470	3100	5790	9300
	B	3.5	4.1	4.9	5.8	7.0	7.7	9.2	10	12	13	15	17	20	24	28	32
25	*A*	2.0	3.65	7.75	14.7	30.6	46.3	90.5	146	263	389	548	1010	1670	3510	6570	10560
	B	4.0	4.6	5.6	6.5	7.9	8.8	10	12	14	15	17	19	22	27	32	36
30	*A*	2.2	4.0	8.55	16.2	33.8	51.2	100	162	290	430	607	1120	1850	3900	7280	11710
	B	4.4	5.1	6.1	7.2	8.7	9.7	11	13	15	17	18	22	25	30	35	40
35	*A*	2.35	4.4	9.3	17.6	36.8	55.7	109	176	316	469	661	1220	2010	4250	7940	12780
	B	4.7	5.5	6.7	7.9	9.5	11	13	14	16	18	20	23	27	33	39	44
40	*A*	2.55	4.7	10.0	18.9	39.6	59.9	117	189	341	505	712	1320	2170	4580	8570	13780
	B	5.1	5.9	7.2	8.4	10	11	13	15	18	20	22	25	29	35	42	47
50	*A*	2.85	5.3	11.3	21.4	44.7	67.7	133	214	386	572	807	1490	2460	5190	9720	15650
	B	5.7	6.7	8.1	9.5	12	13	15	17	20	22	24	29	33	40	47	54
60	*A*	3.15	5.85	12.4	23.6	49.4	74.9	147	238	427	633	893	1650	2730	5760	10780	17360
	B	6.3	7.4	8.9	11	13	14	17	19	22	25	27	32	36	44	52	60
70	*A*	3.45	6.35	13.5	25.7	53.8	81.4	160	258	465	690	973	1800	2970	6280	11760	18950
	B	6.9	8.0	9.7	11	14	15	18	21	24	27	29	35	40	48	57	65
80	*A*	3.7	6.8	14.5	27.6	57.9	87.6	172	278	500	743	1050	1940	3200	6770	12690	20440
	B	7.4	8.6	10	12	15	17	20	22	26	29	32	37	43	52	62	70
100	*A*	4.15	7.7	16.4	31.1	65.4	99.0	194	314	566	840	1190	2200	3630	7680	14400	23200
	B	8.3	9.7	12	14	17	19	22	25	30	33	36	42	48	59	70	80
150	*A*	5.2	9.6	20.4	38.8	81.6	124	243	393	709	1050	1490	2760	4560	9650	18120	29220
	B	10	12	15	17	21	23	28	32	37	41	45	53	61	74	88	101
200	*A*	6.05	11.2	23.9	45.4	95.5	145	285	461	832	1240	1750	3240	5360	11350	21320	34400
	B	12	14	17	20	25	27	33	37	43	48	53	62	71	87	104	118
300	*A*	7.5	13.9	29.7	56.6	119	181	356	577	1040	1550	2190	4060	6730	14270	26830	43300
	B	15	18	21	25	31	34	41	46	54	60	66	78	90	110	131	149
400	*A*	8.75	16.2	34.7	66.2	140	212	417	676	1220	1820	2570	4780	7910	16790	31580	51000
	B	18	21	26	30	36	40	48	54	64	71	78	92	105	129	154	175
500	*A*	9.85	18.3	39.2	74.8	158	239	471	765	1380	2060	2910	5410	8970	19040	35840	57880
	B	20	23	29	33	41	45	54	62	72	80	88	104	119	147	174	199
600	*A*	10.9	20.2	43.2	82.5	174	264	521	846	1530	2280	3220	5990	9930	21100	39740	64210
	B	22	26	32	37	45	50	60	68	80	89	97	115	132	162	193	221
800	*A*	12.7	23.6	50.5	96.5	204	310	610	992	1790	2670	3780	7030	11670	24820	46780	75620
	B	25	30	37	43	52	59	70	80	94	104	114	135	155	191	228	260

[a]For other temperature drops the pipe capacities may be changed correspondingly. For example, with a temperature drop of 30 F the capacities shown in this table are to be multiplied by 1.5.

TABLE 2. HEAT-CARRYING CAPACITY OF TYPE L COPPER TUBING WITH TEMPERATURE DROP OF 20 F[a]

Nominal Tube Sizes 3/8 in. to 4 in., and Friction Heads 60 to 720 milinches per foot. (A = Capacity, Mbh. B = Velocity, inches per second) (One milinch equals 0.001 in.)

Nominal Tube Size, In.		Milinch Friction Loss per Foot of Tube											
		720	600	480	360	300	240	180	150	120	90	75	60
3/8	*A*	10	9	8	6.8	6.2	5.4	4.6	4	3.6	3	2.8	2.4
	B	27	24	21	18	16.5	14	13	11	10	8.5	8	7
1/2	*A*	20	18	16	13.5	12	10.8	9	8	7	6	5.4	4.7
	B	35	30	25	21	19	17	15	13	12	10	9	8
5/8	*A*	36	30	26	22.1	20	17.8	15	13.1	11.8	9.9	9	7.9
	B	37	34	30	24	21	19	17	15	13	11	10	9
3/4	*A*	51	46	40	34	31	28	23.2	20.5	18.1	15.3	13.9	12.1
	B	42	38	33	27	24	21	19	17	14	12	11.5	10
1	*A*	104	94	82	70	63	56	47	42	37	32	28	25
	B	48	45	39	34	30	25	22	19	17	14.5	13	12
1 1/4	*A*	185	169	149	125	112	100	84	75	66	56	50	44
	B	55	51	45	39	35	30	25	22	19	17	15	13
1 1/2	*A*	300	270	235	200	180	160	134	120	105	90	81	71
	B	62	57	51	43	39	35	30	25	22	19	17	15
2	*A*	625	560	495	420	375	335	280	250	200	188	170	150
	B	76	68	59	51	47	42	36	32	27	22	20	18
2 1/2	*A*	1130	1010	890	750	680	600	500	450	395	335	305	270
	B	90	80	69	58	49	47	42	37	33	26	23	21
3	*A*	1840	1650	1450	1210	1100	980	820	740	650	550	490	420
	B	98	90	80	66	59	52	47	42	36	30	27	23
3 1/2	*A*	2750	2480	2170	1840	1650	1450	1210	1100	980	820	740	650
	B	110	100	89	75	66	57	51	45	40	35	30	26
4	*A*	3900	3505	3100	2600	2350	2090	1760	1580	1390	1180	1080	950
	B	120	108	96	83	73	63	55	49	44	37	34	29

[a]For other temperature drops the pipe capacities may be changed correspondingly. For example, with temperature drop of 30 F, the capacities shown in this table are to be multiplied by 1.5.

TABLE 3. IRON AND COPPER ELBOW EQUIVALENTS[a]

Fitting	Iron Pipe	Copper Tubing
Elbow, 90-deg	1.0	1.0
Elbow, 45-deg	0.7	0.7
Elbow, 90-deg long turn	0.5	0.5
Elbow, welded, 90-deg	0.5	0.5
Reducing coupling	0.4	0.4
Open return band	1.0	1.0
Open gate valve	0.5	0.7
Open globe valve	12.0	17.0
Angle radiator valve	2.0	3.0
Radiator or convector	3.0	4.0
Boiler or heater	3.0	4.0
Tee, per cent flowing through branch:		
100	1.8	1.2
50	4.0	4.0
25	16.0	20.0

[a]The friction head in one 90 deg standard elbow is approximately equal to the friction of a length of straight pipe of the same nominal size and 25 diam long. Hence one elbow equivalent equals 25 *D* divided by 12 feet of straight pipe or tubing.

Table 4. Friction Heads (in Milinches) of Central Circular Diaphragm Orifices in Unions

(One milinch equals 0.001 in.)

Diameter of Orifices (Inches)	Velocity of Water in Pipe in Inches per Second									
	2	3	4	6	8	10	12	18	24	36
					3/4-in. Pipe					
0.25	1300	2900	5000	11,300	20,800	32,000	45,000			
0.30	650	1450	2500	5700	10,400	16,000	23,000	57,000		
0.35	330	740	1300	2900	5200	8000	12,000	26,000	47,000	
0.40	170	380	660	1500	2600	4000	6800	13,000	24,000	53,000
0.45		185	330	740	1300	2000	2900	6500	12,000	27,000
0.50			155	350	620	970	1400	3200	5700	13,000
0.55			75	170	300	480	700	1600	2800	6400
					1-in. Pipe					
0.35	900	2000	3500	7800	14,000	22,000	32,000			
0.40	460	1000	1800	4000	7200	12,000	17,000	37,000	65,000	
0.45	270	570	1000	2300	4100	6400	9300	21,000	37,000	
0.50	160	330	580	1400	2300	3700	5400	12,000	22,000	50,000
0.55		190	330	750	1300	2200	3000	7000	13,000	28,000
0.60			200	440	800	1300	1800	4200	7400	17,000
0.65			120	260	460	720	1100	2400	4300	10,000
					1 1/4-in. Pipe					
0.45	1000	2250	4000	8900	16,000	25,000	36,000			
0.50	660	1450	2600	5800	10,400	16,400	23,000	53,000		
0.55	430	950	1700	3800	6800	10,500	15,000	34,000	60,000	
0.60	280	630	1100	2500	4400	6900	10,000	22,000	40,000	
0.65	190	420	750	1700	3000	4700	6700	15,000	27,000	60,000
0.70		285	510	1150	2000	3100	4500	10,000	18,000	40,000
0.75		190	330	750	1300	2100	3000	6700	12,000	26,000
					1 1/2-in. Pipe					
0.55	850	1900	3300	7400	13,000	21,000	30,000			
0.60	600	1300	2300	5400	8600	16,800	21,000	50,000		
0.65	400	850	1500	3600	7200	10,400	14,000	30,000	53,000	
0.70	260	600	1100	2600	4400	7000	10,000	21,000	39,000	
0.75	180	400	760	1800	3000	5000	7000	14,000	28,000	
0.80		300	540	1200	2200	3200	5000	10,200	19,000	45,000
0.85		200	380	860	1600	2300	3000	7800	13,000	30,000
					2-in. Pipe					
0.70	890	1850	3500	7400	14,000	22,300	33,000			
0.80	470	975	1800	3900	7400	11,700	17,000	37,000		
0.90	255	560	1000	2200	4200	6500	9500	20,500	38,000	
1.00	160	340	610	1320	2520	4000	5800	12,500	23,000	49,000
1.10		214	375	850	1600	2500	3700	7900	14,000	30,000
1.20			195	460	950	1360	1910	4200	8100	16,800
1.30				275	525	980	1375	3100	4400	8850

Note—The losses of head for the orifices in the 1½-in. and 2-in. pipe were calculated from those in the smaller pipes, the calculations being based on the assumption that, for any given velocity, the loss of head is a function of the ratio of the diameter of the pipe to that of the orifice. This had been found to be practically true in the tests to determine the losses of head in orifices in ¾-in., 1-in., and 1¼-in. pipe, conducted by the Texas Engineering Experiment Station, and also in the tests to determine the losses of head in orifices in 4-in., 6-in., and 12-in. pipe, conducted by the Engineering Experiment Station of the University of Illinois, (*Bulletin* 109, Table 6, p. 38, Davis and Jordan).

control of hot water circulation. This improved performance merits the small increase in operating cost necessary to circulate the water mechanically. The velocities required should be determined by calculation for the particular system under consideration.

Since forced circulation velocities are higher than those in gravity systems, and since the friction heads in a heating system vary almost as the squares of the velocities, a given error in the calculation or assumption of a velocity is less important in a forced circulation system than in

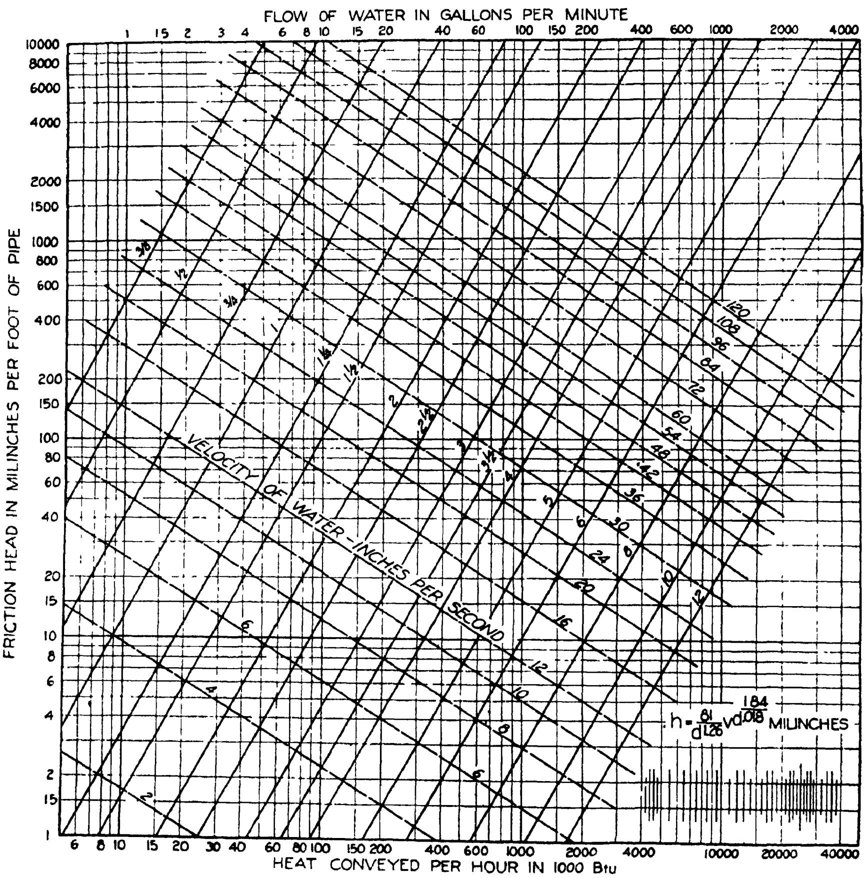


FIG. 3. FRICTION HEADS IN BLACK IRON PIPES FOR A 20 F TEMPERATURE DIFFERENCE OF THE WATER IN THE FLOW AND RETURN LINES

For other temperature drops the pipe capacities may be changed correspondingly. For example, with a temperature drop of 30 F, the capacities shown in this figure are to be multiplied by 1.5.

a gravity circulation system, and, consequently, it is easier to design a satisfactory forced circulation system than a satisfactory gravity circulation system.

In forced hot water systems, it is customary to use a temperature drop of 20 or 30 F between the water entering and leaving the boiler or other heater. The head against which the system is to operate must then be decided. This varies from 2 to 5 ft for small systems and may rise to 100 ft on large jobs with a group of buildings. For iron pipe, the sizes

can be figured using Fig. 3 and Tables 1 and 3. For copper tubing Tables 2 and 3 are to be used. In systems designed with reversed returns, it will generally be found that very little adjustment is necessary to secure even distribution to all radiators. However, orifices may be used to control the flow and the capacities are given in Table 4. In large buildings provision should be made for quickly draining radiators in case of breakage, and it is often advisable to install a lock shield valve on one end of each radiator and a hand controlled valve on the other. In case of

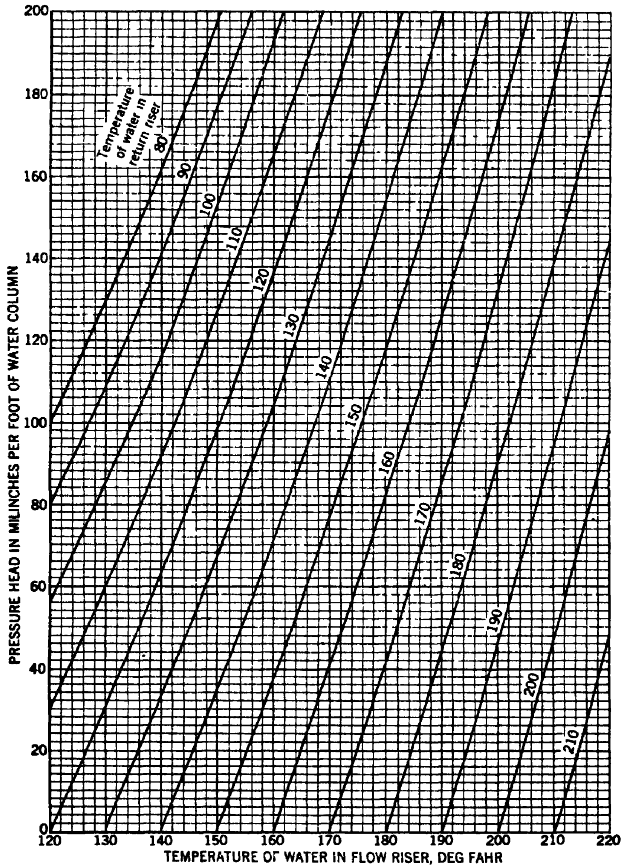


Fig. 4. Gravity Pressure Heads for Various Temperature Differences

breakage the two valves can be closed and the radiator removed without affecting the rest of the system. The lock shield valve can also be used for balancing the water circulation.

GRAVITY CIRCULATION PIPE SIZES

In gravity hot water heating systems the difference in temperature (density) between the flow and return water produces the required natural circulation of the water. The design temperature difference is usually assumed from between 20 to 35 F. After having determined the temperature difference and the temperature of the flow water, data given

in Fig. 4 can be used to obtain the pressure head. With information obtained concerning the required pressure head the same procedure is followed for computing the necessary data for a gravity circulation system as was previously outlined for a forced hot water system. Radiator heat emission rates from 150 to 200 Btu per square foot are commonly used so that flow temperatures generally range from 180 to 200 F or higher. Assuming a flow temperature of 200 F and a 35 F drop, and with the mains located 4 ft above the top of the boiler, a circulating pressure head of 600 milinches results. This is obtained by following the 200 F floor riser line in Fig. 4 to where it intersects the 165 F return riser line and reading horizontally a pressure head of 150 milinches per foot or 600 milinches for 4 ft. Assuming first floor radiators are located 3 ft above the mains and second floor radiators 12 ft above the mains, third floor 21 ft and fourth floor 30 ft, the circulating pressure heads are 450, 1800, 3150 and 4500 milinches respectively.

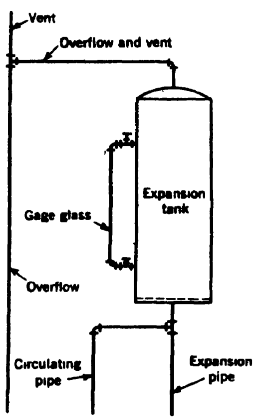


FIG. 5. · AN OPEN EXPANSION TANK

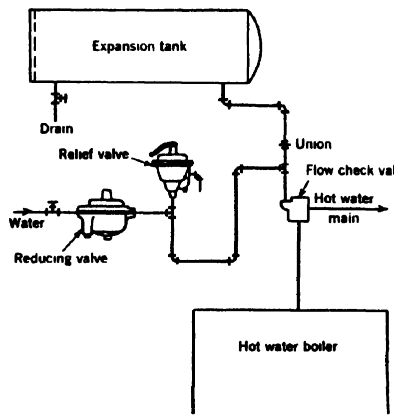


FIG 6. A CLOSED EXPANSION TANK

EXPANSION TANKS

Water heated from 40 F to 200 F expands about 0.04 of the original volume. The expansion tank permits the change in volume of the water in the heating system to take place without producing undesirable stresses due to pressure in any part of the system.

Expansion tanks may be open, as illustrated in Fig. 5, or closed as shown in Fig. 6.

An open expansion tank has free vent to the atmosphere and consequently the pressure on the surface of the water is always that of one atmosphere. The minimum contents of an open tank should be 0.06 of the volume of the water in the system including that in the boiler, heat transmitters, pipes, etc. This capacity is 50 per cent in excess of the actual increase in volume of water due to increase in temperature from 40 F to 200 F. The tank should be located at least 3 ft above the highest radiator. Provision must be made to prevent freezing of the water in the tank as well as in the pipe leading to the tank.

In a thermally circulating hot water heating system, the pipe to the open expansion tank should be connected to the supply riser from the

boiler, so that the air liberated from the water in the boiler will enter the expansion tank.

In a forced circulating hot water heating system, the pipe to the open expansion tank should be connected on the suction side of the circulating pump, so that the pressure head on the suction side of the pump will remain practically constant.

A closed expansion tank is sealed against free venting to the atmosphere. The tank may be above the highest radiator or heat transmitter, or may be below the lowest one. The minimum contents of a closed expansion tank must be such that the expansion of the water due to increase in temperature will be cushioned against a reservoir of compressed air above the water level in the expansion tank. The tank must provide space not only for the change in water volume, but also for variations in air volume within the tank due to changes in air pressure. If the closed expansion tank is below the heat transmitters, the tank should be larger than if it is above them, and the higher the building, under such circumstances, the larger should be the air capacity within the tank in excess of that required merely for increase in water volume due to temperature increase.

The size of a basement-located closed expansion tank should be at least equal to the following:

One story buildings:	$x = 0.10\ V$
Two story buildings:	$x = 0.13\ V$
Three story buildings:	$x = 0.17\ V$
Six story buildings:	$x = 0.28\ V$

where

x = expansion tank size in gallons.
V = water volume in heating system in gallons.

This condition favors, especially in tall buildings, the placing of the closed expansion tank above the highest heat transmitter.

Any closed expansion tank located above the heat transmitters of a hot water heating system should be connected by a direct pipe with the flow main leaving the boiler, in order to enable the air to pass easily to the expansion tank. In a closed hot water heating system the water under pressure tends to absorb air at a rate increasing with pressure increase and decreasing with temperature increase.

Means must be provided to adjust and to observe the proportion of air within any closed expansion tank. This involves the provision of an air inlet valve, a water gage and a relief valve. A source of supply of compressed air for renewing the air cushion is highly desirable, especially in large, high pressure hot water heating systems where it is inconvenient, if not impracticable, to drain down the water in the system so as to permit introduction of atmospheric pressure air.

For every hot water heating system the designer should calculate the volume of water contained in the radiators, piping system, boiler, etc., in order to select the proper size of expansion tank. The water content of the piping can be obtained from Table 5. For a rough selection of size, however, it is sometimes assumed that 50 per cent of the volume of water is contained in the radiators, and that the water content per square foot of radiator heating surface is 0.2 gal for column radiators and 0.13 gal for tube type radiators.

Another rough method for determining the size of an expansion tank

to be located above the highest radiator is to divide the square feet of radiation by the factor 40 to obtain the required capacity in gallons of the expansion tank.

INSTALLATION DETAILS

Items that should be considered in the design of this type of system are:

All piping must be so pitched that all air in the system can be vented either through an open expansion tank, radiators or automatic relief valves.

All piping must be arranged so that the entire system can be drained. Sections of piping individually valved shall have corresponding drain valves.

In large buildings, the piping may be zoned according to exposure of building, usage of building, or method of control.

All piping must be installed so that it is free to expand and contract with changes of temperature without producing undue stresses in the pipes or connections. For this purpose it is generally sufficient to allow for a variation in length of 1 in for 100 ft of pipe.

The pipe system should be designed so that each circuit has its correct friction head

TABLE 5. VOLUME OF WATER IN STANDARD PIPE

PIPE SIZE, IN.	LINEAL FT OF PIPE CONTAINING 1 GAL
½	63 1
¾	36 1
1	22 2
1¼	12 8
1½	9 47
2	5 75
2½	4 02
3	2.60
4	1 52
5	0 96
6	0 67

for balanced water distribution This may be done by change of pipe size or change in piping detail.

The connections from the boiler to the mains should be short and direct, to reduce the friction head and allow for expansion. It is frequently possible to avoid an elbow and to reduce the length of the pipe by running the pipe in a diagonal direction, either in a horizontal or in a vertical plane.

The mains and branches should pitch up and away from the heater, generally not less than 1 in. in 10 ft.

The connections from mains to branches and to risers should be such that circulation through the risers will start in the right direction Hence, in a one-pipe system the flow connection must be nearer the heater than the return connection. In a correctly-designed two-pipe system, the pressure in the flow main is higher than that in the return main, and a slight variation in the distances of the flow and return connections from the heater is not material; but it is generally best to have the two connections about equally distant from the heater.

Generally connections to risers or radiators are taken out of the top of mains either 45 or 90 deg. In some cases it may be advisable to take the flow connection off the top of the main and the return connection from the side.

With forced circulation and high velocities, it is advisable to let the water enter at the top of the radiator and leave at the bottom of the opposite end. With gravity circulation the flow connection may be either at the top or at the bottom of the radiator. With short radiators both flow and return may be at same end, but top and bottom.

Unless used as heating surface, all piping, both flow and return, should be insulated.

EXAMPLES OF PIPING DESIGN

The following graded series of examples of the design of hot water piping systems will illustrate the fundamental principles and methods. The differences between reversed return and direct return systems are

shown, and the methods of balancing the several radiators or circuits are illustrated. A simple gravity system is shown in Fig. 7 and an elementary forced circulation system is diagrammed in Fig. 8.

Elementary Gravity System

Example 1. A simple gravity-circulation system is illustrated in Fig. 7 with one radiator that is giving off heat at the rate of 20,000 Btu per hour or 20 Mbh. The boiler imparts heat to the water at the same rate, and the water circulates at a uniform velocity. The thermal or gravity *pressure head* which produces the circulation is equal to the friction head which resists the circulation. The circuit consists of 1 boiler, 1 radiator, 2 ells, 1 radiator valve and a total of 24 ft of pipe.

Solution. With the average water temperatures of 200 and 180 F in the supply and return risers, respectively, the pressure head will be 90 milinches per foot of water column. This pressure head may be found from Fig. 4. Since the center of the radiator is 10 ft above the center of the boiler, the total pressure head of the circuit is 10 x 90, or 900 milinches, or 0.9 inches of 190 F water. The friction head of the circuit must then also be 900 milinches. The friction head of 1 ft of 1 in. pipe is found from Fig. 3 to be about 46 milinches at 20 Mbh, and the corresponding velocity 9 in. per second. (Note that all values in Fig. 3 are based on a temperature difference of 20 F.)

Similarly, if a 1¼ in. pipe were to be used, the friction head would be about 12 milinches per foot and the corresponding velocity about 5 in. per second, from Fig. 3.

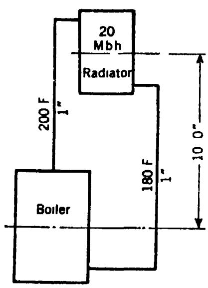


Fig. 7. Gravity System

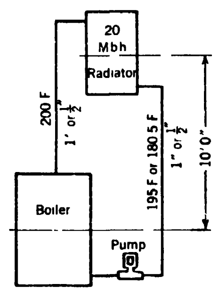


Fig. 8. Forced Circulation System

To find the friction head in the elbows, boiler, radiator and valve, Table 3 is used, and the entire circuit is found to be equal to 10 elbow-equivalents plus 24 ft of pipe. Each elbow-equivalent is equal to a pipe length of 25 times the nominal diameter. Then the equivalent lengths of straight pipe are 45 ft of 1 in. pipe or 50 ft of 1¼ in. pipe.

Hence, if 1 in. pipe is used, the *fh* (friction head) of the circuit will be 45 x 46, or 2070 milinches, and if 1¼ in. pipe is used, the *fh* will be 50 x 12, or 600 milinches. A 1 in. pipe would, therefore, be too small and a 1¼ in. pipe too large to permit the desired circulation with a flow-return temperature difference of 20 F.

If the circuit is of 1 in. pipe, the circulation will take place with a temperature difference greater than 20 F, and if the circuit is of 1¼ in. pipe, the circulation will take place with a temperature difference smaller than 20 F. To find, for example, the temperature difference at which a circuit of 1 in. pipe would transmit the required 20 Mbh, assume the difference to be 40 F. The *ph* (pressure head) would be (Fig. 4, from 200 to 160) 175 milinches per foot, or 1750 for the system. The *fh* for the system may be found from Fig. 3; the chart of this figure is based on a temperature difference of 20 F; if the temperature difference were 40 F, the heat conveyed would be twice that shown in the chart. Hence, find 10 Mbh on the lower scale, proceed vertically upward to the intersection with the 1 in. line, and from there to the left scale and read 13 milinches per foot. Note that the velocity would then be only about 5 in. per second. The total *fh* would then be 45 x 13 or 585 milinches. Since the *ph* would be 1750, circulation would take place with a temperature difference less than 40 F. The required temperature difference may be determined by constructing the diagram of Fig. 9, from which it appears that the temperature difference with which the 1 in. pipe circuit would function is about 30 F. Hence, if the flow riser temperature is 200, the return riser temperature will be 170, and the average water temperature in the radiator, about 185 F.

Elementary Forced Circulation System

Example 2. Design a system for the piping arrangement shown in Fig. 8, according to one of the outlined procedures. The *ph* developed by the circulating pump and the pipe size may be assumed and the flow-return temperature difference found; or, the *ph* developed by the pump and the flow-return temperature difference may be assumed and the pipe size found; or the pipe size and the flow-return temperature difference may be assumed and the *ph* found which the circulating pump must develop.

Solution. Assume that the circulating pump will develop a *ph* of 2 ft or 24,000 milinches and that a 1 in. pipe is to be used. The equivalent length of the circuit will then be 45 ft, as in Fig. 7, and the available *ph* will be 24,000/45, or 533 milinches per foot. In Fig. 3, find 533 on the left scale, move horizontally to the intersection with the 1 in. pipe line, and read about 77 Mbh delivered by the pipe (with a velocity of about 35 in. per second) for a temperature difference of 20 F. Since the circuit is to deliver only 20 Mbh, the temperature difference will be 20 divided by 77 and multiplied by 20, or 5.2 F. Hence, if the flow riser temperature is 200, the return riser temperature will be about 195, and the average water temperature in the radiator, about 197.5 F.

If a ½ in. pipe were used instead of a 1 in., the equivalent length of circuit would be 35 ft instead of 45; the unit *ph*, 686 milinches instead of 533; the velocity, 27 in. per second instead of 35; the temperature difference, 19.5 instead of 5.2; and the average water temperature in the radiator, about 190.5 instead of 197.5 F.

If the 1 in. pipe is used for the circuit, the gravity *ph* will be 22 milinches per foot,

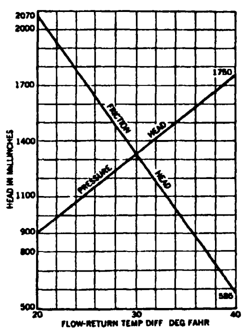


FIG. 9. DETERMINATION OF REQUIRED TEMPERATURE DIFFERENCE

or 220 for the circuit (Fig. 4, 200 to 195). Since this is only 1 per cent of the pump *ph* (24,000 milinches), it may be neglected in the calculation, as was done previously. However, there are cases in which the gravity *ph* is so large compared with the pump *ph*, that it should be included in the calculation.

The methods just described for the design of the two elementary systems are fundamental and apply to the design of all hot water heating systems. In every system, however large and complicated, the pipe system must be such that the *ph* forcing the water from the boiler to any one radiator is equal to the *fh* in that radiator's circuit when the radiator is receiving its proper quantity of hot water and the system is functioning at a steady rate.

Two-Pipe Gravity Circulation System

Example 3. In the system shown in Fig. 10, water leaving the boiler may flow to any one of the three radiators. If the system is designed correctly, each radiator will receive its proper share of the hot water. Since Radiator 3 has the largest load and is also farthest from the boiler, it is the least favorably located with reference to circulation, and its circuit should be designed first. If the pipes leading to it are large enough, it will be easy to secure sufficient circulation for the other two radiators.

The system is to function with a 40 F flow-return temperature difference. The *ph* for each radiator is 7 x 175 (Fig. 4), or 1225 milinches; the *fh* for each radiator circuit must, therefore, also be 1225 milinches.

Solution. In order to design a radiator circuit accurately and systematically, the circuit should be divided into sections. The division points of sections must be where the pipe sizes change or may change, and where the volume of water flowing in the pipe changes The data relating to the several sections may be recorded as shown in Table 6.

Data recorded in Table 6 show that Circuit 3 consists of 70.4 ft of pipe and 21.5 elbow equivalents. Assuming that the average size of the pipe will be 1¼ in., the 21.5 elbow equivalents may be replaced by 21.5 x 2.6 (25⁄12 x 1.25 x 1.0), or 56 ft of pipe, which would make the total equivalent length of the circuit 70.4 plus 56 or 126.4 ft of pipe, and the average *fh*, 1225/126.4, or about 10 milinches per foot. For this unit *fh* and a temperature difference of 40 F (see Fig. 3), a 1 in. pipe will convey 18 Mbh, a 1¼ in. pipe, 37 Mbh, and a 1½ in. pipe, 56 Mbh. The pipe sizes for the several sections of

TABLE 6. TABULATED DATA FOR EXAMPLE 3

Section	Load Mbh	Pipe Length Ft	Elbows No	Pipe Size In	Equivalent Length, Ft	Unit Friction Milinches per Ft	Total Friction, Milinches
Circuit No 3 Available ph 1225 milinches							
0-4	45	3.5	2.5	1½	11.3	6.3	71
4-5	35	10.0	2.0	1¼	15.2	9.2	140
5-6	20	12.0	0.0	1¼	12.0	3.4	41
6-3	20	6.5	5.5	1	17.5	13	228
3-7	20	9.5	5.5	1	20.5	13	267
7-8	20	12.8	1.0	1	14.9	13	194
8-9	35	13.8	2.5	1¼	20.3	9.2	187
9-0	45	2.3	2.5	1½	10.1	6.3	64
Total		70.4	21.5		121.8		1192
Circuit No 2 Available ph 1225 milinches							
0-4	45						71
4-5	35						140
5-2	15	6.2	6	1	18.7	7.5	140
2-8	15	9.5	7	¾	20.4	24.0	490
8-9	35						187
9-0	45						64
Total							1092
Circuit No 1 Available ph 1225 milinches							
0-4	45						71
4-10	10	9.0	1.5	¾	11.3	11	124
10-1	10	5.0	5.5	½	10.7	46	434
1-11	10	9.5	5.5	¾	18.1	11	199
11-9	10	11.5	2.5	¾	15.4	11	169
9-0	45						64
Total							1067

Circuit 3 may be selected as indicated in Table 6. Having selected the pipe sizes, the unit friction heads may be found from Fig. 3 and the total friction head calculated and recorded as shown in Table 6. If the grand total, in the present case 1192 milinches, differs materially from the available *ph*, 1225 milinches, one or more of the pipe sizes must be changed and the calculation repeated until the total *fh* is practically equal to the available *ph* of 1225 milinches.

It is not necessary, in the design of hot water heating systems, to be extremely careful to have the *fh* exactly equal to the available *ph*, because a hot water heating system has the ability to adjust itself to varying conditions of considerable magnitude. For example, in the present case, the calculated *fh* is 1192 milinches, or about 3 per cent less than the calculated available *ph*; the water would, therefore, circulate a little faster than contemplated and the return temperature would be a little higher than 160 F. This would immediately lower the available *ph* and the *ph* and *fh* would come into balance at a value higher than 1192 and lower than 1225.

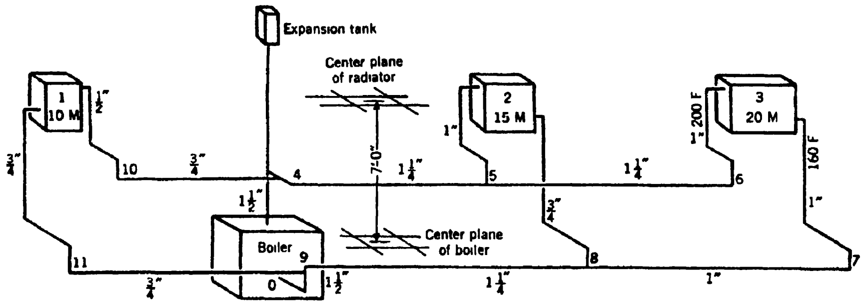


FIG. 10. TWO-PIPE GRAVITY CIRCULATION SYSTEM

Since it is generally not necessary to make extremely refined calculations, Table 1 may often be used instead of the chart of Fig. 3 to determine pipe sizes. For example, in the present case, Table 1 shows that for an *fh* of 10 milinches a 1½ in. pipe would convey 28 Mbh with a temperature difference of 20 F or 56 Mbh with a temperature difference of 40 F. Since the 1½ in. pipe in Fig. 10 is to convey only 45 Mbh with a temperature difference of 40 F, or only 22.5 with a temperature difference of 20 F, it is evident from the table that the *fh* will be between 6 and 8 milinches. For 6 milinches the heat conveyed is 21.2 and for 8 milinches, it is 24.8; for 22.5 Mbh, the *fh* would be estimated to be about 6.5, which would be sufficiently accurate for the present calculation.

Having completed the design of Circuit 3, it is simple to design Circuit 2 because it has four sections in common with Circuit 3 and it is only necessary to design Sections 5-2 and 2-8, as shown in Table 6, so that the total *fh* of Circuit 2 will be practically equal to the total *fh* of Circuit 3.

Having completed the design of Circuit 2, it is necessary to design Circuit 1, as shown in Table 6, so that its total *fh* will be approximately equal to the total *fh* of the other two circuits since all three circuits have equal pressure heads.

Two-Pipe Forced Circulation System

Example 4. In the design of a system for Fig. 11, as in the design of Example 2, there are three unknowns—pressure head, pipe size, and flow-return temperature difference, any two of which may be assumed and the third found. In the design of a system for Fig. 8, the pressure head and the temperature difference were assumed and the pipe sizes found. In this case, the pressure head is to be found.

Solution In selecting the temperature difference and the pipe sizes, it should be borne in mind that the first cost of the pipe system and of the radiation is reduced by reducing the temperature difference and by reducing the pipe sizes, but the *ph* and the cost of pumping the water are increased. The choice of temperature difference and pipe sizes which produce the greatest economy in first cost and in cost of operation can be determined after having made two or three trial designs. For the first design, 20 F will be selected as the temperature difference and the pipe sizes will be chosen as shown in Fig. 11. A calculation similar to that of Table 7 will show that the *fh* of Circuits 1, 2, and 3

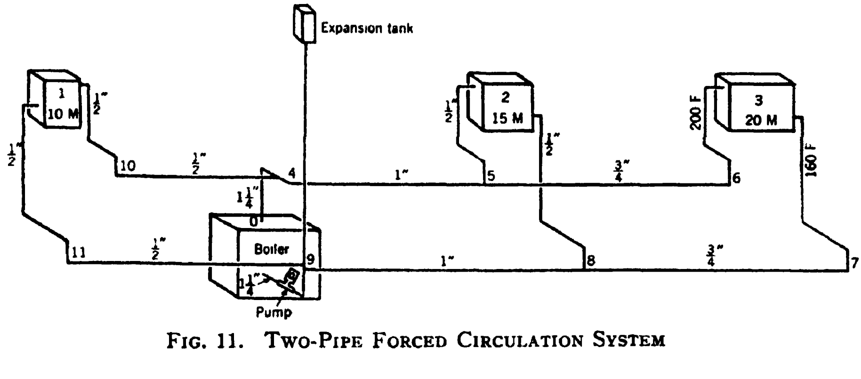


FIG. 11. TWO-PIPE FORCED CIRCULATION SYSTEM

will be, respectively, about 9000, 15,300, and 14,300 milinches. To increase the *fh* of Circuit 1 from 9000 to 15,000 milinches would require the insertion in the circuit of a section of ⅜ in. pipe, or an orifice resistor, or a regulating valve. However, this would cause unnecessary expense. The system will function well with the pipe system shown in Fig. 11. If the circulator maintains a pressure head of 15,300 milinches, the velocity in Circuit 1 will increase until the *fh* of the circuit is also 15,300 milinches; i. e., its *fh* will be increased from the calculated 9000 to the required 15,300, or 6300 milinches.

As calculated, the three radiators are to dissipate 10, 15, and 20 Mbh, respectively, with a temperature difference of 20 F. Consequently, water must flow through these three radiators at the rates of 1, 1.5, and 2 gpm, respectively. When water flows through a ½ in. pipe at the rate of 1 gpm, the velocity in the pipe is (Fig. 3) about 13 in. per second and the unit *fh* is 165 milinches. The equivalent length of sections 4-10, 10-1, 1-11, and 11-9 of this circuit is 50 ft, and its total *fh* is 8250. In order that the *fh* may be increased 6300 milinches, the unit *fh* must be increased 126 milinches; consequently, the velocity in the ½ in. pipe (Fig. 3) must be increased from 13 to 16 in. per second. Hence, when the *fh* of Circuit 1 has been increased to 15,300, water will flow through Radiator 1 at the rate of 16 ÷ 13, or 1.23 gpm. This increase in volume of water will increase the load on the circulating pump in the proportion of 450 to 473 and will increase

TABLE 7. TABULATED DATA FOR EXAMPLE 7

Circuit	Load Mbh	Pipe Length Ft	Elbows No.	Equivalent Length Ft	Size In.	Unit Friction Milinches per Ft	Total Friction, Milinches
0-1	3600	65	1 8	78	6	100	7800
1-2	3150	130	0	130	5	180	23,900
2-3	2700	130	0	130	5	150	19,500
3-4	2250	130	0	130	5	100	13,000
4-5	1800	130	2	142	4	210	29,800
5-6	1350	130	0	130	4	125	16,200
6-7	900	130	0	130	3½	113	14,700
7-8	450	130	0	130	2½	190	24,700
8-16	450	Estimated Friction Head					20,000
16-17	3600	65	1 8	78	6	100	7,800
17-18	7200	130	0	130	8	90	11,670
						Total	188,570

the heat dissipation of Radiator 1 slightly (about 3 per cent) but otherwise will not affect the operation of the system.

One-Pipe Gravity Circulation System

Example 5. A one-pipe system is one in which the water flows through more than one radiator before it returns to the boiler to be reheated. A two-pipe system as shown in Figs. 10 and 11 is one in which the water returns to the boiler to be reheated after it has passed through one radiator. Many large heating systems contain some one-pipe and some two-pipe sections. The piping system shown in Fig. 12 functions with a flow-return temperature difference of 40 F.

Solution. Since the four radiators are each to deliver 15 Mbh, and since the water is to leave the boiler at 200 F and return at 160 F, Radiator 1 will receive 200 F water, Radiator 2, 190 F water, Radiator 3, 180 F water, and Radiator 4, 170 F water. Since the system is to supply 60 Mbh with a temperature difference of 40 F and since 1 lb of water liberates 1 Btu when cooled 1 F, it is necessary that water circulate through this system at the rate of 60,000 ÷ 40, or 1,500 lb per hour, or 25 lb per minute. Assuming one gallon of water to weigh 8.33 lb, water must circulate in the system at the rate of 3 gpm. If the temperature difference were 20 F instead of 40, the circulation would be at the rate of 6 gpm. It is well to remember that, with a temperature difference of 20 F, water circulating at the rate of 1 gpm will convey heat at the rate of 10 Mbh. The chart of Fig. 3 shows the rate of circulation in gpm on the upper scale and the corresponding heat conveyance on the lower scale.

The system may be divided into 5 separate systems. Each of the four radiators with its flow and return lines constitutes an elementary heating system (similar to Example 1), and the flow main with its two risers is also a complete elementary system.

If the center of the boiler is 4 ft below the center of the flow main, and if the flow

riser contains 200 F water and the return riser, 160 F water, the *ph* for the main circuit is (Fig. 4) 4 x 175, or 700 milinches. The circuit consists of 110 ft of pipe and 10 elbow equivalents; its equivalent length is about 150 ft if a 2 in. pipe is used as the main. The average *ph* will be 700 ÷ 150, or 4.7 milinches. According to Fig. 3 or Table 1 for a 4.7 milinch *fh*, a 2 in. pipe will convey about 70 Mbh. Since the pipe is to convey only 60 Mbh, it is slightly too large but should be used. The *fh* will, then, be 3.5 milinches instead of the permissible 4.7. The water will circulate with a temperature difference slightly less than 40 F, and the three last radiators would receive water slightly warmer than indicated in Fig. 12.

As the water flows in the main and arrives at one of the four points marked *A*, the flow will be divided and a portion of the water will take the short path in the main to the point *B*, and the remainder will take the long path through the radiator to the point *B*. Since the two paths together offer less resistance to the flow than the one path alone, the unit *fh* will be less than 3.5 milinches between the points *A* and *B*. If the distance from *A* to *B* is 4 ft, and if, for example, the flow in the short path is 2 gpm, the *ph* forcing the water along the two paths is 4 x 1.6, or about 6 milinches. Since the *fh* of the long path is much greater than the *fh* of the short path, only a comparatively small portion of the water would take the long path. However, the gravity head of the radiator supplies an additional *ph* for the long path. If the center of the radiator is 4 ft above the

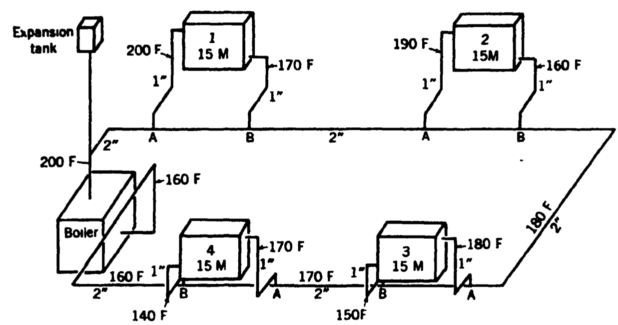


FIG. 12. ONE-PIPE GRAVITY CIRCULATION SYSTEM

main and if the radiator circuit is designed for a temperature difference of 30 F, the radiator *ph* will be about 4 x 120, or 480 milinches. In that case, the *ph* is 6 milinches for the short path and 486 milinches for the long path. The radiator circuit consists of 11 ft of pipe and about 14 elbow equivalents. If the circuit is of 1 in. pipe, its equivalent length is about 40 ft and the unit *fh* should be 486 ÷ 40, or 12 milinches. For this *fh* and 30 F temperature difference, a 1 in. pipe conveys about 15 Mbh (Fig. 3). A 1 in. pipe is, therefore, the correct size, and the water would flow through the radiator with a temperature difference of 30 F.

Since each of the four radiators delivers one fourth of the total heat, and since the total temperature difference is to be 40 F, the water would cool 10 F in every radiator if all the water passed through every radiator. Since the water cools about 30 F in flowing through the radiator only ⅓ of the water flows through the radiator and ⅔ flow through the main from point *A* to point *B*.

It follows from these calculations that, if the main is of 2 in. pipe and the radiator branches are of 1 in. pipe, water will circulate through the system at the rate of 3 gpm with a temperature difference somewhat less than 40 F and that, at the radiator connections, water will circulate through the radiators at the rate of 1 gpm and that, consequently, between radiator branch connections the *fh* is at a lower rate since only 2 gpm flow through in the main between those points.

One-Pipe Forced Circulation System

Example 6. The system shown in Fig. 12 as a gravity circulation system may be changed to a forced circulation system by inserting a circulating pump as shown in Fig. 13. The location of the expansion tank should then be changed as indicated.

Solution. To design this system the pipe sizes and the temperature difference may be assumed; for example, 1¼ in. pipe may be selected for the main and ¾ in. pipe for the radiator branches and risers, and 20 F as the flow-return temperature difference.

Since the system is to deliver 60 Mbh with a temperature difference of 20 F, the pump must circulate 60 ÷ 10, or 6 gpm.

For this load, the unit *fh* for a 1¼ in. pipe is 86 milinches. The main circuit consists of 110 ft of pipe and 10 elbow equivalents and may be placed equal to 136 ft of 1¼ in. pipe. The total *fh* for the main circuit is 136 x 86, or 11,696 milinches, or practically 1 ft for a flow of 6 gpm.

At the points *A*, a portion of the water will be diverted through the radiator circuit, and as a result less than 6 gpm will flow in the main between the points *A* and *B*, and the *fh* will be slightly less than 86 milinches per foot between these points. But the difference will be so small that it may be neglected and the total *fh* from *A* to *B* assumed to be 4 x 86, or 344 milinches. The *ph* forcing the water through the radiator circuit will then be 344 milinches. The radiator circuit consists of 11 ft of pipe and about 14 elbow equivalents and may be placed equal to 32 ft of pipe and the available *ph* 344 ÷ 32, or about 11 milinches per foot. With this *ph*, a ¾ in. pipe will convey about 5 Mbh (Fig. 3), or 0.5 gpm. Hence, only 5 ÷ 60, or about 8 per cent of the water, would flow through the radiator, if the radiator's gravity head is not considered. The water would, then, have to cool 60 F in order to deliver 15 Mbh, and the average radiator temperature

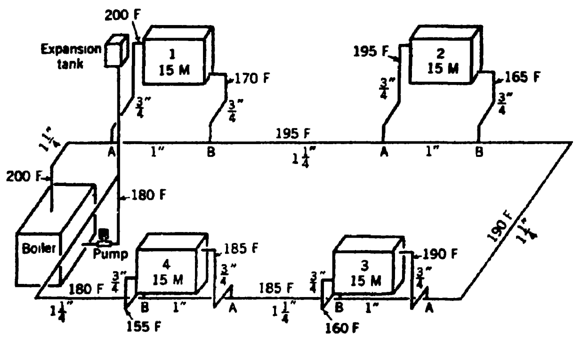


FIG. 13. ONE-PIPE FORCED CIRCULATION SYSTEM

would be 170 if the water entered at 200. This would require a large radiator and result in an unsatisfactory installation.

To secure a larger flow of water through the radiator it is necessary to increase the *fh* of the short path *A*-*B* in the main. This may be done by inserting special resistance tees at points *A* and *B*, or by inserting an orifice resistor between points *A* and *B*, or by reducing the 1¼ in. main between the points *A* and *B* to the next smaller size, *i.e.* 1 in.

The relative quantity of water flowing through the radiator may then be found by trial calculations. Assume, first, that 1 gpm will flow through the radiator and 5 gpm through the main. The *ph* for 1 gpm and a ¾ in. pipe is 40 milinches per foot, or 32 x 40, or 1280 for the radiator circuit.

The main circuit consists of 4 ft of 1 in. pipe and two reducing tees. The two reducing tees may be placed equal to 0.8 elbow equivalents (Table 3), and the equivalent length of the main circuit equal to 5.7 ft.

The *fh* for 5 gpm and a 1 in. pipe is 240 milinches per foot, or 5.7 x 240, or 1370 milinches for the main circuit. Since this is only slightly more than the calculated *fh* for the radiator circuit, it is evident that the flow through the radiator will be slightly more than 1 gpm, and it is not necessary to make a second trial calculation. The quantity of water flowing through the radiator can be varied by varying the distance between the points *A* and *B*, where the radiator branches join the main.

In order to deliver 15 Mbh to the radiator with a temperature difference of 20, it is necessary that 1.5 gpm flow through the radiator; since, in this case, the flow through the radiator is only 1 gpm, the temperature difference must be 30 F.

If the water enters the radiator at 190 F, the average water temperature will be 175 F. The quantity of water circulating through the radiator may be varied considerably without an appreciable effect on the quantity of heat dissipated by the radiator. This is evident from the following calculation.

Assume that Radiator 2 has been designed so that it will dissipate 15 Mbh when its flow of water is at the rate of 1 gpm, and when its average temperature is 175 F. Assume

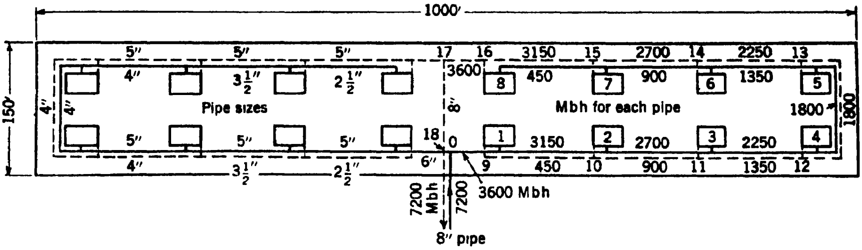


FIG. 14. TWO-PIPE REVERSED RETURN SYSTEM

that the flow of water is increased 50 per cent—from 1 gpm to 1.5 gpm. The water will then flow through the radiator in two-thirds the time and will cool two-thirds as much; *i e.*, it will cool 20 F instead of 30 F, and the average radiator temperature will be 180 F instead of 175 F. If the surrounding temperature is 70 F, the temperature differences, radiator and surroundings, will be 110 and 115 F, respectively. Consequently, the heat dissipation will be increased only about 6 per cent when the quantity of water flowing through the radiator is increased 50 per cent.

By decreasing the main from 1¼ to 1 in. between radiator branches while the flow is decreased from 6 to 5 gpm, the *fh* in that section of the main is increased from 344 to 1370 milinches, or 926 milinches. Hence, for the four radiator sections the increase is 3704 milinches, and the total *fh* for the circuit will be 11,696 plus 3704, or 15.4 in. instead of 11 7 in. as first calculated. The pump must, therefore, circulate 6 gpm against a head of 1 3 ft.

Reversed and Direct Return Systems

In a reversed return system the radiators are connected so that all circuits are practically of equal length and so that the water flowing through the radiator nearest the boiler must travel practically as far as the water flowing through the radiator farthest from the boiler, as illustrated in Fig. 14. In a direct return system the radiators are connected so that all water returns to the boiler along the most direct path after it has passed through its radiator, as illustrated in Fig. 15.

Example 7. In Fig. 14, sixteen air conditioning units, each demanding 450 Mbh, are to be supplied with water from a central plant. The system is divided into two equal parts as shown. Each part supplies eight units and has, therefore, eight circuits. The total length of each of the eight circuits is about 1170 ft.

Solution. If the total *fh* is to be about 15 ft, the unit *fh* must be about 150 milinches per foot. With this preliminary estimate, pipe sizes may be selected from Fig. 3 and recorded with corresponding calculations as shown in Table 7 for Circuit 8, from which it appears that the *fh* of this circuit is 188,570 milinches, or 15.7 ft.

In order that each of the eight air conditioning units may receive an equal supply of water, the *fh* of each of the remaining circuits must also be 15.7 ft. Since all pipe sizes have been selected as shown in Example 6, any adjustments that may be necessary must be made in the connections from the main through the air conditioning unit and back to the main. For Circuit 8 the friction head through the air conditioning unit was assumed to be 20,000 milinches. For Circuit 4, for example, a tabular calculation like

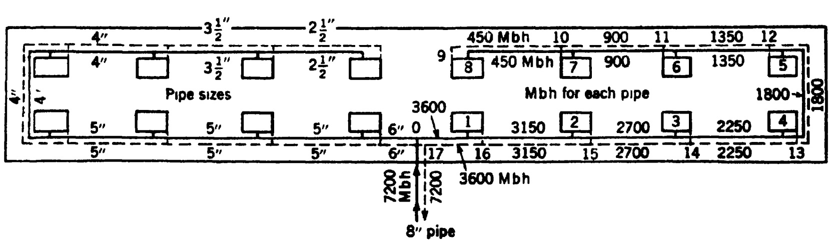


FIG. 15. TWO-PIPE DIRECT RETURN SYSTEM

that for Circuit 8 shows that the *fh* for Section 4-12 must be 19,700 milinches in order that the total *fh* may be 15.7.

This is practically equal to the 20,000 *fh* assumed for Circuit 8 and this shows how simple it is to secure well-balanced circuits in a reversed return system.

Example 8. The direct return, forced circulation system shown in Fig. 15 is similar to Fig. 14, except that the water passing through Unit 1 returns directly to the boiler and the total length of this circuit is about 130 ft, whereas, the total length of Circuit 8 is about 1990 ft, or about 15 times as long.

Solution. The design must begin with Circuit 8. The calculations for this circuit if tabulated as shown for Circuit 8 of Fig. 14 will show that the total *fh* is 318,200 milinches, or 26.5 ft, as compared with 15.7 ft for the reversed return system.

In order that each of the eight units will receive its correct share of the water, the *fh* of each of the other seven circuits must also be 26.5 ft. For Circuit 1, for example, the total *fh* for Section 0-1 and 16-17 is 15,600 milinches; hence, the *fh* in Section 1-16 (through the air conditioning unit) must be 302,600 milinches, or 25.15 ft to prevent Unit 1 having an advantage over Unit 8.

Comparing the reversed return system of Fig. 14 with the direct return system of Fig. 15, it appears that the head against which the pump must deliver the 720 gpm is 15.7 ft as compared with 26.5 ft for the direct return and that the installation of the reversed return would require 130 ft of 8 in. pipe not necessary for the direct return system. The *fh* in the lines joining the pump to the pipe system shown in Figs. 14 and 15 is not included in this calculation.

CHAPTER 15

Pipe, Fittings, Welding

Pipe Material, Types of Pipe Used, Dimensions of Pipe Commercially Available, Expansion and Flexibility of Pipe, Pipe Threads and Hangers, Types of Fittings, Welding as Applied to Erection of Piping, Valves, Corrosion of Piping

IMPORTANT considerations in the selection and installation of pipe and fittings for heating, ventilating, and air conditioning work are dealt with in this chapter.

PIPE MATERIALS

Use of corrosion-resistant materials for pipe, including special alloy steels and irons, wrought-iron, copper and brass, has increased considerably during the past few years. The recent development of copper, brass, and bronze fittings which can be assembled by soldering or sweating permits the use of thin-wall pipe and thereby has reduced the initial cost of such installation. The following brief discussion indicates the variety of pipe materials and the types of pipe available.

Wrought-Steel Pipe. Because of its low price, the great bulk of wrought pipe used for heating and ventilating work at the present time is of wrought steel. The material used for steel pipe is a mild steel made by the acid-bessemer, the open-hearth, or the electric-furnace process. Ordinary wrought-steel pipe is made either by shaping sheets of metal into cylindrical form and welding the edges together, or by forming or drawing from a solid billet. The former is known as *welded pipe*, the latter as *seamless pipe*.

Many types of welded pipe are available, although the smaller sizes most frequently used in heating and ventilating work are made by the lap-weld, resistance-weld, or butt-weld process. While the lap-weld and resistance-weld processes produce a better weld than the butt type, lap-weld and resistance-weld pipe are seldom manufactured in nominal pipe sizes less than 2 in. Seamless pipe can be obtained in the small sizes at a somewhat higher cost.

Seamless steel pipe is frequently used for high pressure work or where pipe is desired for close coiling, cold bending, or other forming operation. Its advantages are its somewhat greater strength which permits use of a thinner wall and, in the small sizes, its freedom from the occasional tendency of welded pipe to split at the weld when bent.

Wrought-Iron Pipe. Wrought-iron pipe is claimed to be more corrosion-resisting than ordinary steel pipe and therefore its somewhat higher first cost is said to be justified on the basis of longer life expectancy. Wrought-iron pipe may be identified by the spiral line marked into each length, either knurled into the metal or painted on it in red or other bright color. Otherwise, there is little difference in the appearance of wrought iron and steel pipe, although microscopic examination of polished and etched specimens will readily disclose the difference.

Cast-Ferrous Pipe. There are now available several types of cast-ferrous metal pipe made of a good grade of cast-iron with or without additions of nickel, chromium, or other alloy. This pipe is available in sizes from 1½ in. to 6 in., and in standard lengths of 5 or 6 ft with external and internal diameters closely approximating those of extra strong

wrought pipe. Cast-ferrous pipe may be obtained coupled, beveled for welding, or with ends plain or grooved for the several types of couplings. It is easily cut and threaded as well as welded. The fact that it is readily welded enables the manufacturers to supply the pipe in any lengths practicable for handling.

Alloy Metal Pipe. Steel pipe bearing a small alloy of copper or other alloying element and iron pipe bearing a small alloy of copper and molybdenum have been claimed to possess more resistance to corrosion than plain

TABLE 1. DIMENSIONS OF SCHEDULES 30 AND 40 AND STANDARD WEIGHT PIPE[a]

Size	Diameter In.		Thickness[b], In.	Weight per Ft, Lb		Threads per In.	Circumference, In		Transverse Area, Sq In.			Length of Pipe, Ft per Sq Ft		Length of Pipe, Ft Containing 1 Cu Ft	Weight of Water, Lb per Ft
	External	Internal		Plain Ends	Threads and Couplings		External	Internal	External	Internal	Metal	External Surface	Internal Surface		
1/8	0.405	0.269	**0.068**	0.244	0 245	27	1 272	0 845	0.129	0 057	0.072	9 431	14.199	2533.775	0.025
1/4	0.540	0.364	**0.088**	0.424	0 425	18	1 696	1 144	0 229	0 104	0 125	7 073	10.493	1383 789	0 045
3/8	0.675	0.493	**0.091**	0.567	0 568	18	2 121	1 549	0 358	0 191	0.167	5 658	7.748	754.360	0 083
1/2	0.840	0 622	**0.109**	0 850	0 852	14	2 639	1 954	0 554	0 304	0.250	4.547	6 141	473.906	0 132
3/4	1.050	0 824	**0.113**	1 130	1.134	14	3 299	2 589	0.866	0 533	0 333	3.637	4 635	270.034	0 231
1	1.315	1 049	**0.133**	1 678	1 684	11½	4 131	3 296	1 358	0.864	0 494	2 904	3 641	166 618	0 375
1¼	1 660	1 380	**0.140**	2.272	2 281	11½	5 215	4 335	2 164	1.495	0.669	2 301	2.768	96 275	0 65
1½	1.900	1.610	**0.145**	2.717	2 731	11½	5 969	5 058	2 835	2 036	0.799	2 010	2 372	70 733	0 88
2	2 375	2 067	**0.154**	3 652	3 678	11½	7 461	6 494	4 430	3 355	1.075	1.608	1.847	42 913	1 45
2½	2 875	2.469	**0.203**	5 793	5 819	8	9 032	7 757	6 492	4 788	1 704	1.328	1.547	30.077	2 07
3	3 500	3 068	**0.216**	7 575	7 616	8	10.996	9 638	9 621	7 393	2 228	1.091	1 245	19 479	3 20
3½	4.000	3.548	**0.226**	9 109	9 202	8	12 566	11 146	12 566	9 886	2 680	0 954	1 076	14 565	4 29
4	4.500	4 026	**0.237**	10 790	10.889	8	14 137	12 648	15 904	12 730	3 174	0 848	0.948	11 312	5 50
5	5.563	5 047	**0.258**	14 617	14 810	8	17 477	15 856	24 306	20 006	4 300	0 686	0.756	7.198	8 67
6	6.625	6 065	**0.280**	18.974	19 185	8	20 813	19 054	34 472	28 891	5 581	0.576	0.629	4 984	12 51
8[c]	8.625	8.071	0 277	24 696	25 000	8	27 096	25 356	58 426	51 161	7.265	0.443	0.473	2 815	22 18
8	8 625	7 981	**0.322**	28 554	28 809	8	27 096	25 073	58 426	50 027	8 399	0 443	0 478	2 878	21 70
10[c]	10 750	10 136	0 307	34 240	35 000	8	33 772	31 843	90 763	80 691	10 072	0.355	0.376	1 785	34 95
10	10 750	10.020	**0.365**	40 483	41 132	8	33 772	31 479	90 763	78 855	11 908	0 355	0.381	1 826	34 20
12[c]	12.750	12.090	0.330	43 773	45 000	8	40 055	37 982	127 676	114.800	12 876	0 299	0 315	1 254	49 70
12	12.750	12 000	0 375	49 562	50.706	8	40 055	37 699	127 676	113 097	14 579	0 299	0 318	1 273	49 00

[a]Standard-weight wrought-iron pipe has approximately the same wall thicknesses and weights as contained herein for steel pipe. For exact dimensions, see *American Standard for Wrought-Iron and Wrought-Steel Pipe, A S A* B36.10.

[b]Thicknesses shown in bold face type are identical with thicknesses for Schedule 40 pipe of *A S.A* B36.10.

[c]Same as Schedule 30, *A.S A*. B30 10

steel pipe and they are advertised and sold under various trade names.

Copper Pipe and Fittings. Owing to its inherent resistance to corrosion, copper and brass pipe have always been used in heating, ventilating, and water supply installations, but the cost with standard dimensions for threaded connections has been high. The recent introduction of fittings which permit erection by soldering or sweating allows the use of pipe with thinner walls than are possible with threaded connections, thereby reducing the cost of installations.

The initial cost of brass and copper pipe installations generally runs higher than the corresponding job with steel pipe and screwed connections in spite of the use of thin wall pipe, but the corrosive nature of the fluid conveyed or the inaccessibility of some of the piping may warrant use of a more expensive material than plain steel. The advantages of corrosion-

resisting pipe and fittings should be weighed against the correspondingly higher initial cost.

COMMERCIAL PIPE DIMENSIONS

The two weights of steel and wrought-iron pipe commonly used are known as *standard weight* and *extra strong*, which correspond to Schedules 40 and 80 respectively of the *American Standard for Wrought-Iron and Wrought-Steel Pipe, A.S.A.* B36.10. The same external diameter is used for both weights of each nominal size for manufacturing reasons as well

TABLE 2. STANDARD WEIGHTS AND DIMENSIONS OF WELDED AND SEAMLESS STEEL PIPE[a]

Size	Outside Diameter, In	No of Threads per In	Standard-Weight Pipe				Extra-Strong Pipe				Double Extra-Strong Pipe[b]	
			Schedule 30		Schedule 40		Schedule 60		Schedule 80			
			Wall Thickness, In	Weight per Ft, Lb T & C	Wall Thickness, In	Weight per Ft, Lb T & C	Wall Thickness, In	Weight per Ft, Lb Plain Ends	Wall Thickness, In	Weight per Ft, Lb Plain Ends	Wall Thickness In	Weight per Ft, Lb Plain Ends
1/8	0.405	27			0.068	0.25			0.095	0.31		
1/4	0.540	18			0.088	0.43			0.119	0.54		
3/8	0.675	18			0.091	0.57			0.126	0.74		
1/2	0.840	14			0.109	0.85			0.147	1.09	0.294	1.71
3/4	1.050	14			0.113	1.13			0.154	1.47	0.308	2.44
1	1.315	11 1/2			0.133	1.68			0.179	2.17	0.358	3.66
1 1/4	1.660	11 1/2			0.140	2.28			0.191	3.00	0.382	5.21
1 1/2	1.900	11 1/2			0.145	2.73			0.200	3.63	0.400	6.41
2	2.375	11 1/2			0.154	3.68			0.218	5.02	0.436	9.03
2 1/2	2.875	8			0.203	5.82			0.276	7.66	0.552	13.70
3	3.500	8			0.216	7.62			0.300	10.25	0.600	18.58
3 1/2	4.000	8			0.226	9.20			0.318	12.51	0.636	22.85
4	4.500	8			0.237	10.89			0.337	14.98	0.674	27.54
5	5.563	8			0.258	14.81			0.375	20.78	0.750	38.55
6	6.625	8			0.280	19.19			0.432	28.57	0.864	53.16
8	8.625	8	0.277	25.00	0.322	28.81			0.500	43.39	0.875	72.42
10[c]	10.750	8	0.307	35.00	0.365	41.13	0.500	54.74				
12[d]	12.750	8	0.330	45.00	0.375	50.71	0.500[d]	65.41				

From Standard Specifications for Welded and Seamless Steel Pipe of the *American Society for Testing Materials, A.S.T.M.* Designation A120

[a]Sizes larger than those shown in the table are measured by their outside diameter, such as 14 in. outside diameter, etc. These larger sizes will be furnished with plain ends, unless otherwise specified. The weights will correspond to the manufacturers' published standards although it is possible to calculate the theoretical weights for any given size and wall thickness on the basis of 1 cu in of steel weighing 0.2833 lb.

[b]The American Standard for Wrought-Iron and Wrought-Steel Pipe *A.S.A.* B36.10-1939 has assigned no schedule number to *Double Extra-Strong* pipe

[c]A 10 in Standard Weight pipe is also available with 0.279 in. wall thickness, but this wall is not covered by a Schedule Number

[d]Owing to a departure from the *Standard-Weight* and *Extra-Strong* wall thicknesses for the 12 in nominal size, Schedules 40 and 60, Table 2 of the *A.S.A.* B36.10-1939, Standard for Wrought-Iron and Wrought-Steel Pipe, the regular *Standard* and *Extra-Strong* wall thicknesses (0.375 in. and 0.500 in.) have been substituted.

as to afford interchangeability in threading, and other elements associated with fabrication and erection. Hence the difference in wall thickness is accompanied by a corresponding change in inside diameter. In sizes up to 14 in., pipe is designated by its nominal size which corresponds roughly to the inside diameter of Schedule 40 pipe. In sizes 14 in. and upward, pipe is designated by its outside diameter (O. D.), and the wall thickness is specified.

While the demands for pipe for the heating and ventilating industry are reasonably well served by Schedule 40 (standard weight) pipe, the erection of pipe by welding sometimes warrants using lighter wall thicknesses. The considerations governing pipe wall thickness and its relation to joint design are covered in the *American Standard Code for Pressure Piping, A.S.A.* B31.1-1942, see Section 122. Standard schedules of pipe thick-

nesses are contained in the *American Standard for Wrought-Iron and Wrought-Steel Pipe, A.S.A.* B36.10, which includes standard-weight and extra-strong thicknesses in Schedules 40 and 80, respectively, and eight other schedules of varying wall thickness to provide for different service conditions. Dimensions and other useful data for Schedules 30 and 40 pipe are given in Table 1. Table 2 from *A.S.T.M.* Specifications A53 and A120 combines the schedule thicknesses of *A.S.A.* B36.10 and the old series designations.

Standard-weight pipe is generally furnished with threaded ends in

TABLE 3. STANDARD DIMENSIONS AND WEIGHTS, AND TOLERANCES IN DIAMETER AND WALL THICKNESS FOR COPPER WATER TUBES[a]

(All tolerances in this table are plus and minus except as otherwise indicated)

Standard Water Tube Size, In	Actual Outside Diameter In.	Average Outside Diameter Tolerance, In		Wall Thickness In, Type K		Type L		Type M		Theoretical Weight, Lb per Ft		
		Annealed	Drawn Temper	Nominal	Tolerance	Nominal	Tolerance	Nominal	Tolerance	Type K	Type L	Type M
⅛	0.250	0 002	0 001	0 032	0 003	0 025	0 0025	0 025	0 0025	0 085	0 068	0 068
¼	0 375	0 002	0 001	0 032	0 004	0 030	0 0035	0 025	0 0025	0 134	0 126	0 107
⅜	0 500	0 0025	0 001	0 049	0.004	0 035	0 0035	0 025	0 0025	0 269	0 198	0 145
½	0.625	0 0025	0 001	0 049	0 004	0 040	0 0035	0 028	0 0025	0 344	0.285	0 204
⅝	0.750	0 0025	0 001	0 049	0 004	0 042	0 0035	0 030	0 0025	0 418	0 362	0 263
¾	0.875	0 003	0 001	0 065	0 0045	0 045	0 004	0 032	0 003	0 641	0 455	0 328
1	1.125	0 0035	0 0015	0 065	0 0045	0 050	0 004	0 035	0 0035	0 839	0 655	0 465
1¼	1.375	0.004	0 0015	0 065	0 0045	0 055	0 0045	0 042	0 0035	1.04	0 884	0 682
1½	1 625	0 0045	0 002	0 072	0 005	0 060	0 0045	0 049	0 004	1 36	1.14	0 940
2	2.125	0 005	0 002	0 083	0 007	0 070	0 006	0 058	0 006	2 06	1 75	1 46
2½	2 625	0 005	0 002	0 095	0 007	0 080	0 006	0 065	0 006	2 93	2 48	2 03
3	3.125	0 005	0 002	0 109	0 007	0 090	0 007	0 072	0 006	4 00	3 33	2 68
3½	3 625	0.005	0 002	0 120	0 008	0 100	0 007	0 083	0 007	5 12	4 29	3 58
4	4 125	0 005	0 002	0 134	0 010	0 110	0 009	0 095	0 009	6 51	5 38	4 66
5	5 125	0 005	0 002	0 160	0 010	0 125	0 010	0 109	0 009	9 67	7.61	6 66
6	6.125	0.005	0 002	0 192	0 012	0 140	0 010	0 122	0 010	13 9	10 2	8 92
8	8.125	0 006	+0 002 −0 004	0 271	0 016	0 200	0 014	0 170	0 014	25 9	19 3	16 5
10	10.125	0 008	+0 002 −0 006	0 338	0 018	0 250	0 016	0 212	0.015	40 3	30 1	25 6
12	12 125	0.008	+0.002 −0 006	0 405	0 020	0 280	0 018	0 254	0 016	57 8	40 4	36 7

[a]From Standard Specifications for Copper Water Tube of the *American Society for Testing Materials, A.S.T.M.* Designation B88-41

NOTE 1.—For copper gas and oil burner tubes, the tolerances shown above for various wall thicknesses (type K) apply irrespective of diameter

NOTE 2.—For tubes other than round no standard tolerances are established These tolerances do not apply to condenser and heat exchanger tubes.

random lengths of 16 to 22 ft, although when ordered with plain ends, 5 per cent may be in lengths of 12 to 16 ft. Five per cent of the total number of lengths ordered may be *jointers* which are two pieces coupled together. Extra-strong pipe is generally furnished with plain ends in random lengths of 12 to 22 ft, although 5 per cent may be in lengths of 6 to 12 ft.

In addition to *IPS* copper pipe, several varieties of copper tubing are in use with either flared or compression couplings or soldered joints. Dimensions of copper water tubing intended for plumbing, underground water service, fuel-oil lines, gas lines, etc., have been standardized by the U. S.

Government and the *American Society for Testing Materials.* There are three standard wall-thickness schedules of copper water tubing classified in accordance with their principal uses as follows:

Type *K*—Designed for underground services and general plumbing service.
Type *L*—Designed for general plumbing purposes.
Type *M*—Designed for use with soldered fittings only.

In general, Type *K* is used where corrosion conditions are severe, and Types *L* and *M* where such conditions may be considered normal as, for instance, in heating work. Types *K* and *L* are available in both hard and soft tempers; Type *M* is available only in hard temper. Where flexibility is essential as in hidden replacement work or where as few joints as possible are desired as in fuel-oil lines, the soft temper is commonly used. New or

TABLE 4. THERMAL EXPANSION OF PIPE IN INCHES PER 100 FT[a]
(For superheated steam and other fluids refer to temperature column)

Saturated Steam			Elongation in Inches per 100 ft from −20 F up				Saturated Steam		Elongation in Inches per 100 ft from −20 F up			
Vacuum Inches of Hg	Pressure Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought-Iron Pipe	Copper Pipe	Pressure Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought-Iron Pipe	Copper Pipe
		−20	0	0	0	0	2.5	220	1.634	1.852	1.936	2.720
		0	0.127	0.145	0.152	0.204	10.3	240	1.780	2.020	2.110	2.960
		20	0.255	0.293	0.306	0.442	20.7	260	1.931	2.183	2.279	3.189
		40	0.390	0.430	0.465	0.655	34.5	280	2.085	2.350	2.465	3.422
29.39	....	60	0.518	0.593	0.620	0.888	52.3	300	2.233	2.519	2.630	3.665
28.89	....	80	0.649	0.725	0.780	1.100	74.9	320	2.395	2.690	2.800	3.900
27.99	....	100	0.787	0.898	0.939	1.338	103.3	340	2.543	2.862	2.988	4.145
26.48	....	120	0.926	1.055	1.110	1.570	138.3	360	2.700	3.029	3.175	4.380
24.04	....	140	1.051	1.209	1.265	1.794	180.9	380	2.859	3.211	3.350	4.628
20.27	....	160	1.200	1.368	1.427	2.008	232.4	400	3.008	3.375	3.521	4.870
14.63	....	180	1.345	1.528	1.597	2.255	293.7	420	3.182	3.566	3.720	5.118
6.45	....	200	1.495	1.691	1.778	2.500	366.1	440	3.345	3.740	3.900	5.358

[a]From *Piping Handbook*, by Walker and Crocker. This table gives the expansion from −20 F to the temperature in question. To obtain the amount of expansion between any two temperatures take the difference between the figures in the table for those temperatures. For example, if a steel pipe is installed at a temperature of 60 F and is to operate at 300 F, the expansion would be 2.519 − 0.593 = 1.926 in.

exposed work generally employs copper pipe of a hard temper. All three classes are extensively used with soldered fittings.

Standard dimensions, weights, and diameter and wall-thickness tolerances for these classes of copper tubing are given in Table 3. Copper pipe is also available with dimensions of steel pipe.

Refrigeration lines used in connection with air conditioning equipment also employ copper tubing extensively. For refrigeration use where tubing absolutely free from scale and dirt is required, bright annealed copper tubing that has been deoxidized is used. This tubing is available in a variety of sizes and wall thicknesses.

EXPANSION AND FLEXIBILITY

The increase in temperature of a pipe from room temperature to an operating steam or water temperature 100 F or more above room temperature results in an increase in length of the pipe for which provision must be made. The amount of linear expansion (or contraction in the case of refrigeration lines) per unit length of material per degree change in

temperature is termed the *coefficient of linear expansion* of that material, or commonly, the *coefficient of expansion*. This coefficient varies with the material.

The linear expansion of cast-iron, steel, wrought-iron, and copper pipe, the materials most frequently used in heating and ventilating work, can be determined from Table 4.

The three methods by which the elongation due to thermal expansion may be taken care of are:

1. Expansion joints.
2. Swivel joints.
3. Inherent flexibility of the pipe itself utilized through pipe bends, right-angle turns, or offsets in the line.

Expansion joints of the slip-sleeve, diaphragm, or corrugated types made of copper, rubber, or other gasket material are all used for taking up expansion, but generally only for low pressures or where the inherent flexibility of the pipe cannot readily be used as in underground steam or hot water distribution lines.

Swivel joints are used to some extent in low-pressure steam and hot-

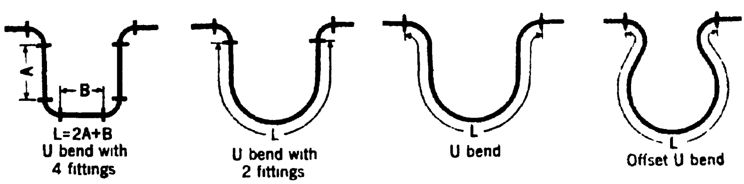


FIG. 1. MEASUREMENT OF L ON VARIOUS PIPE BENDS

water heating systems, and in hot-water supply lines. Since swivel joints permit the expansive movement of the pipe by turning of threaded joints, which may ultimately result in a leak, it is preferable to provide sufficient flexibility without resorting to swiveling in the threads.

Probably the most economical method of providing for expansion of piping in a long run is to take advantage of the directional changes which must necessarily occur in the piping and proportion the offsets so that sufficient flexibility is secured. Ninety-degree bends with long, straight tangents in either a horizontal or a vertical plane are an excellent means for securing adequate flexibility with larger sizes of pipe. When flexibility cannot be obtained in this manner, it is necessary to make use of some type of expansion bend. The exact calculation of the size of expansion bends required to take up a given amount of thermal expansion is relatively complicated[1]. The following approximate method, however, has been found to give reasonably good results and is deemed to be sufficiently accurate for most heating work.

Fig. 1 shows several types of expansion bends commonly used for taking up thermal expansion. The amount of pipe, L, required in each of these bends may be computed from Equation 1.

$$L = 6.16 \sqrt{D \Delta} \tag{1}$$

[1]See (1) Piping Handbook, by Walker and Crocker (McGraw-Hill Co); (2) A Manual for The Design of Piping for Flexibility by the Use of Graphs, by E. A. Wert, S. Smith, E. T. Cope, published by The Detroit Edison Company.

where

L = length of pipe, feet.
D = outside diameter of the pipe used, inches.
Δ = the amount of expansion to be taken up, inches

This formula, based on the use of mild-steel pipe with wall thicknesses not heavier than extra-strong, assumes a maximum safe value of fiber stress of 16,000 psi. When square type bends are used, the width of the bend should not exceed about twice the height, since for a given total length of pipe in the bend, the height of the bend becomes progressively less with increase in width until the height approaches zero and no flexibility exists. Actually, wide bends utilize to best advantage the inherent flexibility of the line, but such bends cannot be proportioned on the basis of Equation 1. For such applications, more accurate methods[2] should be employed. It is further assumed that the corners are made with screwed or flanged elbows or with arcs of circles having radii five to six times the pipe diameter. Use of welding elbows with radii of 1½ times the pipe diameter will decrease the end thrusts somewhat but will raise the fiber stress correspondingly.

All risers must be anchored and safeguarded so that the difference in length when hot from the length when cold shall not disarrange the normal and orderly provisions for drainage of the branches.

Proper anchoring of piping is especially necessary with light-weight radiators, to allow for freedom of expansion in order that no pipe strain will distort the radiators. When expansion strains from the pipes are permitted to reach these light metal heaters they usually emit disturbing sounds.

HANGERS AND SUPPORTS

Heating system piping requires careful and substantial support. Where changes in temperature of the line are not large, such simple methods of support may be utilized as hanging the line by means of rods or perforated strip from the building structure, or supporting it by brackets or on piers.

When fluids are conveyed at temperatures of 150 F or above, however, hangers or supporting equipment must be fabricated and assembled to permit free expansion or contraction of the piping. This can be accomplished by the use of long rod hangers, spring hangers, chains, hangers or supports fitted with rollers, machined blocks, elliptical or circular rings of larger diameter than the pipe giving contact only at the bottom, or trolley hangers. In all cases, allowance should be made for rod clearance to permit swinging without setting up severe bending action in the rods.

For pipes of small size, perforated metal strip is often used. For horizontal mains, the rod or strip usually is attached to the joists or steel work of the floor above. For long runs of vertical pipe subject to considerable thermal expansion, either the hangers should be designed to prevent excessive load on the bottom support when expansion takes place, or the bottom support should be designed to withstand the entire load.

THREADING PRACTICE

In all threaded pipe for heating and ventilating installations the American Standard taper pipe thread, *A.S.A.* B2.1-1942 is used. This thread is cut with a taper of 1 in 16 measured on the diameter of the pipe

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[2]Loc Cit. Note 1

so as to secure a tight joint. The number of threads per inch varies with the pipe size. Threads for fittings are the same, except that it is regular practice to furnish straight tapped couplings for Schedule 40 pipe 2 in. and smaller. For steam pressures in excess of 25 psi, it is recommended that taper-tapped couplings be used to obtain a tight joint. These may be secured by ordering *line* pipe[3] which is used for oil piping, the couplings of which are provided with taper-tapped threads and may be used with regular mill-threaded standard weight pipe. Thread lengths should be in accordance with *A.S.A.* B2.1. Right-hand threads are used unless otherwise ordered. To facilitate drainage, some elbows have the thread tapped at an angle to provide a pitch of the connecting pipe of ¼ in. to the foot. These elbows are known to the trade as pitch elbows and are

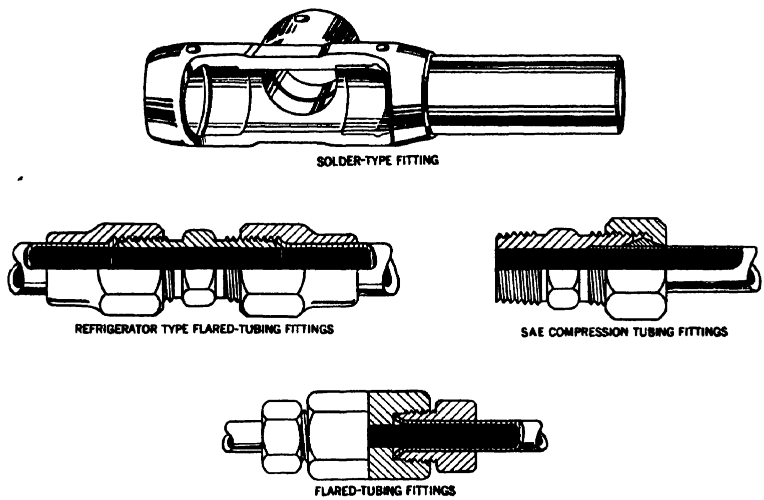


Fig. 2. Copper or Brass Tubing Fittings

commercially available. All threaded pipe joints should be made up with a thread paste suitable for the service for which the pipe is to be used.

TYPES OF FITTINGS

Fittings for joining the separate lengths of pipe together are made in a variety of forms, and are either screwed or flanged, the former being generally used for the smaller sizes of pipe up to and including 3½ in., and the latter for the larger sizes, 4 in. and above. Screwed fittings of large size as well as flanged fittings of small size are also made and are used for certain classes of work at the proper pressure.

The material used for fittings is generally cast-iron, but in addition to this, malleable-iron, steel and steel alloys are also used, as well as various grades of brass or bronze. The material to be used depends on the character of the service and the pressure. Malleable iron fittings, like brass fittings, are cast with a round instead of a flat band or bead, or with no bead at all. Fittings are designated as male or female, depending on whether the threads are on the outside or inside, respectively. Screwed galvanized fittings are made according to the 150 lb American Standard.

[3]See *API Specification* 5L *for Line Pipe, American Petroleum Institute.*

As in the case of pipe, several weights of fittings are manufactured. Recognized American Standards for the various weights are as follows:

Cast-iron pipe flanges and flanged fittings for 25 lb (sizes 4 in. and larger), 125 lb, and 250 lb maximum saturated steam pressure, *A.S.A.* B16b2, B16a, and B16b respectively.

Malleable iron screwed fittings for 150 lb maximum saturated steam pressure, *A.S.A.* B16c.

Cast-iron screwed fittings for 125 and 250 lb maximum saturated steam pressure, *A.S.A.* B16d.

Steel flanged fittings for 150 and 300 lb maximum steam service pressure, *A.S.A.* B16e.

The allowable cold water working pressures for these standards vary from 43 lb for the 25 lb standard to 500 lb for the 300 lb steel standard.

War standard ratings in effect for the duration of the emergency now permit higher ratings for certain sizes of the 125 lb cast-iron flanged fitting standard, and for 300 lb steel flanges and flanged fittings than are shown in the regular American Standards mentioned previously.

Screwed fittings include: nipples or short pieces of pipe of varying lengths; couplings of steel or wrought-iron; elbows for turning angles of either 45 deg or 90 deg; return bends, which may be of either the close or open pattern, and may be cast with either a back or side outlet; tees; crosses; laterals or *Y* branches; and a variety of plugs, bushings, caps, lock-nuts, flanges and reducing fittings. Reducing fittings as well as bushings, both of which are used in changing from one pipe size to another, may have the smaller connection tapped eccentrically to permit free drainage of the water of condensation in steam lines or free escape of air in water lines.

Fittings for copper tubing are available in the soldered, flared, or compression types. Illustrations of each of these types are shown in Fig. 2. Fittings for copper pipe of *IPS* dimensions are available in screwed or soldered types of connection. Table 5 from *A.S.A.* Standard A40.3-1941 contain dimensions for soldered joint elbows, tees, crosses, and 45 deg elbows.

The compression type fitting is generally limited to smaller size tubing while the flared and soldered types are used in both large and small sizes. An American Standard, *A.S.A.* A40.2-1936 has been prepared to standardize dimensions for brass fittings for flared copper water tubes. Flared tube fittings are widely used in refrigerating work where *S.A.E.* dimensions and a 45-deg flare render most fittings interchangeable, although for refrigeration use, thread fits and tolerances on thread gages must be maintained within close limits. Brass fittings with *S.A.E.* dimensions are not interchangeable with the American Standard fittings for water tubes.

Ammonia pipe fittings made of cast-iron were formerly used extensively in handling refrigerants in large installations. Replacement of ammonia by other refrigerants operating at lower pressures has seriously curtailed the market for these fittings. For this reason formulation of an American Standard for these fittings was abandoned by the *A.S.A.* in 1936.

FLANGE FACINGS AND GASKETS

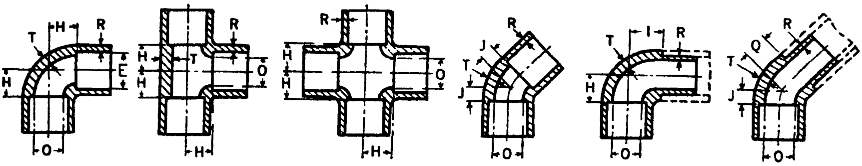
A number of different flange facings in common use are plain face, raised face, tongue and groove, and male and female. Cast-iron fittings for 125 psi and below are normally furnished with a plain face, while the 250 lb cast-iron fittings are supplied with a 1/16-in. raised face. The standard facing for steel flanged fittings for 150 and 300 psi is a 1/16-in. raised face although these fittings are obtainable with a variety of facings. The gasket surface of the raised face may be finished smooth

or may be machined with concentric or spiral grooves often referred to as serrated face or phonograph finish, respectively.

The dimensions of elbows, tees and crosses for 125 lb cast-iron screwed fittings are given in Table 6, whereas the dimensions for 125 lb cast-iron flanged fittings are given in Table 7.

For low temperature service not to exceed about 220 F, a number of paper or vegetable fiber gasket materials will prove satisfactory; for plain

TABLE 5. AMERICAN STANDARD DIMENSIONS OF ELBOWS, TEES, CROSSES, AND 45 DEG ELBOWS, SOLDERED-JOINT FITTINGS, *A.S.A.* A40.3-1941



Nominal Size[a]	Cast Brass[b]							Wrought Metal	Bore of Fittings
	Laying Length, Tee, Ell, and Cross[b]	Laying Length, Ell With External Shoulder	Laying Length, 45 Deg Ell	Laying Length, 45 Deg Ell External Shoulder	Inside Diameter of Fittings,[c] Min	Metal Thickness[d]		Metal Thickness[e] Min [f]	
	H	I	J	Q	O	T	R	T and R	E, Min
1/4	1/4	3/8	3/16	1/4	0.31	0.08	0.048	0.030	0.378
3/8	5/16	7/16	3/16	5/16	0.43	0.08	0.048	0.035	0.503
1/2	7/16	9/16	3/16	5/16	0.54	0.09	0.054	0.040	0.628
3/4	9/16	11/16	1/4	3/8	0.78	0.10	0.060	0.045	0.878
1	3/4	7/8	5/16	7/16	1.02	0.11	0.066	0.050	1.1285
1 1/4	7/8	1	7/16	9/16	1.26	0.12	0.072	0.055	1.3785
1 1/2	1	1 1/8	1/2	5/8	1.50	0.13	0.078	0.060	1.629
2	1 1/4	1 3/8	9/16	3/4	1.98	0.15	0.090	0.070	2.129
2 1/2	1 1/2	1 5/8	5/8	7/8	2.46	0.17	0.102	0.080	2.629
3	1 3/4	1 7/8	3/4	1	2.94	0.19	0.114	0.090	3.129
3 1/2	2	2 1/8	7/8	1 1/8	3.42	0.20	0.120	0.100	3.629
4	2 1/4	2 3/8	15/16	1 1/4	3.90	0.22	0.132	0.110	4.129
5	3 1/8	......	1 7/16	......	4.87	0.28	0.168	0.125	5.129
6	3 5/8	......	1 5/8	......	5.84	0.34	0.204	0.140	6.129

All dimensions given in inches

[a]This size is the nominal bore of the tube

[b]These dimensions may be used for wrought-metal fittings as well as for cast-brass fittings at manufacturer's option.

[c]This dimension is the same as the inside diameter Class L tubing (American Standard Specifications for Copper Water Tube, *A S A*. H23 1-1939 (*A S T M* B88)

[d]Patterns shall be designed to produce body thicknesses given in the table Metal thickness at no point shall be less than 90 per cent of the thicknesses given in the table.

[e]This dimension has the same thickness as Type L tubing

[f]These dimensions are minimum, but in every case the thickness of wrought fittings should be at least as heavy as the tubing with which it is to be used

NOTE 1 —Wrought fittings, as well as cast fittings, must be provided with a shoulder or stop at the bottom end of socket

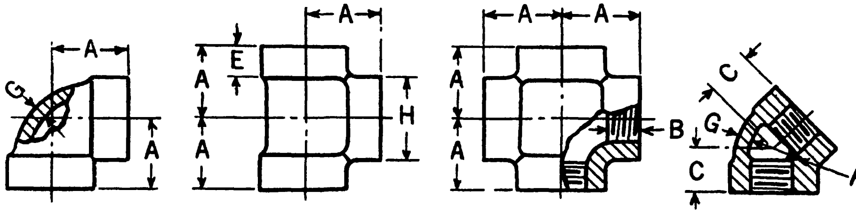
NOTE 2.—Street fittings with male ends are for use in connection with other fittings illustrated

raised face flanges, rubber or rubber inserted gaskets are commonly employed. Asbestos composition gaskets are probably the most widely used, particularly where the temperature exceeds 250 F. Jacketed asbestos and metallic gaskets may be used for any pressure and temperature conditions, but preferably only with a narrow recessed facing.

WELDING

Erection of piping in heating and ventilating installations by means of fusion welding has been commonly accepted in the past few years as an alternate method to the screwed and flanged joint. Since the question of economy of welding as against the use of screwed and flanged fittings

TABLE 6. AMERICAN STANDARD DIMENSIONS OF ELBOWS, 45-DEG ELBOWS, TEES, AND CROSSES (STRAIGHT SIZES) FOR CLASS 125 CAST-IRON SCREWED FITTINGS, *A.S.A.* B16a-1939



Elbow Tee Cross 45 deg elbow

	A	C	B	E	F		G	H
NOMINAL PIPE SIZE	CENTER TO END, ELBOWS, TEES AND CROSSES	CENTER TO END, 45 DEG ELBOWS	LENGTH OF THREAD, MIN	WIDTH OF BAND, MIN	INSIDE DIAMETER OF FITTING		METAL THICKNESS,[a] MIN	OUTSIDE DIAMETER OF BAND, MIN.
					Min	Max		
¼	0.81	0.73	0.32	0.38	0.540	0.584	0.110	0.93
⅜	0.95	0.80	0.36	0.44	0.675	0.719	0.120	1.12
½	1.12	0.88	0.43	0.50	0.840	0.897	0.130	1.34
¾	1.31	0.98	0.50	0.56	1.050	1.107	0.155	1.63
1	1.50	1.12	0.58	0.62	1.315	1.385	0.170	1.95
1¼	1.75	1.29	0.67	0.69	1.660	1.730	0.185	2.39
1½	1.94	1.43	0.70	0.75	1.900	1.970	0.200	2.68
2	2.25	1.68	0.75	0.84	2.375	2.445	0.220	3.28
2½	2.70	1.95	0.92	0.94	2.875	2.975	0.240	3.86
3	3.08	2.17	0.98	1.00	3.500	3.600	0.260	4.62
3½	3.42	2.39	1.03	1.06	4.000	4.100	0.280	5.20
4	3.79	2.61	1.08	1.12	4.500	4.600	0.310	5.79
5	4.50	3.05	1.18	1.18	5.563	5.663	0.380	7.05
6	5.13	3.46	1.28	1.28	6.625	6.725	0.430	8.28
8	6.56	4.28	1.47	1.47	8.625	8.725	0.550	10.63
10	8.08[b]	5.16	1.68	1.68	10.750	10.850	0.690	13.12
12	9.50[b]	5.97	1.88	1.88	12.750	12.850	0.800	15.47

All dimensions given in inches

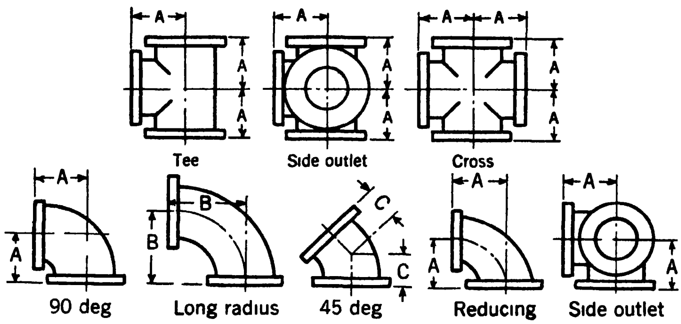
[a]Patterns shall be designed to produce castings of metal thickness given in the table. Metal thickness at no point shall be less than 90 per cent of the thickness given in the table.

[b]Applies to elbows and tees only.

is dependent on the individual job, the use of welding is generally recommended on the basis of a greatly reduced cost of maintenance and repair, of less weight resulting from the use of a lighter-weight pipe, and of increased economy in pipe insulation, hangers, and supports rather than on the basis of any economy that might be effected in actual erection by welding on low to medium pressure heating jobs.

Fusion welding, commonly used in erection of piping, is defined as the process of joining metal parts in the molten, or molten and vapor states, without the application of mechanical pressure or blows. Fusion welding

TABLE 7. AMERICAN STANDARD DIMENSIONS OF TEES, CROSSES[a] (STRAIGHT SIZES), AND ELBOWS FOR CLASS 125 CAST-IRON FLANGED FITTINGS, *A.S.A.* B16a-1939



Nominal Pipe Size[b-c]	A: Center to Face Tees, Crosses[c-d] and Elbows	AA: Face to Face Tees and Crosses[c-d]	B: Center to Face Long Radius Elbow[e,f,g]	C: Center to Face 45 Deg Elbows[g]	Diameter of Flange	Thickness of Flange, Min	Metal[h] Thickness of Body
1	3½	7	5	1¾	4¼	7/16	5/16
1¼	3¾	7½	5½	2	4⅝	½	5/16
1½	4	8	6	2¼	5	9/16	5/16
2	4½	9	6½	2½	6	⅝	5/16
2½	5	10	7	3	7	11/16	5/16
3	5½	11	7¾	3	7½	¾	⅜
3½	6	12	8½	3½	8½	13/16	7/16
4	6½	13	9	4	9	15/16	½
5	7½	15	10¼	4½	10	15/16	½
6	8	16	11½	5	11	1	9/16
8	9	18	14	5½	13½	1⅛	⅝
10	11	22	16½	6½	16	1 3/16	¾
12	12	24	19	7½	19	1¼	13/16
14 O.D.	14	28	21½	7½	21	1⅜	⅞
16 O.D.	15	30	24	8	23½	1 7/16	1
18 O.D.	16½	33	26½	8½	25	1 9/16	1 1/16
20 O.D.	18	36	29	9½	27½	1 11/16	1⅛
24 O.D.	22	44	34	11	32	1⅞	1¼
30 O.D.	25	50	41½	15	38¾	2⅛	1 7/16
36 O.D.	28	56	49	18	46	2⅜	1⅝
42 O.D.	31	62	56½	21	53	2⅝	1 13/16
48 O.D.	34	68	64	24	59½	2¾	2

All dimensions given in inches.

[a]Crosses both straight and reducing sizes 18 in and larger shall be reinforced to compensate for the inherent weakness in the casting design.

[b]Size of all fittings listed indicates nominal inside diameter of port

[c]Tees, side outlet tees, and crosses, 16 in. and smaller, reducing on the outlet, have the same dimensions center to face, and face to face as straight size fittings corresponding to the size of the larger opening. Sizes 18 in and larger, reducing on the outlet, are made in two lengths, depending on the size of the outlet.

[d]Tees and crosses, reducing on run only, carry same dimensions center to face and face to face as a straight size fitting of the larger opening.

[e]Reducing elbows and side outlet elbows carry same dimensions center to face as straight size elbows corresponding to the size of the larger opening.

[f]Special degree elbows, ranging from 1 to 45 deg, inclusive, have the same center to face dimensions as given for 45-deg elbows and those over 45 deg and up to 90 deg, inclusive, shall have the same center to face dimensions as given for 90-deg elbows The angle designation of an elbow is its deflection from straight line flow and is the angle between the flange faces.

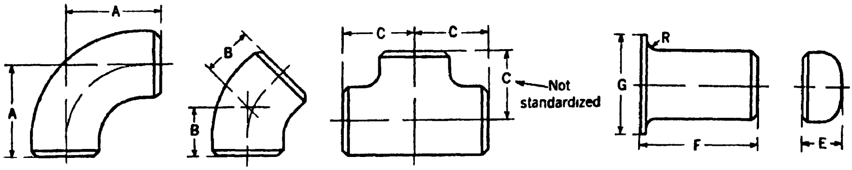
[g]Side outlet elbows shall have all openings on intersecting center-lines.

[h]Body thickness at no point shall be less than 87½ per cent of the dimensions given in the table

embraces gas welding and electric arc welding, both of which are commonly used to produce acceptable welds.

Welding application requires the same basic knowledge of design as do the other types of assembly, but in addition, requires a generous knowledge of the sciences involved, particularly as to welding qualities of metal, their reaction to extremely high temperatures, and the ability to determine and use only the best quality welding rods. This requirement applies equally to employer and employee with the employer accepting

TABLE 8 AMERICAN STANDARD DIMENSIONS FOR BUTT-WELDING ELBOWS, TEES, CAPS, AND LAPPED-JOINT STUB ENDS, *A.S.A.* B16.9-1940



Nominal Pipe Size	Outside Diameter	Center-to-End			Caps E[b c]	Lapped-Joint Stub Ends		
		90-Deg Elbows *A*	45-Deg Elbows *B*	Of Run Tee *C*[a]		Length *F*[b]	Radius of Fillet *R*	Diam of Lap *G*[d]
1	1.315	1½	⅞	1½	1½	4	⅛	2
1¼	1 660	1⅞	1	1⅞	1½	4	3/16	2½
1½	1.900	2¼	1⅛	2¼	1½	4	¼	2⅞
2	2.375	3	1⅜	2½	1½	6	5/16	3⅝
2½	2.875	3¾	1¾	3	1½	6	5/16	4⅛
3	3.500	4½	2	3⅜	2	6	⅜	5
3½	4 000	5¼	2¼	3¾	2½	6	⅜	5½
4	4 500	6	2½	4⅛	2½	6	7/16	6 3/16
5	5.563	7½	3⅛	4⅞	3	8	7/16	7 5/16
6	6 625	9	3¾	5⅝	3½	8	½	8½
8	8 625	12	5	7	4	8	½	10⅝
10	10.750	15	6¼	8½	5	10	½	12¾
12	12.750	18	7½	10	6	10	½	15

All dimensions given in inches.

[a]The dimensions of welding tees cover those which have side outlets from one size less than half the size of the run-way opening of the tees to full size

[b]Dimensions *E* and *F* are applicable only to these fittings in schedules up to and including Schedule 80, *A.S A.* Standard B36 10-1939

[c]The shape of these caps shall be ellipsoidal and shall conform to the requirements of the *A.S.M.E.* Boiler Construction Code.

[d]This dimension is for standard machined facings in accordance with American Standard for Steel Pipe Flanges and Flanged Fittings (*A S A.* B16e-1939). The back face of the lap shall be machined to conform to the surface of the flange on which it seats Where ring joint facings are to be applied, use dimension *K* as given in *A S.A* B16e-1939.

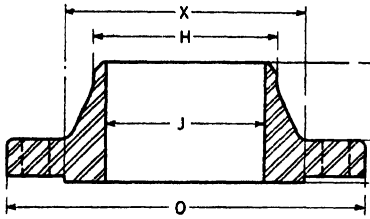
all of the responsibility. Thus the employer should select his welding mechanics with good judgment, provide them with first-class equipment and tools, arrange for their training and use of acceptable workmanship standards, and at regular intervals subject their work to prescribed tests. Industry will not accept the employment of mechanics of undetermined ability nor on the basis of past experience. Neither does industry accept the statement that a weld is only as good as the workman who makes it.

For piping which is to be operated at pressures in excess of 15 psi, rules for fusion welding of pipe joints and the qualification of welders and welding procedures as contained in the *American Standard Code for Pressure Piping* should be observed. For pressures 15 psi or below, the foregoing rules or those set forth in the *Standard Manual on Pipe Welding*

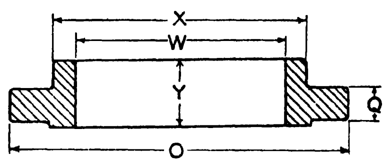
of the *Heating, Piping and Air Conditioning Contractors National Association* should be followed to insure a satisfactory welded joint.

A complete line of manufactured steel welding fittings is now available and a dimensional standard has been prepared under the procedure of the

TABLE 9. AMERICAN STANDARD DIMENSIONS OF STEEL WELDING NECK AND SLIP-ON WELDING FLANGES FOR STEAM SERVICE PRESSURE RATING OF 150 PSI (GAGE) AT A TEMPERATURE OF 500 F, AND 100 PSI (GAGE) AT 750 F, *A.S.A.* B16e-1939



WELDING NECK



SLIP-ON WELDING

NOMINAL PIPE SIZE	DIAMETER OF FLANGE	THICKNESS OF FLG[a] MIN	DIAMETER OF HUB	HUB DIAM BEGINNING OF CHAMFER[b-c]	LENGTH THRU HUB[a]	INSIDE DIAM OF PIPE SCHEDULE 40[c]	BORE OF SLIP-ON FLANGES MIN	DIAM OF BOLT CIRCLE	NO OF BOLTS	SIZE OF BOLTS
	O	*Q*	*X*	*H*	*Y*	*J*	*W*			
½	3½	7/16	1 3/16	0.84	1 7/8	0 62*	0 88	2 3/8	4	½
¾	3 7/8	½	1½	1.05	2 1/16	0 82*	1.09	2¾	4	½
1	4¼	9/16	1 15/16	1.32	2 3/16	1.05*	1 38	3 1/8	4	½
1¼	4 5/8	5/8	2 5/16	1.66	2¼	1.38*	1 72	3½	4	½
1½	5	11/16	2 9/16	1.90	2 7/16	1.61*	1 97	3 7/8	4	½
2	6	¾	3 1/16	2.38	2½	2.07*	2 44	4¾	4	5/8
2½	7	7/8	3 9/16	2.88	2¾	2.47*	2.94	5½	4	5/8
3	7½	15/16	4¼	3 50	2¾	3 07*	3 56	6	4	5/8
3½	8½	15/16	4 13/16	4.00	2 13/16	3.55*	4 06	7	8	5/8
4	9	15/16	5 5/16	4.50	3	4.03*	4 56	7½	8	5/8
5	10	15/16	6 7/16	5.56	3½	5 05*	5 66	8½	8	¾
6	11	1	7 9/16	6.63	3½	6 07*	6.72	9½	8	¾
8	13½	1 1/8	9 11/16	8 63	4	7 98*	8 72	11¾	8	¾
10	16	1 3/16	12	10.75	4	10.02*	10 88	14¼	12	7/8
12	19	1¼	14 3/8	12.75	4½	....	12.88	17	12	7/8
14 O.D.	21	1 3/8	15¾	14.00	5	To Be	14.19	18¾	12	1
16 O.D.	23½	1 7/16	18	16.00	5	Specified	16 19	21¼	16	1
18 O.D.	25	1 9/16	19 7/8	18.00	5½	by	18.19	22¾	16	1 1/8
20 O.D.	27½	1 11/16	22	20.00	5 11/16	Purchaser	20.19	25	20	1 1/8
24 O.D.	32	1 7/8	26 1/8	24 00	6		24.19	29½	20	1¼

All dimensions given in inches

[a]A raised face of 1/16 in is included in *thickness of flange minimum* and in *length through hub*

[b]The outside surface of the welding end of the hub shall be straight or tapered at not more than 6 deg

[c]Dimensions *H* and *J* correspond to the outside and inside diameters of pipe as given in *A.S.A.* B36 10-1939, Schedule 40

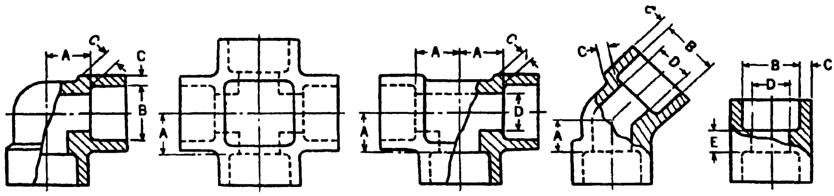
*These diameters are identical with the diameters of what was formerly designated as *Standard Weight Pipe* of the corresponding sizes

American Standards Association to unify heretofore divergent dimensions for the same type welding fittings as produced by different manufacturers. Standard dimensions for steel butt-welding elbows, tees, caps, and lapped-joint stub ends are given in Table 8. Dimensions for eccentric and concentric reducers, and 180-deg return bends are not shown in Table 8 but are included in the American Standard. Larger sizes also are available in some types of fittings. The welding bevel which is a straight 37½-deg V for wall thickness ¾ in. and below, and a U-bevel for thicknesses heavier than ¾ in., conforms to the recommended practice of *A.S.A.*

Standard B16e-1939, American Standard for Steel Pipe Flanges and Flanged Fittings. The latter also contains dimensions for steel welding neck flanges for pressures up to 2500 psi, and slip-on welding flanges for 150 and 300 psi. Table 9 gives these dimensions for welding-neck and slip-on welding flanges suitable for 150 psi gage pressure.

Socket-welding fittings also are commercially available. These fittings have a machined recess for inserting the pipe which is attached by a fillet weld between the pipe wall and socket end. Use of socket-welding fittings generally is restricted to nominal pipe sizes 3 in. and smaller in which range commercial fittings are available. This type of fitting has

TABLE 10. PROPOSED AMERICAN STANDARD DIMENSIONS OF SOCKET-WELDING ELBOWS, TEES, CROSSES, 45-DEG ELBOWS, AND COUPLINGS



Nominal Pipe Size	Minimum Depth of Socket	Center to Bottom of Socket: 90-Deg Ells, Tees, Crosses, Sched 40 & 80	Sched 160	45-Deg Ells, Sched 40 & 80	Sched 160	Couplings, Distance Between Bottom Sockets	Bore Diameter of Socket, Minimum	Minimum Socket Wall Thickness: Sched. 40	Sched 80	Sched 160	Bore Diameter of Fittings: Sched 40	Sched 80	Sched 160
		A		A		E	B	C[a]			D		
1/4	3/8	17/32		5/16		1/4	0.555	0.156	0.156		0.364	0 302	
3/8	3/8	17/32		5/16		1/4	0 690	0 156	0.158		0.493	0 423	
1/2	3/8	5/8	3/4	7/16	1/2	3/8	0.855	0 156	0 184	0 234	0.622	0 546	0 466
3/4	1/2	3/4	7/8	1/2	9/16	3/8	1.065	0 156	0 193	0 273	0.824	0 742	0.614
1	1/2	7/8	1 1/16	9/16	11/16	1/2	1.330	0 166	0.224	0.313	1.049	0.957	0.815
1 1/4	1/2	1 1/16	1 1/4	11/16	13/16	1/2	1 675	0 175	0.239	0 313	1.380	1.278	1 160
1 1/2	1/2	1 1/4	1 1/2	13/16	1	1/2	1.915	0.181	0 250	0.351	1.610	1.500	1.338
2	5/8	1 1/2	1 5/8	1	1 1/8	3/4	2 406	0.193	0.273	0 429	2.067	1.939	1.689
2 1/2	5/8	1 5/8	2 1/4	1 1/8	1 1/4	3/4	2.906	0.254	0 345	0.469	2 469	2 323	2.125
3	5/8	2 1/4	2 1/2	1 1/4	1 3/8	3/4	3.535	0.270	0.375	0 546	3.068	2.900	2.626

All dimensions are given in inches

[a]Dimension C is 1 1/4 times the nominal pipe thickness, minimum, but not less than 5/32 in.

Reducing sizes have same center to bottom of socket dimension as the largest size of reducing fitting.

gained rapid acceptance owing to its ease of installation, low cost, and ability to make a pressure tight joint without weakening the pipe as is the case with threading. Dimensions for socket-welding fittings as offered by most manufacturers of the product are given in Table 10.

VALVES

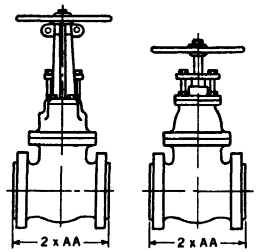
Valves are made with both threaded and flanged ends for screwed and bolted connections just as are pipe fittings.

The material used for valves of small size is generally brass or bronze for low pressures and forged steel for high pressures, while in the larger sizes either cast-iron, cast-steel or some of the steel alloys are employed. Practically all iron or steel valves intended for steam or water work are bronze-mounted or trimmed.

Brass, bronze, and iron valves are generally designed for standard or extra heavy service, the former being used up to 125 lb and the latter up

to 250 lb saturated steam working pressure, although most manufacturers also make valves for medium pressure up to 175 lb steam working pressure. The more common types are gate valves or straightway valves, globe valves, angle valves, check valves and automatic valves, such as reducing and back-pressure valves.

TABLE 11. AMERICAN STANDARD CONTACT SURFACE TO CONTACT SURFACE DIMENSIONS OF CAST-IRON AND STEEL FLANGED WEDGE GATE VALVES, *A.S.A.* B16.10-1939



Nominal Pipe Size	Contact Surface to Contact Surface Dimensions, (2 × AA)				
	Cast-Iron[a]			Steel	
	125	175[bc]	250[b]	150[b]	300[b]
1	....	....	....	....	....
1¼	....	....	....	....	....
1½	....	....	....	....	7½
2	7	7¼	8½	7	8½
2½	7½	8	9½	7½	9½
3	8	9¼	11⅛	8	11⅛
3½	8½	10	11⅞	8½	11⅞
4	9	10½	12	9	12
5	10	11½	15	10	15
6	10½	13	15⅞	10½	15⅞
8	11½	14¼	16½	11½	16½
10	13	16¾	18	13	18
12	14	17½	19¾	14	19¾
14 O.D.	15	....	22½	15	30
16 O.D.	16	....	24	16	33
18 O.D.	17	....	26	17	36
20 O.D.	18	....	28	18	39
24 O.D.	20	....	31	20	45

All dimensions given in inches.
[a]These dimensions are the same for Cast-Iron Double Disc Flanged Gate Valves.
[b]These are pressure designations which refer to the primary service ratings in pounds per square inch of the connecting end flanges.
[c]The connecting end flanges of 175 lb valves are the same as those on 250 lb valves.
NOTE 1:—Where dimensions are not given, the sizes either are not made or there is insufficient demand to warrant the expense of unification
NOTE 2:—Female and groove joint facings have bottom of groove in same plane as *flange edge*, and center to contact surface dimensions for these facings are reduced by the amount of the raised face

Gate valves are the most frequently used of all valves since in their open position the resistance to flow is a minimum, but they should not be used where it is desired to throttle the flow; globe valves should be used for this purpose. These valves may be secured with either a rising or a non-rising stem, although in the smaller sizes the rising stem is more commonly used. The rising stem valve is desirable because the positions of the handle and stem indicate whether the valve is open or closed, although space limitations may prevent its use. The globe valve is less expensive

to manufacture than the gate valve, but its peculiar construction offers a high resistance to flow and may prevent complete drainage of the pipe line. These objections are of particular importance in heating work.

An American Standard, *A.S.A.* B16.10-1939, has been prepared giving the face-to-face dimensions of ferrous flanged and welding-end valves. The following types are covered: wedge gate, double disc gate, globe and angle, and swing check. One purpose of establishing these dimensions is to insure that gate valves of a given rating and flange dimension of either the wedge or double disc design will be interchangeable in a pipe line. Contact surface to contact surface dimensions of cast-iron and steel flanged-wedge-gate valves are given in Table 11. End-to-end dimensions for steel butt-welding valves in sizes up to 8 in., inclusive, are the same as those given in Table 11 for steel valves.

Check valves are automatic in operation and permit flow in only one direction, depending for operation on the difference in pressure between the two sides of the valve. The two principal kinds of check valves are the swing check in which a flapper is hinged to swing back and forth, and the lift check in which a dead weight disc moves vertically from its seat.

Valves commonly used for controlling steam or water supply to radiators constitute a special class since they are manufactured to meet heating system requirements. These valves are generally of the angle type and are usually made of brass. Graduations on the heads or lever handles are often supplied to indicate the relative opening of the valve.

Automatic control of steam supply to individual radiators can be effected by use of direct-acting radiator valves having a thermostatic element at the valve, or near to it. The direct-acting valve is usually an angle-type valve containing a thermostatic element which permits the flow of steam in accordance with room temperature requirements. These valves usually are capable of adjustment to permit variation in room temperature to suit individual taste.

Ordinary steam valves may be used for hot water service by drilling a $\frac{1}{16}$-in. hole through the web forming the seat to insure sufficient circulation to prevent freezing when the valve is closed. Valves made for use in hot water heating systems are of simpler design, one type consisting of a simple butterfly valve, and another of a quick opening type in which a part in the valve mechanism matches up with an opening in the valve body.

In one-pipe steam-heating systems, automatic air valves are required at the radiators. Two common types of air valves available are the vacuum type and the straight-pressure type. Vacuum valves permit the expulsion of air from the radiators when the steam pressure rises and, in addition, act as checks to prevent the return of air into the radiator when a vacuum is formed by the condensation of steam after the supply pressure has dropped. Ordinary air valves permit the expulsion of air from the radiator when steam is supplied under pressure, but when a vacuum tends to be formed the air is drawn back into the radiator.

CORROSION

Corrosion is sometimes encountered in heating work on the outside of buried pipes or the inside of steam heating systems; it is seldom experienced in hot water heating systems unless the water is frequently renewed. Piping buried in the ground is quite successfully protected by coatings of the asphaltic type which are usually applied hot and often reinforced with fabric wrappings. Galvanizing by the hot-dip process and painting with specially prepared mixtures also afford some protection.

Internal corrosion[4] in steam heating systems occurs principally in the condensate return pipes and is nearly always caused by oxygen or carbon dioxide, or both, in solution in the condensate. Oxygen may enter the heating system with the steam, owing to its presence in the boiler-feed water, or it may enter as air through small leaks, particularly in systems which operate at sub-atmospheric pressures. When a steam heating system is operated intermittently, air rushes in during each shutdown period and oxygen is absorbed by the condensate which clings to the interior surfaces of the pipes and radiators. The rate of corrosion depends upon the amounts of oxygen and carbon dioxide present in solution, upon the operating temperature, and upon the length of time that the pipe surfaces are in contact with gas-laden condensate.

Another possible cause of corrosion is a flow of electric current sometimes resulting from faulty electrical circuits which should be corrected. Electrolytic corrosion also may occur because of the presence of two dissimilar metals, such as brass and iron, but the condensate in practically all steam heating systems is such a weak electrolyte that this cause of corrosion is very infrequent.

If trouble is experienced from corrosion, oxygen should be eliminated from the feed water by proper deaeration with commercial apparatus. The elimination of the oxygen due to air leakage is more difficult because of the multitude of small leaks which exist around valve stems and in pipe joints. In vacuum systems, however, an attempt should be made to minimize such leakage.

Carbon dioxide in varying amounts is contained in steam produced from the majority of water supplies. It is formed from the breaking down of carbonates and bicarbonates which are present in nearly all natural waters. It can be partly removed by chemical treatment and deaeration, but there is no simple method whereby it can be entirely eliminated.

These gases cause corrosion only when in solution in the condensate; when they are mixed with dry steam their corrosive effect is negligible. The amount of gas in solution depends upon the partial pressure of that gas in the atmosphere above the surface of the solution, in accordance with the well known physical law of Henry and Dalton[5]. The correct application of this law, however, requires equilibrium conditions which do not always exist under the flow conditions prevailing in a heating system.

There is a distinction between corrosion in heating systems proper and in the condensate discharge lines from other apparatus using steam at relatively high rates, particularly at the times of the cycle when the steam consumption is at its heaviest. In such equipment the gases tend to accumulate in the steam space and to become dissolved in the condensate in high concentrations, thus greatly increasing the possibilities of corrosion. The condensate will more nearly approach in composition the composition of the steam than will the normal condensate from apparatus such as room radiators, and will, therefore, normally include in solution more contaminants. It is possible that careful venting of such equipment would reduce the amount of contaminants dissolved in the

[4]New Light on Heating System Corrosion, by J. H. Walker (*Heating and Ventilating*, May, 1933) A.S.H.V.E. Research Report No. 983—Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman and G. E. Smedberg, (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 253). A.S.H.V.E Research Report No 1037—Corrosion Studies in Steam Heating Systems, by R R. Seeber, F. A. Rohrman and G. E. Smedberg, (A.S.H.V.E. Transactions, Vol. 42, 1936, p. 263). A.S H.V.E. Research Report No. 1071—Corrosion Studies in Steam Heating Systems, by R. R. Seeber and Margaret R. Holley (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 461). Corrosion in Steam Heating Systems, by L. F. Collins and E. L. Henderson, (*Heating, Piping and Air Conditioning*, September, 1939 to May, 1940).

[5]Some Fundamental Considerations of Corrosion in Steam and Condensate Lines, by R. E. Hall and A. R. Mumford (A.S.H.V.E. Transactions, Vol. 38, 1932, p. 121).

condensate, thus giving less corrosion. There is evidence that the partial pressures of the gases and the possibility of corrosion are much lower in heating systems than in high usage equipment. Hence, corrosion observed in the condensate discharge lines from high usage equipment does not necessarily indicate that equally serious corrosion is taking place in the heating system.

The seriousness of corrosive conditions is best determined by actual measurement rather than by inference from isolated instances of pipe failures. The *National District Heating Association* has perfected a corrosion tester for measuring the inherent corrosiveness of existing conditions. This corrosion tester consists of a frame supporting three coils of wire which are carefully weighed. After the tester has been inserted in the pipe line for a definite length of time, the loss of weight of the coils, referred to an established scale, indicates the relative corrosivensss of the condensate. Accompanying such corrosion measurements, a careful chemical analysis should be made of the condensate, and the findings will serve as a basis for an intelligent study of the problem[6].

There are some indications that after a condensate containing carbon dioxide has dissolved some iron and thereby raised its *pH* value, its corrosive action is greatly reduced and the solution will remain comparatively inactive until admission of oxygen permits the precipitation of the dissolved iron as ferric oxide. The *pH* value of the condensate may be buffered to a fairly high value by the solution of iron and not correspond to the *pH* value to be expected in the unbuffered solution containing the same amount of carbon dioxide.

Although inhibitors of various types have had considerable trial and experimentation and successes have been reported, they require further study. Among these inhibitors are oil, sodium silicate, sodium hydroxide, tannin, and various other organic compounds, some of which release ammonia gas. The possible toxic effects, particularly if the steam is used for contact cooking of food, should not be overlooked.

In view of the fact that corrosion is most frequently found in the return lines from special equipment, which constitute a relatively small part of the total piping in a building, a simple solution of the corrosion problem may be to use non-corroding materials in those certain portions of the piping system, since the higher cost will usually be an unappreciable portion of the total. Brass and copper are undoubtedly less subject to this type of corrosion than the ferrous metals, and considerable attention is now being given to corrosion-resistant linings for ferrous pipe. Cast-iron pipe, sometimes alloyed with other metals, also deserves consideration.

Eighteen ferrous and non-ferrous metals and alloys were recently tested in a large air conditioning installation[7]. Observations were made in the wash water of the dehumidifier and in the air stream beyond the eliminator plates. The corrosion rates of all metals and alloys utilizing a dichromated-treated wash water were extremely low. Localized attack in the form of pitting was found to occur on steel in crevices or under solid accumulations. Just beyond the dehumidifier eliminator plates corrosive conditions were observed to be particularly severe and in such locations non-ferrous metals and alloys and stainless steels were most resistant. Alloy steels were found to be superior to mild steel.

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[6]A Method of Measuring Corrosiveness, by J. H. Walker, *Proceedings, American Society for Testing Materials*, 1940.

[7]A.S.H.V.E. RESEARCH PAPER—Corrosion Tests in a Water-Recirculating Air Conditioning System, by W. Z. Friend (A S.H V E TRANSACTIONS, Vol. 48, 1942, p. 233)

CHAPTER 17

Pipe Insulation

Heat Losses from Bare and Insulated Pipes, Low Temperature Pipe Insulation, Insulation of Pipes to Prevent Freezing, Economical Thickness of Pipe Insulation, Underground Pipe Insulation

THE heat loss from uninsulated pipes may be of considerable magnitude if the temperature of the surrounding medium differs appreciably from that of the fluid conveyed. Losses are increased by rapid motion of the surrounding air or by contact of the pipe with bodies of high conductivity. Careful consideration must, therefore, be given to this factor in a properly designed system and adequate insulation provided, if necessary.

HEAT LOSSES FROM BARE PIPES

Heat losses from horizontal bare steel pipes, based on tests at *Mellon Institute* and calculated from the fundamental radiation and convection equations (Chapter 3), are given in Table 1. Heat losses from horizontal copper tubes and pipes with bright, lacquered and tarnished surfaces, are given in Tables 2, 3 and 4[1].

The area in square feet per linear foot of pipe is given in Table 5 for various standard pipe sizes, and Table 6 for copper tubing, while Table 7 gives the area in square feet of flanges and fittings for various standard pipe sizes. These tables can be used to advantage in estimating the amount of insulation required.

Very often, when pipes are insulated, flanges and fittings are left bare so as to allow for easy access to the fittings in case of repairs. The fact that a pair of 8-in. standard flanges having an area of 2.41 sq ft would lose, at 100 lb steam pressure, an amount of heat equivalent to more than a ton of coal per year shows the necessity for insulating such surfaces.

Examples 1 and *2* show how the annual heat loss from uncovered pipe and its dollar value may be computed from the data in Table 1.

Example 1. Compute the total annual heat loss from 165 ft of 2 in. bare pipe in service 4000 hours per year. The pipe is carrying steam at 10 lb pressure and is exposed to an average air temperature of 70 F.

Solution. The pipe temperature is taken as the steam temperature, which is 239.4 F, obtained by interpolation from Table 8, Chapter 1. The temperature difference between the pipe and air = 239.4 − 70 = 169.4 F. By interpolation of Table 1 between temperature differences of 157.1 and 227.7 F, the heat loss from a 2-in. pipe at a temperature difference of 169.4 F is found to be 1.624 Btu/(hr) (linear ft) (°F). The total annual heat loss from the entire line = 1.624 × 169.4 × 165 (linear feet) × 4000 (hours) = 181,600 *Mb*.

Example 2. Coal costing $11.50 per ton and having a calorific value of 13,000 Btu per pound is being burned in the furnace supplying steam to the pipe line given in the previous example. If the system is operating at an over-all efficiency of 55 per cent, determine the monetary value of the annual heat loss from the line.

Solution. The cost of heat per 1000 *Mb* supplied to the system = 1,000,000 × 11.5 (dollars) ÷ [13,000 (Btu) × 2000 (lb) × 0.55 (efficiency)] = $0.804. The total cost of heat lost per year = 0.804 × 181.6 (thousand *Mb*) = $146.00.

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[1]Heat Loss from Copper Piping, by R H. Heilman (*Heating, Piping and Air Conditioning*, September, 1933, p. 458).

PIPE INSULATIONS

Pipe insulations are of several general forms and are made of various types of material. The most common form is the rigid sectional covering either split longitudinally into halves or cut through on one side and scored on the other, to facilitate assembling on pipes. Preformed materials are supplied in segments for assembly on large pipes. The sectional coverings are generally supplied with a pasted on canvas

TABLE 1. HEAT LOSSES FROM HORIZONTAL BARE STEEL PIPES

Expressed in Btu per (hour) (linear foot) (degree Fahrenheit difference between the pipe and surrounding still air at 70 F)

Nominal Pipe Size (Inches)	Hot Water				Steam		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	299.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.455	0.495	0.546	0.584	0.612	0.706	0.760
¾	0.555	0.605	0.666	0.715	0.748	0.866	0.933
1	0.684	0.743	0.819	0.877	0.919	1.065	1.147
1¼	0.847	0.919	1.014	1.086	1.138	1.324	1.425
1½	0.958	1.041	1.148	1.230	1.288	1.492	1.633
2	1.180	1.281	1.412	1.512	1.578	1.840	1.987
2½	1.400	1.532	1.683	1.796	1.883	2.190	2.363
3	1.680	1.825	2.010	2.153	2.260	2.630	2.840
3½	1.900	2.064	2.221	2.433	2.552	2.974	3.215
4	2.118	2.302	2.534	2.717	2.850	3.320	3.590
5	2.580	2.804	3.084	3.303	3.470	4.050	4.385
6	3.036	3.294	3.626	3.886	4.074	4.765	5.160
8	3.880	4.215	4.638	4.960	5.210	6.100	6.610
10	4.760	5.180	5.680	6.090	6.410	7.490	8.115
12	5.590	6.070	6.670	7.145	7.500	8.800	9.530

TABLE 2. HEAT LOSS FROM HORIZONTAL BARE BRIGHT COPPER PIPE

Expressed in Btu per (hour) (linear foot) (degree Fahrenheit difference between the pipe and surrounding still air at 70 F)

Nominal Pipe Size (Inches)	Hot Water (Type K Copper Tube)				Steam (Standard Pipe Size Pipe)		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.180	0.210	0.218	0.229	0.299	0.338	0.355
¾	0.236	0.275	0.291	0.307	0.357	0.408	0.418
1	0.290	0.338	0.354	0.373	0.440	0.492	0.523
1¼	0.340	0.400	0.418	0.443	0.510	0.571	0.598
1½	0.390	0.463	0.473	0.507	0.598	0.671	0.710
2	0.490	0.525	0.600	0.628	0.719	0.813	0.851
2½	0.580	0.675	0.709	0.750	0.840	0.953	1.008
3	0.680	0.788	0.848	0.871	0.987	1.107	1.165
3½	0.760	0.888	0.946	1.000	1.114	1.235	1.307
4	0.940	1.000	1.045	1.107	1.210	1.361	1.456
4½	....	...	.	...	1.335	1.495	1.488
5	1.020	1.200	1.255	1.320	1.465	1.670	1.755
6	1.160	1.375	1.410	1.500	1.685	1.890	1.942
8	1.460	1.725	1.820	1.890	2.100	2.373	2.510

TABLE 3. HEAT LOSS FROM BRIGHT COPPER PIPE GIVEN ONE THIN COAT OF CLEAR LACQUER

Expressed in Btu per (hour) (linear foot) (degree Fahrenheit difference between the pipe and surrounding still air at 70 F)

Nominal Pipe Size (Inches)	Hot Water (Type K Copper Tube)				Steam (Standard Pipe Size Pipe)		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.240	0.265	0.282	0 307	0.401	0.461	0 478
¾	0.320	0.356	0.373	0.414	0.477	0.571	0.578
1	0.390	0.437	0 463	0.507	0 598	0 681	0.710
1¼	0.470	0.537	0.554	0.614	0.700	0.812	0.840
1½	0.540	0.612	0 645	0.714	0.830	0.966	0.990
2	0.690	0.762	0.818	0.892	1.005	1 164	1.201
2½	0.840	0.937	0 991	1 085	1.178	1.361	1.420
3	0.960	1.025	1.135	1.270	1.400	1.625	1.700
3½	1.100	1.250	1.318	1.442	1.580	1.845	1.905
4	1.241	1.400	1.480	1.556	1.750	2.040	2.130
4½	.....	.....	.....	.....	1.910	2.240	2.350
5	1.480	1.685	1.790	1.965	2.130	2.415	2.610
6	1.700	1.936	2.052	2.272	2.450	2.810	2.990
8	2 200	2 500	2 630	2 854	3.120	3 425	3.730

jacket. Blanket insulations are sometimes used for wrapping large pipes particularly where removal for frequent servicing the pipe is necessary. Fittings and bends are commonly covered with portions of standard pre-formed insulation or, when irregular in contour, with plastic materials known as insulating cements.

Insulation is secured to pipes with staples which are used to bridge the joint between half sections, and with metal pipe covering bands or rings of wire which secure individual sections and effect a junction between

TABLE 4. HEAT LOSS FROM HORIZONTAL TARNISHED COPPER PIPE

Expressed in Btu per (hour) (linear foot) (degree Fahrenheit difference between the pipe and surrounding still air at 70 F)

Nominal Pipe Size (Inches)	Hot Water (Type K Copper Tube)				Steam (Standard Pipe Size Pipe)		
	120 F	150 F	180 F	210 F	227 1 F (5 Lb)	297.7 F (50 Lb)	337 9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157 1 F	227.7 F	267.9 F
½	0.250	0 287	0 300	0.321	0.433	0.500	0 530
¾	0.340	0.381	0 409	0 429	0.533	0.543	0 654
1	0.440	0.475	0 509	0 536	0.636	0.746	0 803
1¼	0.500	0 559	0 618	0 622	0.764	0 878	0 934
1½	0.580	0.656	0 710	0 750	0 904	1 053	1.120
2	0.730	0.825	0 890	0 957	1 101	1 273	1 364
2½	0.880	1 000	1 091	1 143	1 305	1 490	1 605
3	1.040	1.175	1 272	1.343	1 560	1.800	1 940
3½	1.180	1.350	1 454	1 535	1 750	2.020	2.170
4	1.460	1.500	1 635	1 715	1 941	2.240	2 430
4½	.....	..	.	. ..	2 131	2.465	2 650
5	1.600	1.812	1 980	2 071	2.387	2.770	2.990
6	1.840	2.125	2 270	2.430	2 740	3.210	3.440
8	2 400	2 685	2 910	3 110	3.310	4.050	4.370

TABLE 5. EXTERNAL SURFACE PER LINEAR FOOT OF PIPE

Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)
½	0.22	2	0 622	5	1 456
¾	0 275	2½	0 753	6	1.734
1	0 344	3	0 917	8	2 257
1¼	0 435	3½	1 047	10	2.817
1½	0 498	4	1 178	12	3 338

abutting sections. A number of surface finishes are used over pipe insulation depending upon the service encountered and appearance desired. Canvas jackets are most common although asbestos paper or asbestos finishing cements are sometimes employed. Insulation outdoors should be waterproof and is generally protected with an asphalt felt for piping and asphaltic cements for fittings. Insulation on lines carrying cold water, brine, or other cold fluids is carefully finished to obtain adequate sealing against the penetration of water vapor.

The selection of pipe insulation for a particular service condition must be made with full consideration of a number of properties in addition to thermal conductivity. Factors which may be of more importance than the thermal conductivity are: ease of application, fire resistance, heat stability, weathering stability, resistance to damage by physical abuse, and others which may apply to a particular installation. A complete evaluation of pipe insulation cannot be included here. Insulation manufacturers should be consulted in regard to the selection of insulation which is to meet specific requirements

HEAT LOSSES FROM INSULATED PIPES

The conductivities of various materials used for insulating steam and hot water systems are given in Table 8 They are given as functions of the mean temperatures or the arithmetic mean of the inner and outer surface temperatures of the insulations. It should be emphasized that they are the average values obtained from a number of tests made on each type of material, also, that in the use of *conductivity* all variables due to differences in thickness, pipe sizes, and air conditions are eliminated. Individual manufacturer's materials will, of course, vary in conductivity to some extent from these values.

The heat losses through 1, 1½, and 2-in. thick 85 per cent Magnesia type of insulation for temperature differences between the pipe and the surrounding atmosphere up to 280 F are shown in Figs. 1, 2, and 3. Standard thicknesses of 85 per cent Magnesia pipe covering are not exactly 1 in. However, the loss through any given thickness of insulation can be obtained by interpolation. Also, the losses through any of the

TABLE 6. EXTERNAL SURFACE PER LINEAR FOOT OF COPPER TUBING
Outside diameter ⅛ in. greater than nominal size

Tube Size (Inches)	Surface Area (Sq Ft)	Tube Size (Inches)	Surface Area (Sq Ft)	Tube Size (Inches)	Surface Area (Sq Ft)
½	0.164	2	0.556	5	1.342
¾	0 229	2½	0 687	6	1.604
1	0.295	3	0 818	8	2.128
1¼	0.360	3½	0 949	..	.....
1½	0.426	4	1 080	..	.....

TABLE 7. AREAS OF FLANGED FITTINGS, SQUARE FEET[a]

Nominal Pipe Size (Inches)	Flanged Coupling		90 Deg Ell		Long Radius Ell		Tee		Cross	
	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy
1	0.320	0.438	0.795	1.015	0.892	1.083	1.235	1.575	1.622	2.07
1¼	0.383	0.510	0.957	1.098	1.084	1.340	1.481	1.925	1.943	2.53
1½	0.477	0.727	1.174	1.332	1.337	1.874	1.815	2.68	2.38	3.54
2	0.672	0.848	1.65	2.01	1.84	2.16	2.54	3.09	3.32	4.06
2½	0.841	1.107	2.09	2.57	2.32	2.76	3.21	4.05	4.19	5.17
3	0.945	1.484	2.38	3.49	2.68	3.74	3.66	5.33	4.77	6.95
3½	1.122	1.644	2.98	3.96	3.28	4.28	4.48	6.04	5.83	7.89
4	1.344	1.914	3.53	4.64	3.96	4.99	5.41	7.07	7.03	9.24
4½	1.474	2.04	3.95	5.02	4.43	5.46	6.07	7.72	7.87	10.07
5	1.622	2.18	4.44	5.47	5.00	6.02	6.81	8.52	8.82	10.97
6	1.82	2.78	5.13	6.99	5.99	7.76	7.84	10.64	10.08	13.75
8	2.41	3.77	6.98	9.76	8.56	11.09	10.55	14.74	13.44	18.97
10	3.43	5.20	10.18	13.58	12.35	15.60	15.41	20.41	19.58	26.26
12	4.41	6.71	13.08	17.73	16.35	18.76	19.67	26.65	24.87	34.11

[a]Including areas of accompanying flanges bolted to the fitting.

insulations given in Table 8 can be obtained by multiplying the losses obtained from Figs. 1, 2, or 3 by the factors given in Table 9.

Pipes operating at high temperatures are frequently insulated to the best advantage by combining a high temperature insulation near the pipe with a moderate or low temperature insulation around it as an outer layer. By this method an efficient material may be used for each of the two temperature ranges encountered. In calculating the heat loss through such a combination the mean temperature of each layer must be determined along with the thickness of each. This is readily done in three or four calculations performed as a series of approximations in which assumptions of thickness and mean temperature are adjusted as indicated.

The rate of heat loss from a surface maintained at constant temperature is greatly increased by air circulation over the surface. In the case of well-insulated surfaces, the increases in losses due to air velocity are very small as compared with increases from bare surfaces, because of the fact

TABLE 8. THERMAL CONDUCTIVITY (k) OF VARIOUS TYPE PIPE INSULATIONS FOR MEDIUM AND HIGH TEMPERATURE PIPE[a]

Expressed in Btu per (hour) (square foot) (degree Fahrenheit temperature difference per inch)

Types of Insulating Materials	Density Lb/Cu Ft	Temp Range of Accepted Use	Mean Temperature, Deg F				
			100	200	300	400	500
85% Magnesia—Type	13–15	Up to 600 F	0.41	0.45	0.48	0.52	....
Corrugated Asbestos—Type							
4 Ply per 1 in.	11–13	Up to 300 F	0.57	0.68	0.80		....
6 Ply per 1 in	15–17	Up to 300 F	0.51	0.59	0.69		....
8 Ply per 1 in.	18–20	Up to 300 F	0.49	0.57	0.65	..	...
Laminated Asbestos—Type							
(35–40 laminations per 1 in.)	30–35	Up to 700 F	0.39	0.44	0.49	0.54	....
Mineral Wool—Type	10–15	Up to 800 F	0.40	0.45	0.50	0.55	....
Diatomaceous Silica—Type	25–30	Up to 1900 F	0.63	0.66	0.69	0.72	0.75
Brown Asbestos Fiber—Type	13–15	Up to 1200 F	0.34	0.39	0.44	0.49	0.54

[a]Average values from various laboratories for insulating materials of various manufacturers.

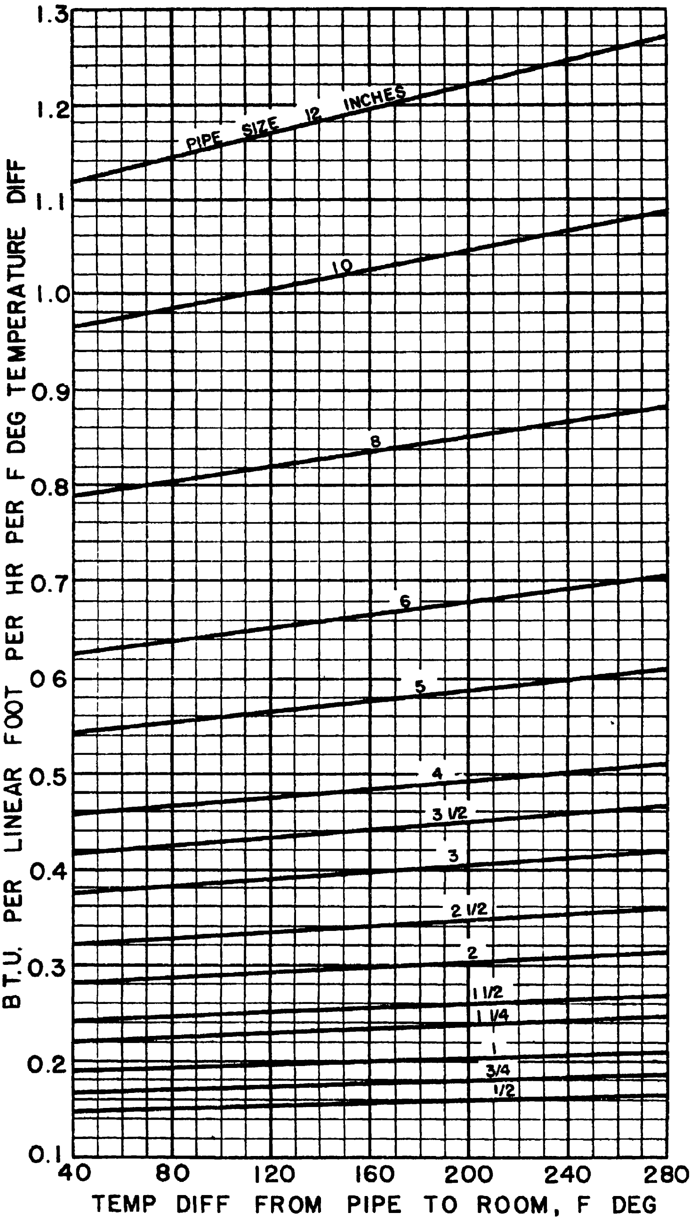


Fig. 1. Heat Loss Through 1 In. Thick 85 per cent Magnesia Type Covering

that air flowing over the surface of the insulation can increase only the conductance of heat from surface to air, and cannot change the internal conductance of the insulation itself. The maximum increase in heat loss due to air velocity ranges from about 15 per cent in the case of 1-in. thick

insulation, to about 5 per cent in the case of 3-in. thick insulation, provided that the insulation is thoroughly sealed so that air can flow only over the surface. If the conditions are such that the air may circulate through cracks and crevices in the insulation, the increases may be far greater than those given. Therefore, it is essential that insulation be

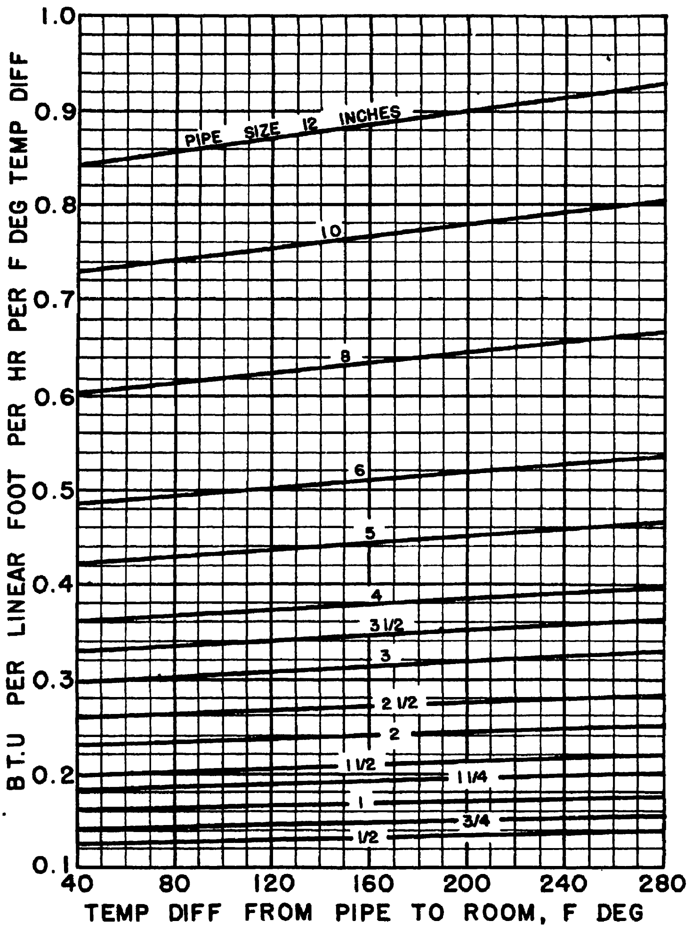


Fig. 2. Heat Loss Through 1½ In. Thick 85 per cent Magnesia Type Covering

applied in such a manner that air circulation within it or between it and the pipe is avoided.

The transfer of heat from a bare pipe to a highly conducting body in contact with it is far greater than that possible from an insulated pipe to a similar body. When piping is in contact with steel or mortar structures, insulation is of greater value than indicated by tabular data given herein.

The saving due to use of insulation on piping is illustrated in *Example 3*.

Example 3. If the steam line given in *Examples 1* and *2* is covered with 1 in. thick 85 per cent magnesia, determine the resulting total annual loss through the insulation.

Also compute the monetary value of the annual saving and the percentage of saving over the heat loss from the bare pipe.

Solution. By referring to Fig. 1, the coefficient for 1 in. magnesia on a 2-in. pipe is found to be 0.300 Btu per hour per linear foot of pipe per degree temperature difference at a temperature difference of 169.4 F. The total hourly loss per linear foot of pipe will then be 0.300 × 169.4 = 50.8 Btu. The total annual loss through the insulation = 50.8 × 165 (linear feet) × 4000 (hours) = 33,500 *Mb*. The annual bare pipe loss as determined in the solution of *Example 1* was found to be 181,600 *Mb*. The saving due to insulation is then 181,600 − 33,500 = 148,100 *Mb* per year.

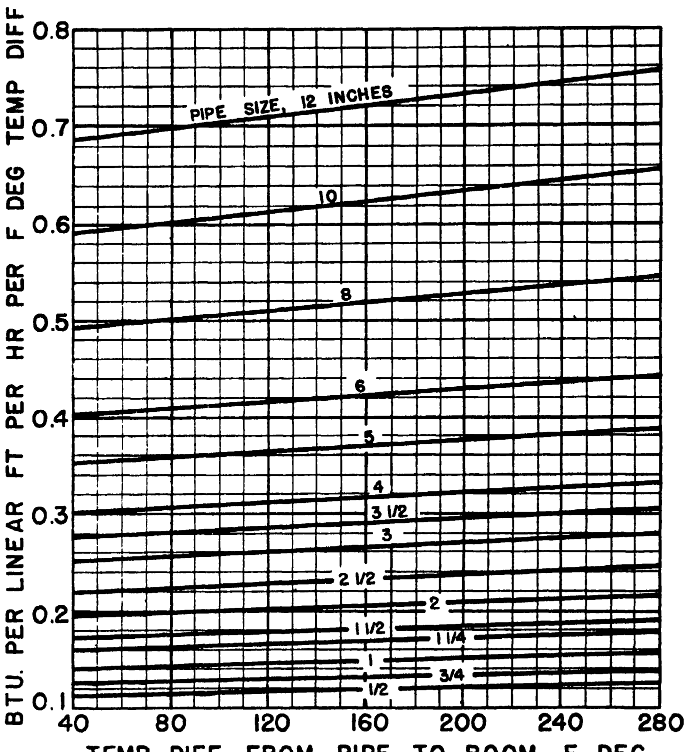


FIG. 3. HEAT LOSS THROUGH 2 IN. THICK 85 PER CENT MAGNESIA TYPE COVERING

From the solution of *Example 2*, it was found that the heat supplied to the system cost $0.804 per thousand *Mb*. Therefore, the monetary value of the saving = 0.804 (dollars) × 148.1 (thousand *Mb*) = $119.07, or 81.5 per cent of the cost when using uninsulated pipe.

LOW TEMPERATURE PIPE INSULATION

Surfaces maintained at temperatures lower than the surrounding air are insulated to reduce the flow of heat and to prevent condensation. The insulating material should absorb a minimum amount of moisture, because the absorption of moisture substantially increases the conductivity of the material. This property is particularly important in the insulation of surfaces that are below the dew-point of the surrounding air. In such cases, due to vapor pressure difference, it is necessary to seal the surface of the insulating material against the penetration of

water vapor which would condense within the material, causing a serious increase in heat flow, possible breakdown of the material, and corrosion of metal surfaces. An insulating material with a high degree of moisture absorption might pick up moisture before application and then, when the seal is in place and the temperature of the insulated surface reduced, release that moisture to the cold surface. There are a number of methods of producing vapor seals, some of which have been worked out by insulation manufacturers to suit their products and others by applicators and users. Unless time proven methods are known, specifications of insulation manufacturers should be obtained and followed carefully.

The thickness of insulation required to prevent condensation on the outer surface is that thickness which will raise the temperature of the outer surface of the insulation to a point slightly higher than the dew-point of the surrounding vapor. The dew-point for various humidities can be readily ascertained from a psychrometric chart.

The approximate required thickness of insulation to prevent conden-

TABLE 9. PIPE COVERING FACTORS

Types of Insulating Materials	Temperature Difference, Pipe to Air, Deg F				
	100	200	300	400	500
Corrugated Asbestos—Type					
4 Ply per 1 in.	1.30	1.36	1.42	.	
6 Ply per 1 in.	1.19	1.23	1.27	..	
8 Ply per 1 in.	1.15	1.19	1.23	....	
Laminated Asbestos—Type	0.96	0.98	1.00	1.02	1.04
Mineral Wool—Type	0.98	1.00	1.02	1.05	1.07
Diatomaceous Silica—Type	1.37	1.36	1.35	1.35	1.34
Brown Asbestos Fiber—Type	0.86	0.88	0.91	0.93	0.96

sation on pipes and flat metallic surfaces may be obtained from Fig. 4 in which a surface resistance of 0.606 corresponding to a film conductance of 1.65, was used in calculating the curves. This value provides a slight factor of safety and its use is known to give satisfactory field results. In using the chart it is advisable to specify the next thicker, rather than the next thinner, commercial insulation in cases where an intermediate thickness is indicated.

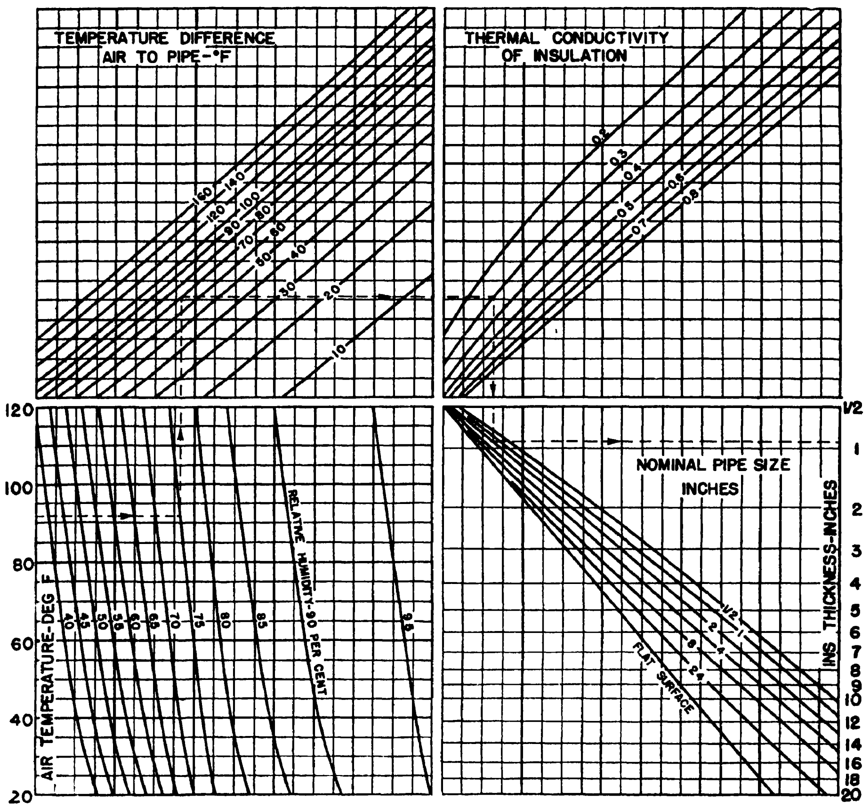
Heat gains for pipes insulated with a material having an installed conductivity of 0.30 Btu per (sq ft) (hr) (deg F per inch) are given in Table 10. This table may be used for any of the commercial insulations offered for this purpose since they have conductivities very near the 0.3 value used.

INSULATION OF PIPES TO PREVENT FREEZING

If the surrounding air temperature remains sufficiently low for an ample period of time, insulation cannot prevent the freezing of still water, or of water flowing at such a velocity that the quantity of heat carried in the water is not sufficient to take care of the heat losses which will result and cause the temperature of the water to be lowered to the freezing point. Insulation can materially prolong the time required for the water to give up its heat, and if the velocity of the water flowing in the pipe is maintained at a sufficiently high rate, freezing may be prevented.

Table 11 may be used for making estimates of the thickness of insu-

lation necessary to take care of still water in pipes at various water and surrounding air temperature conditions. Because of the damage and service interruptions which may result from frozen water in pipes, it is essential that an efficient insulation be utilized. This table is based on the use of a material having a conductivity of 0.30. The initial water temperature is assumed to be 10 F above, and the surrounding air temperature 50 F below the freezing point of water (temperature difference, 60 F).



ᵃSolve problems as indicated by dotted line, entering chart at lower left-hand scale.

FIG. 4. THICKNESS OF PIPE INSULATION TO PREVENT CONDENSATION ON OUTER SURFACE[a]

The last column of Table 11 gives the minimum quantity of water at initial temperature of 42 F which should be supplied every hour for each linear foot of pipe, in order to prevent the temperature of the water from being lowered to the freezing point. The weights given in this column should be multiplied by the total length of the exposed pipe line expressed in feet. As an additional factor of safety, and in order to provide against temporary reductions in flow occasioned by reduced pressure, it is advisable to double the rates of flow listed in the table. It must be emphasized that the flow rates and periods of time designated apply only for the conditions stated. To estimate for other service conditions the following method of procedure may be used.

TABLE 10. HEAT GAINS FOR INSULATED COLD PIPES

Rates of heat transmission given in Btu per (hour) (degree Fahrenheit temperature difference between fluid in pipe and surrounding still air)

Based on materials having conductivity, k = 0.30

Nominal Pipe Size (Inches)	Ice Water Thickness			Brine Thickness			Heavy Brine Thickness		
	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface
½	1.5	0.110	0.502	2.0	0.098	0.446	2.8	0.087	0.394
¾	1.6	0.119	0.431	2.0	0.111	0.405	2.9	0.094	0.340
1	1.6	0.139	0.403	2.0	0.124	0.352	3.0	0.104	0.294
1¼	1.6	0.155	0.357	2.4	0.131	0.300	3.1	0.113	0.260
1½	1.5	0.174	0.351	2.5	0.134	0.270	3.2	0.118	0.238
2	1.5	0.200	0.322	2.5	0.151	0.244	3.3	0.134	0.214
2½	1.5	0.228	0.303	2.6	0.170	0.226	3.3	0.147	0.197
3	1.5	0.269	0.293	2.7	0.186	0.202	3.4	0.162	0.176
3½	1.5	0.295	0.282	2.9	0.191	0.183	3.5	0.176	0.167
4	1.7	0.294	0.248	2.9	0.209	0.176	3.7	0.182	0.154
5	1.7	0.349	0.239	3.0	0.241	0.165	3.9	0.202	0.138
6	1.7	0.404	0.233	3.0	0.259	0.150	4.0	0.228	0.130
8	1.9	0.455	0.201	3.0	0.318	0.140	4.0	0.263	0.116
10	1.9	0.559	0.198	3.0	0.383	0.135	4.0	0.309	0.110
12	1.9	0.648	0.194	3.0	0.438	0.131	4.0	0.364	0.108

If water enters the pipe at 52 F instead of 42 F, the time required to cool it to the freezing point will be prolonged to twice that given in the table, or the rate of flow of water may be reduced so that the quantity required will be one-half that shown in the last column of Table 11. However, if the water enters the pipe at 34 F it will be cooled to 32 F in one-fifth of the time given in the table. It will then be necessary to increase the rate of flow so that five times the specified quantity of water will have to be supplied in order to prevent freezing.

If the minimum air temperature is −38 F (temperature difference 80 F) instead of −18 F, the time required to cool the water to the freezing point will be 60/80 of the time given in the table, or the necessary quantity of water to be supplied will be 80/60 of that given.

TABLE 11. DATA FOR ESTIMATING REQUIREMENTS TO PREVENT FREEZING OF WATER IN PIPES WITH SURROUNDING AIR AT −18 F

Nominal Pipe Size (Inches)	Number of Hours to Cool 42 F Water to Freezing Point			Water Flow Required at 42 F to Prevent Freezing, Pounds per Linear Foot of Pipe per Hour		
	Thickness of Insulation in Inches (Conductivity, k = 0.30)					
	2	3	4	2	3	4
½	0.42	0.50	0.57	0.54	0.45	0.40
1	0.83	1.02	1.16	0.68	0.55	0.48
1½	1.40	1.74	2.02	0.84	0.68	0.58
2	1.94	2.48	2.90	0.95	0.75	0.64
3	3.25	4.27	5.08	1.24	0.94	0.79
4	4.55	6.02	7.20	1.47	1.11	0.93
5	5.92	7.96	9.69	1.73	1.29	1.06
6	7.35	9.88	12.20	1.98	1.46	1.19
8	10.05	13.90	17.25	2.46	1.78	1.43
10	13.00	18.10	22.70	2.96	2.12	1.70
12	15.80	22.20	28.10	3.43	2.45	1.93

TABLE 12. THICKNESSES OF PIPE INSULATION ORDINARILY USED INDOORS[a]

Steam Pressures (Lb Gage) or Conditions	Steam Temperatures Degrees Fahrenheit	Thickness of Insulation		
		Pipes Larger Than 4 In.	Pipes 2 In. to 4 In.	Pipes ½ In. to 1½ In.
0 to 25	212 to 267	1 in.	1 in.	1 in.
25 to 100	267 to 338	1½ in.	1 in.	1 in.
100 to 200	338 to 388	2 in.	1½ in.	1 in.
Low Superheat	388 to 500	2½ in.	2 in.	1½ in.
Medium Superheat	500 to 600	3 in.	2½ in.	2 in.
High Superheat	600 to 700	3½ in.	3 in.	2 in.

[a]All piping located outdoors or exposed to weather is ordinarily insulated to a thickness ½ in. greater than shown in this table, and covered with a waterproof jacket.

In making calculations to arrive at the values given in Table 11, the loss of heat stored in the insulation, the effect of a varying temperature difference due to the cooling of pipe and water, and the resistance of the outer surface of the insulation to the transfer of heat to the air have all been neglected. When these factors enter into the computations it is necessary to enlarge the factor of safety. Also as stated, the time shown in the table is that required to lower the water to the freezing point. A longer period would be required to freeze the water but the danger point is reached when freezing starts. The flow of water will stop and the entire line will be in danger as soon as the water freezes across the section of the pipe at any point.

When water must remain stationary longer than the times designated in Table 11, the only safe way to insure against freezing is to install a steam or hot water line or to place an electric resistance heater along the side of the exposed water line. The heating system and the water line are then insulated so that the heat losses from the heating system are not excessive, and the heating effect is concentrated against the water pipe where it is needed. For this form of protection 2 in. of an efficient insulation may be applied.

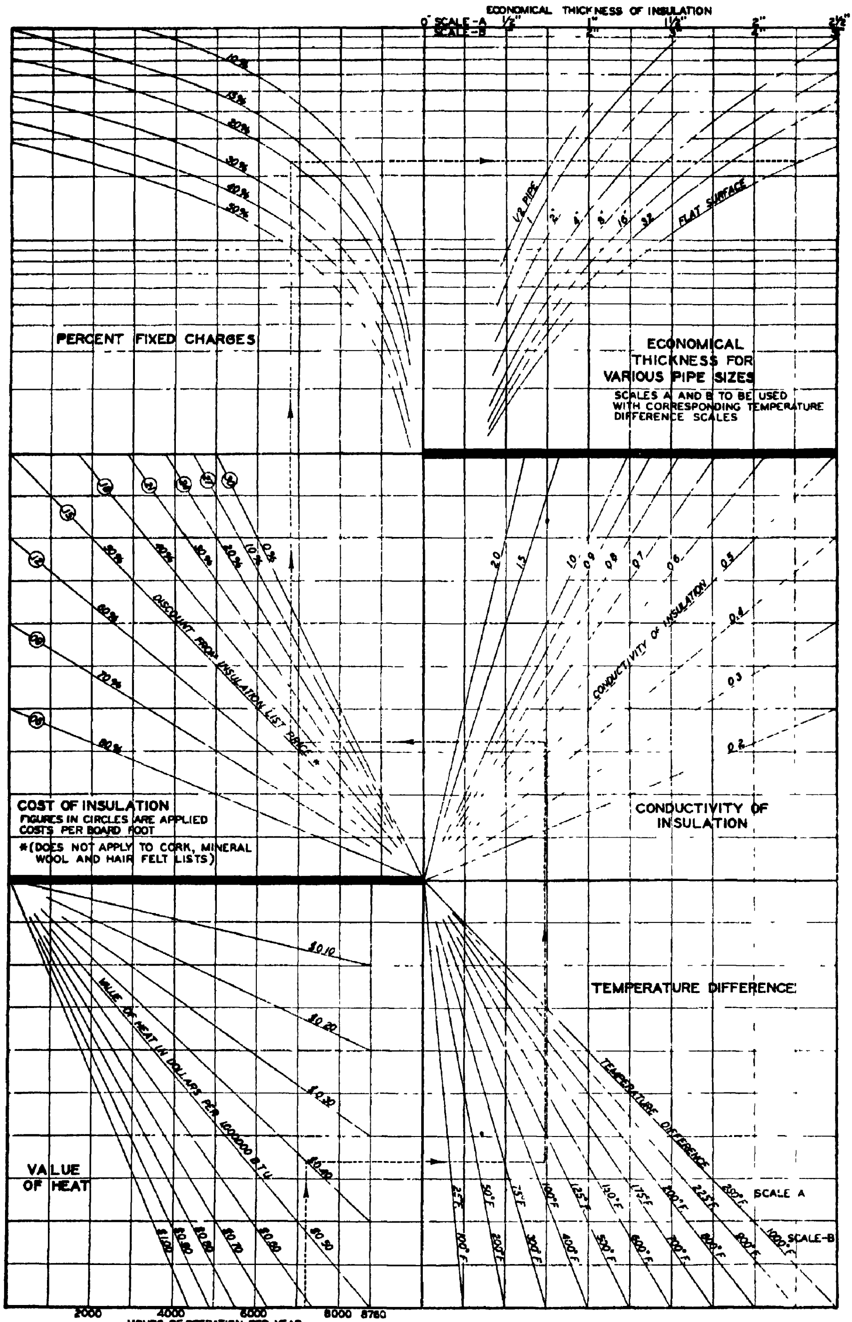
ECONOMICAL THICKNESS OF PIPE INSULATION

The thicknesses of insulation which ordinarily are used for various temperature conditions are given in Table 12. Where a thorough analysis of economic thickness is desired this may be accomplished through the use of the chart, Fig. 5.

The dotted line on the chart illustrates its use in solving a typical example. In using the chart, start with the scale at the left bottom

TABLE 13. THICKNESS OF LOOSE INSULATION FOR USE AS FILL IN UNDERGROUND CONDUIT SYSTEMS

Steam Pressures (Lb Gage) or Conditions	Steam Temperatures Degrees Fahrenheit	Minimum Thickness of Insulation in Inches					Minimum Distance Between Steam and Return
		Steam Lines			Return Lines		
		Pipes Less than 4 In.	Pipes 4 In. to 10 In.	Pipes Larger than 12 In.	Pipes Less than 4 In.	Pipes 4 In. and Larger	
Hot Water, or 0 to 25	212 to 267	1½	2	2½	1¼	1½	1
25 to 125	267 to 352	2	2½	3	1¼	1½	1¼
Above 125, or superheat	352 to 500	2½	3	3½	1¼	1½	1½



(L. B. McMillan, *Proc National District Heating Association*, Vol. 18, p. 138).

FIG. 5. CHART FOR DETERMINING ECONOMICAL THICKNESS OF PIPE INSULATION

margin representing the given number of hours of operation per year; then proceed vertically to the line representing the given value of heat; thence horizontally to the right, to the line representing the given temperature difference; thence vertically to the line representing the conductivity of the given material; thence horizontally, to the left, to the line representing the given discount on that material; thence vertically to the curve representing the required per cent return on the investment; thence horizontally to the right, to the curve representing the given pipe size; thence vertically to the scale at the top right margin where the economical thickness may be read off directly.

UNDERGROUND PIPE INSULATION

Underground steam distribution lines are carried in protective structures of various types, sizes and shapes (See Chapter 43). Detailed data on commonly used forms of tunnels and conduit systems have been published by the *National District Heating Association*[2].

Pipes in tunnels are covered with sectional insulation to provide maximum thermal efficiency and are also finished with good mechanical protection in the form of metal or waterproofing membrane outer jackets. In some instances, where actual submersion of hot lines may occur it has been found good practice to firmly secure the covering with corrosion resistant wire, then sew on a wire-inserted asbestos fabric jacket with wire. This jacket is porous. The principle of withstanding submersion is that water may enter as water, then actually boil at the pipe surfaces and escape as steam without rupturing the insulation or jacket. Conduit systems are in more general use than tunnels. Pipes carried in conduits may be insulated with sectional insulation; however, the more usual practice is to fill the entire section of the conduit around the pipes with high quality, loose insulating material. The insulation must be kept dry at all times, and for this purpose effective waterproofing membranes enclose the insulation. A drainage system is also provided to divert water which may tend to enter the conduit.

The economical thickness of insulation for underground work is difficult to determine accurately due to the many variables which have to be considered. As a result of theories[3] previously developed, together with other experimental data which have been presented, the usual endeavor is to secure not less than 90 per cent efficiency for underground piping. Table 13 can be used as a guide in arriving at the minimum thickness of loose insulation fills to use for laying out conduit systems. Other factors such as the number of pipes and their combination of sizes, as well as the standard conduit sizes, are primary controlling factors in the amount and thickness of insulation for use.

When sectional insulation is applied to lines in tunnels or conduits, usual practice is to apply the most efficient materials ½ in. less in thickness than that determined by the use of Fig. 5. The data in Fig. 5 are based on conditions of insulation exposed to the air, whereas normal ground temperature is substituted for air temperature in determining the temperature difference for use with the chart when applying it for underground pipe line estimates.

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[2]Handbook of the *National District Heating Association*, Second Edition, 1932.

[3]Theory of Heat Losses from Pipes Buried in the Ground, by J. R. Allen (A.S.H V.E. TRANSACTIONS Vol 26, 1920, p. 335).

CHAPTER 18

Gravity Warm Air Furnace Systems

Warm Air Leaders, Stacks, and Registers; Return Air Ducts, Grilles, and Shoe Connections; Outline of Design Procedure; Furnace Capacity

WARM air heating systems of the gravity type are described in this chapter[1], and those of the mechanical type are described in Chapter 19. In the gravity type, the motive head producing flow depends upon the difference in weight between the heated air leaving the top of the casing and the cooled air entering the bottom of the casing, while in the mechanical type a fan may supply all or part of the motive head. Booster fans are often used in conjunction with gravity-designed systems to increase air circulation.

In general, a warm-air furnace heating plant consists of a fuel-burning furnace or heater, enclosed in a casing of sheet metal or brick, which is placed in the basement of the building. The heated air, taken from the top or sides near the top of the furnace casing, is distributed to the various rooms of the building through sheet metal warm-air pipes. The warm-air pipes in the basement are known as leaders, and the vertical warm-air pipes which are run in the inside partitions of the building are called stacks. The heated air is finally discharged into the rooms through registers which are set in register boxes placed either in the floor or in the side wall, usually at or near the baseboard. A sectional view of a typical plant showing good installation practice is given in Fig. 1.

The air supply to the furnace is usually taken entirely from inside the building through one or more recirculating ducts, although in some cases an outside air supply duct is provided.

WARM AIR LEADERS, STACKS, AND REGISTERS

In a gravity circulating warm-air furnace system, the size of the leader pipe to a given room depends upon the length of the leader and the temperature of the warm air entering the room at the register. For most successful operation, the furnace should be centrally located with respect to register and stack positions so that the leaders will be of uniform length and as short as possible, in which case the frictional resistance to air flow and the temperature loss from the ducts will be about the same for all runs.

In the Standard Code for Installation of Gravity Warm Air Heating Systems, the design was originally based on the heat carrying capacities per square inch of leader pipe area, as shown in Table 1. These values were based on register air temperatures of 175 F. In a recent revision of the entire design procedure, as outlined in the section entitled *Outline of Design Procedure*, the carrying capacities of leader pipes have been expressed directly in terms of Btu per hour.

In residences requiring a total leader pipe area of less than about 650 sq in., it is advisable to use two or more leader pipes to rooms requiring more than the capacity of a 12 in. round pipe. The tops of all sizes of leader pipes should be cut into the furnace bonnet at the same elevation, and from this point there should be a uniform upgrade of at least 1 in.

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[1]The engineering data which follow are from University of Illinois, *Engineering Experiment Station Bulletins* Nos. 141, 188, 189 and 246; Warm Air Furnaces and Heating Systems, by A. C. Willard, A. P Kratz, V. S. Day, and S. Konzo. See also Standard Code Application Manual for Gravity Warm Air Heating Systems, published by the *National Warm Air Heating and Air Conditioning Association*.

per foot of run. Leaders over 12 ft in length, or having a large number of elbow fittings should be avoided if possible. In cases where such leaders are necessary, it is recommended that smooth transition fittings be used, and that duct insulation be applied. Asbestos paper, unless of the corrugated type, should not be considered as insulation. To assist in balancing the air distribution of the system, a damper should be placed

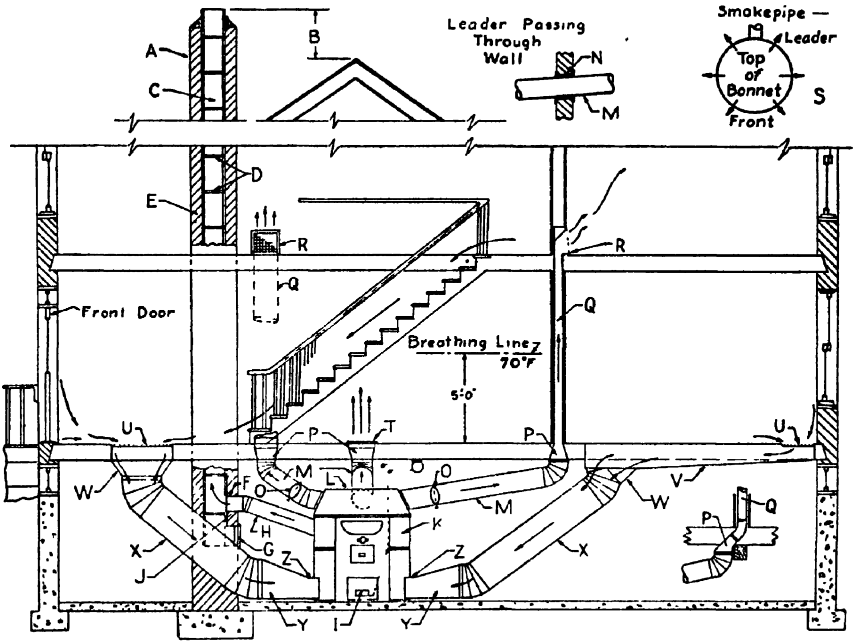

FIG. 1. A SECTIONAL VIEW OF A TYPICAL PLANT SHOWING GOOD INSTALLATION PRACTICE[a]

A. House chimney, no bends nor offsets.
B Top of chimney at least 2 feet above ridge of roof
C. Flue lining, fireclay
D. All joints air tight.
E At least 8 in brick
F. No other connection beside that to furnace
G. Cleanout frame and door, airtight
H. Smoke pipe, end flush with inner surface of flue
I. Draft door
J. Use flue thimble.
L. Casing hood or bonnet, top of all leader collars on same level.
K. Casing body.
M. Round leader, pitch 1 in. per foot.
N. Sleeve with air space around leader where passing through wall
O. Dampers in all leaders.
P. Transition fittings
Q. Rectangular wall stack
R. Baseboard register.
S. Distribute pipes equally around bonnet
T. Floor register.
U Return air face
V. Panning under joist.
W. Transition collar.
X. Round return pipe
Y. Transition shoe.
Z Top of shoe at casing not above grate level.

[a](From N.W.A.H &A C. Assn Standard Code Application Manual)

in each leader pipe except one, this latter leader preferably being connected to a room heated at all times, such as a living room.

In a gravity circulating system, the ratio of stack to leader area is quite important, although little is gained by providing wall stacks with areas in excess of 75 per cent of their connected leader pipe area. In most cases a 3¼ in. × 12 in. stack is the largest which can be installed in normal wall construction. Hence, any room having a heat loss much in excess of 9000 Btu per hour, will require two or more stacks, or one oversized stack built into a 6 in. studding space, providing the design

register temperature is to be retained at the value of 175 F as recommended.

Registers used for discharging warm air into rooms should have a net area not less than the area of the leader pipe to which the register is attached. First story registers should be connected through boot and register box extensions having areas at least equal to leader areas. Upper story registers should be of the same width as the wall stack, and should be placed either in the baseboard or sidewall, preferably without offsets. First story registers may be of the baseboard or floor type, with the former location preferred. High sidewall locations for warm-air registers in gravity systems deliver a greater quantity of warm air into the room than do baseboard registers, but most of the additional air merely results in high temperatures at the ceiling.

RETURN AIR DUCTS, GRILLES, AND SHOE CONNECTIONS

The ducts through which air is returned to the furnace should be designed to minimize resistance to air flow. They should be of ample area, in excess of the total area of warm-air pipes, and should be stream-lined. Horizontal ducts should pitch at least ½ in. per foot downward

TABLE 1. FORMER METHOD OF EXPRESSING LEADER PIPE CARRYING CAPACITIES

Register Location	Carrying Capacity of Each Square Inch of Leader Pipe for 175 F Register Temperature, in Btu per Hour	
First Story	105[a]	111[b]
Second Story	170[a]	166[b]
Third Story	208[a]	200[b]

[a]Actual capacity. [b]Used for average calculations.

toward the furnace, avoiding fittings which would require lifting of the return air after the duct has passed under some obstacle.

The return air grilles should have free areas at least equal to the ducts to which they connect. These return air grilles should be installed in the floor, or in the baseboard with the top edge of the grille not more than about 8 in. above the floor line. Frictional resistance in the return air system is as detrimental to proper performance as is resistance in the warm-air system, so that care should be exercised in locating return air grilles which require long return ducts.

The placement and number of return grilles will depend upon the size, details, and exposure of the house. Small compactly built houses may be adequately served by a single return grille effectively placed in the central hall. It is usually desirable to have two or more returns, provided that in two-story residences one return is placed to effectively receive the return air at the foot of the stairs. A return air connection must be carried to any heated room whose floor level is below that of adjacent rooms.

Where a divided system of two or more returns is used, the grilles must be placed to serve the maximum area of cold wall or windows. Thus, in rooms having only small windows the grille can be brought as close to the furnace as possible, but if the room has large window exposure the grille should be located near the exposure. If long ducts are used in parallel with short return ducts, the frictional resistance of the long ducts must be reduced to compensate for the length. Return ducts from up-stairs rooms may be necessary in spaces which are closed off from the

rest of the house or which have much outdoor exposure. Return grilles on different floor levels should not be connected to the same vertical return duct.

Ducts returning air to the furnace should avoid heat sources which tend to reheat the return air. If the duct must be run over the top of the furnace, or above the vent pipe from the furnace, insulation should be interposed between the heat source and the duct.

Circulation of air is facilitated if the air can slide down a pipe inclined at approximately 45 deg and into a furnace shoe connection having a cross-sectional area equal to that of the pipe. The top of the return shoe should enter the casing below the level of the grate in the case of a coal furnace, and not more than 14 in. above the floor in the case of oil or gas furnaces. In order to accomplish this the shoe is made wide.

OUTLINE OF DESIGN PROCEDURE

The data underlying the design procedure are given in detail in a circular[2] issued by the University of Illinois. In this procedure the design of the warm-air duct system is considered as an entire unit, so that for a given heat loss the sizes of leaders, stacks, boots, stackheads, and registers are

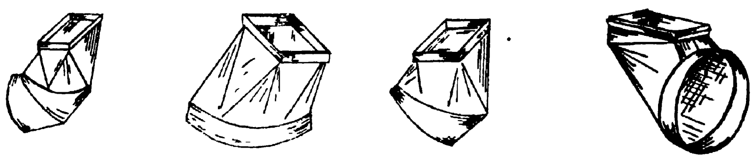

FIG. 2. TYPICAL BOOTS

all correlated. Similarly in the case of return ducts, the selection specifies a complete unit consisting of return grille, return duct, and shoe connection.

Recommended Standard Sizes

For the purpose of simplification and standardization, selected combinations of commercial sizes of warm air pipes, return air pipes, ducts, grilles, fittings, and registers are designated as *units*. The unit numbers assigned and the combinations selected as standard are listed in the following tables[3]:

Table 2—Units 1 to 5—First Story Warm Air Ducts and Registers.

Table 3—Units 11 to 16—Second Story Warm Air Ducts, Single Wall Stacks, Fittings, and Registers.

Table 4—Units 21 to 24—Second Story Warm Air Ducts, Double Wall Stacks, Fittings, and Registers.

Table 5—Units 31 to 38—Return Air Ducts, Fittings, and Grilles.

The selected types of boots are shown in Fig. 2. It is essential that free areas be maintained throughout fittings.

The selected types of return air ducts and fittings are shown in Fig. 3.

Carrying Capacity

The Btu carrying capacities of the selected units of warm air and return air combinations are shown in Tables 6, 7 and 8:

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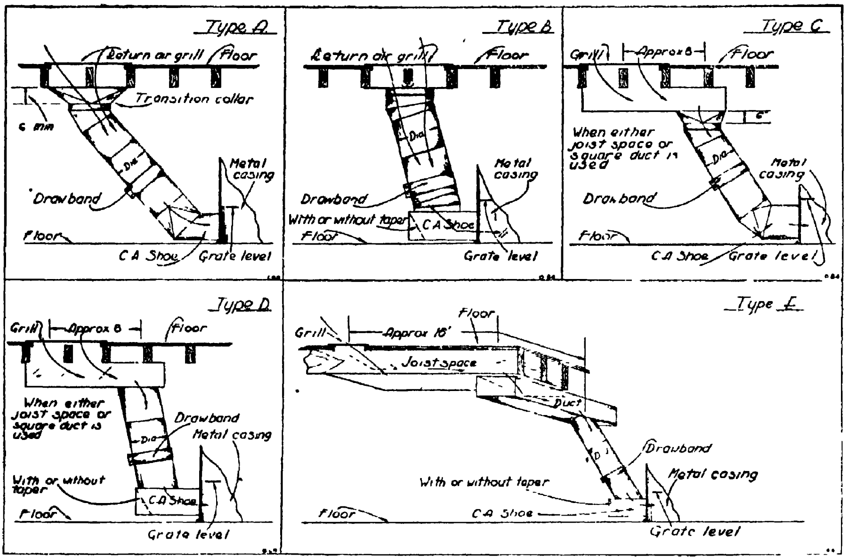
[2]Simplified Procedure for Selecting Capacities of Duct Systems for Gravity Warm-Air Heating Plants, by A. P. Kratz and S. Konzo (*University of Illinois, Engineering Experiment Station Circular* 46, Dec., 1942).

[3]Standard Code Application Manual for Gravity Warm Air Heating Systems, First Edition, 1941, *National Warm Air Heating and Air Conditioning Association.*

Table 6—Units 1 to 5—Warm Air Carrying Capacities to first story registers with 1 to 5 elbows and with leaders 4 to 24 ft long.

Table 7—Units 11 to 16 and 21 to 24—Warm Air Carrying Capacities to second story registers with 1 to 5 elbows and with leaders 4 to 24 ft long.

Table 8—Units 31 to 38—Return Air Carrying Capacities for types *A*, *B*, *C*, *D* and *E* return combinations.



aFull areas to be maintained from grille to furnace.

FIG. 3, RETURN AIR COMBINATIONS[a]

Notes:

Type A return air is direct from grille to casing, using transition collar and transition type *C*. *A* shoe

Type B same as *A*, but without transition fittings (capacity for same size grille and pipe 84 per cent of *A*)

Type C uses joist space *or* square duct and transition fittings (capacity for same size grille and pipe 84 per cent of *A*).

Type D same as *C*, but without transition fittings (capacity for same size grille and pipe 69 per cent of *A*)

Type E uses *both* joist space and square duct, without transition fittings (capacity for same size grille and pipe 44 per cent of *A*).

Design Procedure

The steps to be taken in designing a gravity warm air duct system are:

1. Calculate the heat loss from each room as explained in Chapters 4, 5 and 6.

2. Prepare a layout showing (*a*) furnace, (*b*) chimney connection, (*c*) warm air registers (whether floor, base board or wall), (*d*) return air grilles

3. Indicate on each warm air run (using symbols shown in Fig. 4): (*a*) whether the room to be heated is on the first or second story, (*b*) the approximate length of leader pipe in the basement, (*c*) the number of right angle elbows required, including the elbow at the boot connection. (Consider each 45 deg elbow as equal to ½ of 90 deg elbow, consider the two sharp 90 deg elbows in a cross-over connection as equivalent to three 90 deg elbows), (*d*) whether the register is to be located in the floor, in the baseboard, or in the wall.

4. Show the number and proposed locations of return air grilles and the type of return air system (see Fig. 3).

5. From Table 6, for first story, or from Table 7 for second story, select the unit number for the warm air system which will supply the heat required to each room, with the number of elbows and length of leader pipe previously determined. Then, using the unit number as found, read directly in Tables 2, 3, or 4 the leader, stack, and register sizes required.

6. From Table 8 select the unit number for the return air system to correspond with the Btu serviced and the type of return air system. Then from Table 5 select the duct and grille sizes, etc., corresponding to the same unit number.

7. Select a furnace having a capacity rating, in Btu per hour, equal to the total heat loss from the structure.

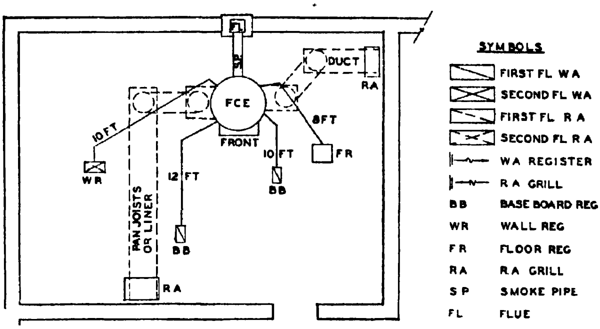


Fig. 4. Typical Basement Line Drawing

Examples

Examples 1 and 2 will illustrate the use of the tables in selecting warm air and return system sizes.

Example 1. For a room which has a heat loss of 22,500 Btu per hour select the size of first story warm air system. There are three elbows and the leader is approximately 10 ft long.

Solution: Since 22,500 Btu is beyond the capacities shown in Table 6, it is necessary to select two units of 11,250 each. From Table 6 in 10 ft leader column and in section for three elbows find 11,400 as nearest capacity which corresponds to Unit No. 4 in first column. Refer to Unit No. 4 in Table 2 and find that the leader should be 12 in. in diameter and should be used with a 12 x 14 in. floor register or a 13 x 11 in. baseboard register with a 5¼ in. extension.

Example 2. What is the size of a return system of Type D which is to service 35,000 Btu per hour?

Solution: From Table 8 find Unit No. 37 which will service 37,800 Btu per hour. Refer to Table 5 to find that Unit No. 37 will require a 22-in diameter duct, a shoe area of 420 sq in., a metal grille 18 x 30 in., a duct 42 x 10 in. or 36 x 12 in. If joist lining is used the minimum depth should be 15 in. for two 2-joist spaces 14 in. wide, or 10 in. for three joist spaces

Table 2. First Story Warm-Air Ducts[a]

Unit No.	Leader Pipe Diameter, In	Register Size, In.		
		Floor	Baseboard	
			Size	Extension
1	8	8 x 10	10 x 8	2¼
2	9	9 x 12	12 x 8	2¼
3	10	10 x 12	12 x 9	3¼
4	12	12 x 14	13 x 11	5¼
5	14	14 x 16	....	......

[a]When the calculations indicate a requirement for a given room greater than Unit No. 4, two or more smaller units totalling the required capacity are recommended.

Table 3. Second Story Warm-Air Ducts—Single Wall Stacks and Fittings

Unit No.	Leader Pipe Diameter, In.	Stack[b] Size In.	Register Size, In.			
			Floor	Baseboard		Sidewall
				Size	Extension	
11	8	10 x 3¼	8 x 10	10 x 8	2¼	10 x 8
12	9	12 x 3¼	8 x 12	12 x 8	2¼	12 x 8
13	9	13 x 3¼	8 x 14	12 x 8	2¼	12 x 8
14	10	14 x 3¼	8 x 14	12 x 8	2¼	12 x 8
15	12	12 x 5¼	....	12 x 10	3¼	....
16	12	14 x 5¼	....	13 x 11	3¼	... .

[b]Recommended stack sizes. Tables may also be applied to 3 in and 3½ in stack depths

Table 4. Second Story Warm-Air Ducts—Double Wall Stacks and Fittings

Unit No	Leader Pipe Diameter, In.	Stack Size, In.		Register Size, In			
		Internal	External	Floor	Baseboard		Sidewall
					Size	Extension	
21	8	2½[c] x 10	3⅛[c] x 10⅝	8 x 10	10 x 8	2¼	10 x 8
22	8	3 x 10	3⅝ x 10⅝	8 x 10	10 x 8	2¼	10 x 8
23	9	2½[c] x 12	3⅛[c] x 12⅝	8 x 12	12 x 8	2¼	12 x 8
24	9	3 x 12	3⅝ x 12⅝	8 x 12	12 x 8	2¼	12 x 8

[c]Commercial sizes vary ⅛ in from values shown.

Table 5. Return Air Ducts

Unit No.	Duct Dia. In.	Area at Shoe Connection, Sq In.	Metal Grille Sizes			When Joist Lining is Used[d]		When Duct is Used	
			Choose One			No of Joists Lined	Minimum[e] Depth, In.	Choose One	
			A	B	C				
31	10			8 x 14	10 x 12	1	7	14 x 6	12 x 8
32	12		6 x 30	8 x 24	12 x 14	1	9	22 x 6	16 x 8
33	14	170	8 x 30	10 x 24	14 x 16	1	12	28 x 6	22 x 8
34	16	220	10 x 30	12 x 24		2	8	28 x 8	22 x 10
35	18	280	12 x 30	14 x 24		2	10	36 x 8	28 x 10
36	20	340	14 x 30	18 x 24		2	12.5	36 x 10	30 x 12
37	22	420	18 x 30			2	15.0	42 x 10	36 x 12
						3	10.0	42 x 10	36 x 12
38	24	500	20 x 30			2	18.0	42 x 12	36 x 14
						3	12.0	42 x 12	36 x 14

[d]Based on 14 in. space between joists.

[e]Use full depth of joist except when joist depth is less than minimum depth required, *when pan must be used.*

TABLE 6. WARM AIR CARRYING CAPACITY, BTU DELIVERED, FIRST STORY REGISTERS[a,b,c]

Length of Leader Pipe—in Feet

UNIT NO.	NO. OF ELBOWS	4 FT	6 FT	8 FT	10 FT	12 FT	14 FT	16 FT	18 FT	20 FT	22 FT	24 FT
1	1	6,020	5,850	5,680	5,510	5,340	5,170	5,000	4,830	4,660	4,490	4,320
2		7,620	7,400	7,180	6,970	6,760	6,540	6,320	6,110	5,890	5,680	5,460
3		9,400	9,140	8,870	8,600	8,340	8,070	7,810	7,540	7,270	7,010	6,740
4		13,350	12,970	12,590	12,210	11,830	11,450	11,080	10,700	10,320	9,950	9,560
5		17,520	17,020	16,530	16,040	15,550	15,050	14,550	14,050	13,560	13,060	12,560
1	2	5,850	5,660	5,490	5,330	5,160	5,000	4,840	4,670	4,510	4,340	4,180
2		7,360	7,150	6,940	6,730	6,520	6,320	6,110	5,910	5,700	5,500	5,290
3		9,090	8,840	8,580	8,320	8,060	7,800	7,550	7,290	7,040	6,780	6,520
4		12,910	12,540	12,170	11,800	11,430	11,060	10,690	10,320	9,950	9,580	9,210
5		16,940	16,450	15,990	15,500	15,040	14,550	14,080	13,600	13,120	12,650	12,150
1	3	5,620	5,460	5,310	5,150	4,990	4,830	4,670	4,510	4,350	4,190	4,030
2		7,120	6,910	6,710	6,510	6,310	6,110	5,900	5,700	5,500	5,300	5,100
3		8,780	8,530	8,280	8,030	7,780	7,530	7,290	7,040	6,800	6,550	6,300
4		12,450	12,100	11,750	11,400	11,050	10,700	10,350	10,000	9,650	9,300	8,950
5		16,360	15,900	15,440	14,970	14,510	14,050	13,600	13,130	12,660	12,200	11,750
1	4	5,420	5,260	5,110	4,960	4,800	4,650	4,500	4,350	4,190	4,040	3,890
2		6,860	6,660	6,460	6,270	6,080	5,890	5,690	5,500	5,300	5,110	4,910
3		8,460	8,200	7,980	7,740	7,500	7,260	7,020	6,780	6,550	6,310	6,070
4		12,010	11,670	11,330	10,990	10,650	10,310	9,970	9,630	9,290	8,950	8,610
5		15,770	15,320	14,880	14,420	13,990	13,540	13,100	12,650	12,200	11,750	11,310
1	5	5,240	5,090	4,940	4,790	4,640	4,500	4,350	4,200	4,050	3,910	3,760
2		6,630	6,440	6,250	6,060	5,880	5,690	5,500	5,320	5,130	4,940	4,750
3		8,180	7,950	7,720	7,490	7,260	7,030	6,800	6,560	6,330	6,100	5,860
4		11,610	11,290	10,950	10,620	10,300	9,970	9,640	9,320	8,990	8,660	8,320
5		15,250	14,800	14,380	13,950	13,520	13,090	12,650	12,230	11,800	11,370	10,940

[a]Table 6 applies to baseboard and floor locations of warm air registers.

[b]Consider each 45 deg elbow as equal to one-half of a 90 deg elbow.

[c]Consider each 90 deg offset between boot and stackhead equal to one 90 deg round elbow.

TABLE 7. WARM AIR CARRYING CAPACITY, BTU DELIVERED, SECOND STORY REGISTERS[a,b,c,d]

Length of Leader Pipe—in Feet

UNIT No[e,f]	No. OF ELBOWS	4 FT	6 FT	8 FT	10 FT	12 FT	14 FT	16 FT	18 FT	20 FT	22 FT	24 FT
11-22	1	8,370	8,140	7,900	7,670	7,430	7,190	6,950	6,710	6,470	6,240	6,000
12-24		10,040	9,760	9,470	9,190	8,900	8,620	8,330	8,050	7,770	7,480	7,200
13		10,880	10,570	10,260	9,960	9,650	9,340	9,030	8,730	8,420	8,110	7,800
14		11,710	11,380	11,050	10,720	10,390	10,060	9,720	9,390	9,060	8,730	8,400
15		16,200	15,750	15,300	14,840	14,380	13,920	13,460	13,000	12,550	12,100	11,640
16		18,920	18,390	17,850	17,310	16,780	16,240	15,710	15,180	14,640	14,100	13,570
11-22	2	7,940	7,720	7,500	7,280	7,050	6,830	6,600	6,370	6,150	5,930	5,700
12-24		9,540	9,270	9,000	8,730	8,460	8,190	7,920	7,650	7,380	7,110	6,840
13		10,340	10,050	9,750	9,460	9,170	8,880	8,580	8,290	8,000	7,700	7,410
14		11,120	10,810	10,500	10,180	9,870	9,550	9,230	8,920	8,610	8,290	7,980
15		15,400	14,970	14,530	14,100	13,670	13,230	12,800	12,360	11,930	11,500	11,070
16		17,980	17,470	16,960	16,450	15,950	15,430	14,830	14,420	13,910	13,400	12,890
11-22	3	7,530	7,320	7,110	6,900	6,680	6,470	6,250	6,040	5,830	5,620	5,400
12-24		9,030	8,780	8,520	8,270	8,010	7,750	7,500	7,240	6,990	6,730	6,470
13		9,800	9,520	9,240	8,960	8,690	8,410	8,130	7,850	7,580	7,300	7,020
14		10,530	10,240	9,940	9,650	9,350	9,050	8,750	8,450	8,160	7,860	7,560
15		14,580	14,180	13,780	13,370	12,950	12,530	12,120	11,710	11,300	10,890	10,480
16		17,040	16,550	16,070	15,580	15,110	14,620	14,140	13,660	13,180	12,700	12,210
11-22	4	7,120	6,920	6,720	6,520	6,310	6,110	5,900	5,700	5,500	5,300	5,100
12-24		8,530	8,290	8,050	7,810	7,570	7,330	7,080	6,840	6,600	6,360	6,120
13		9,250	8,990	8,720	8,460	8,200	7,940	7,680	7,410	7,150	6,890	6,630
14		9,950	9,670	9,390	9,110	8,830	8,550	8,260	7,980	7,700	7,420	7,140
15		13,780	13,390	13,000	12,610	12,220	11,830	11,440	11,050	10,670	10,280	9,890
16		16,080	15,620	15,170	14,710	14,260	13,810	13,350	12,900	12,440	11,980	11,530
11-22	5	6,700	6,510	6,320	6,130	5,940	5,750	5,560	5,370	5,180	4,990	4,800
12-24		8,040	7,810	7,580	7,350	7,130	6,900	6,670	6,440	6,220	5,990	5,760
13		8,710	8,460	8,210	7,970	7,720	7,470	7,230	6,980	6,730	6,490	6,240
14		9,370	9,110	8,850	8,580	8,310	8,050	7,780	7,510	7,250	6,980	6,720
15		12,970	12,600	12,240	11,870	11,500	11,140	10,770	10,400	10,040	9,680	9,310
16		15,140	14,710	14,280	13,850	13,420	13,000	12,570	12,140	11,710	11,280	10,850

[a]*Btu deliveries* shown in Table 7 apply only to baseboard locations of warm air registers. *When floor registers are used deduct 15 per cent* from the *Btu deliveries.*
[b]Consider each 45 deg elbow as equal to one-half of a 90 deg elbow
[c]Consider the two sharp elbows in a cross-over connection as equivalent to three 90 deg round elbows.
[d]Consider each 90 deg offset between boot and stackhead equal to one 90 deg round elbow.
[e]No. 21 for Btu values multiply 11-22 values by 0 83.
[f]No. 23 for Btu values multiply 12-24 values by 0 83.

TABLE 8. RETURN AIR—CARRYING CAPACITY—BTU SERVICED

UNIT No.	TYPE A BTU PER HOUR	TYPES B AND C BTU PER HOUR	TYPE D BTU PER HOUR	TYPE E BTU PER HOUR
31	11,300	9,500	7,800	5,000
32	16,300	13,700	11,300	7,200
33	22,200	18,700	15,300	9,800
34	29,000	24,400	20,000	12,800
35	36,700	30,800	25,300	16,200
36	45,300	38,000	31,300	20,000
37	54,800	46,000	37,800	24,100
38	65,200	54,800	45,000	28,700

FURNACE CAPACITY

The size of the furnace should be such as will provide the necessary air heating capacity. In the case of gas, oil, and some coal furnaces, the capacity of the unit may be expressed in Btu per hour *at the furnace bonnet*, and this value reduced by 25 per cent should equal or exceed the entire heat load of the building due to heat loss, as well as outside air introduction, or other loads. In the case of most coal furnaces, the capacity is expressed in square inches of leader pipe area, which when

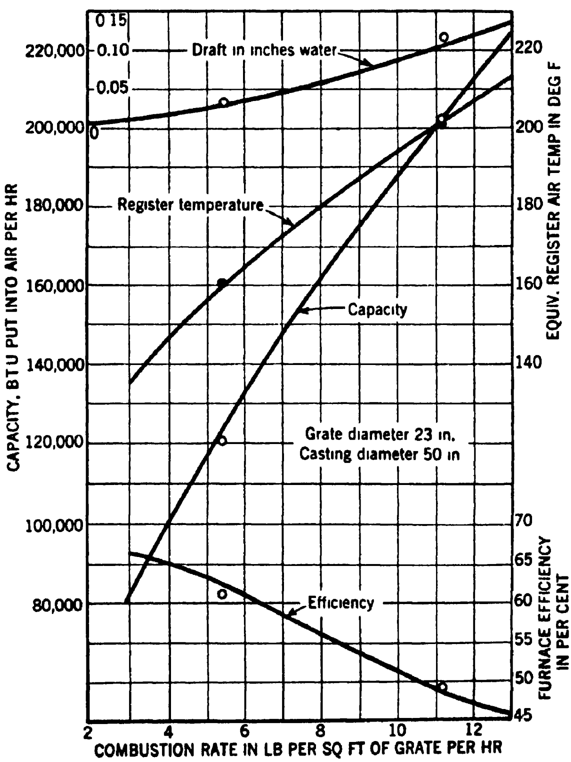


FIG. 5. TYPICAL PERFORMANCE CURVES FOR A WARM-AIR FURNACE AND INSTALLATION IN A THREE-STORY TEN LEADER PLANT, OPERATING ON RECIRCULATED AIR

multiplied by 136 Btu per square inch, indicates the heat delivery *at the register*. The latter figure should equal or exceed the heat loss figure.

The register delivery of a gravity coal-fired furnace may be established by the following equations:

$$H = \frac{G \times p \times f \times E_1 \times E_2 \times 0.866\,[1 + 0.02\,(R\text{-}20)]}{144} \qquad (1)$$

in which:

G = grate area, square inch.
p = combustion rate, pounds coal per square foot of grate per hour.
f = heating value of the coal, Btu per pound.
E_1 = efficiency at bonnet, ratio of heat delivered at bonnet to heat developed in furnace.
E_2 = efficiency of duct transmission, ratio of heat delivered at register to heat delivered at bonnet.
0.866 = factor of safety to allow for contingencies under service conditions such as accumulations of soot and ashes, ineffective firing methods, etc.
H = total heat loss from structure.
R = ratio of heating surface to grate area.

The preceding empirical formula provides a 2 per cent increase in furnace capacity for each unit of value whereby the ratio of actual heating surface to grate area exceeds 20. This addition is based on tests conducted at the University of Illinois on seven types of furnaces having varying ratios of heating surface to grate area. Neither this correction, nor the Standard Code[4] Ratings, apply to values of the ratio less than 15 and greater than 30.

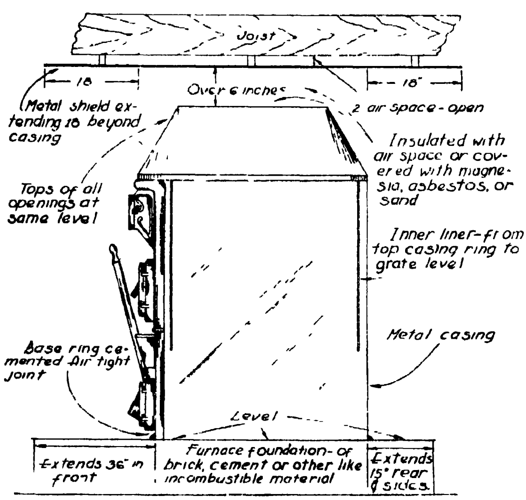


FIG. 6. DETAILS OF FURNACE BONNET, CASING, AND FOUNDATION (FROM STANDARD CODE APPLICATION MANUAL)

[4]Standard Gravity Code for the Design and Installation of Gravity Warm Air Heating Systems in Residences. This code has been sponsored by the *National Warm Air Heating and Air Conditioning Association*, the *National Association of Sheet Metal Contractors*, and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. It is recommended that the installation of all gravity warm-air heating systems in residences be governed by the provisions of this code, the eleventh edition of which may be obtained from the *National Warm Air Heating and Air Conditioning Association*.

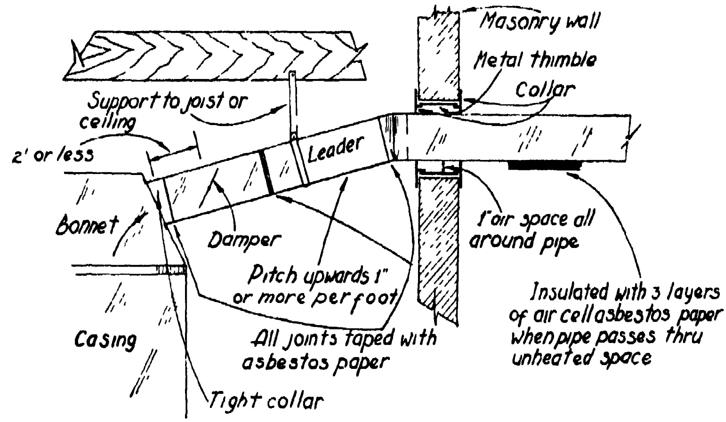


FIG. 7. DETAILS OF BONNET AND LEADER OF GRAVITY WARM-AIR FURNACE (FROM STANDARD CODE APPLICATION MANUAL)

Fig. 5 represents Laboratory data on a typical furnace, illustrating the fact that for a given register temperature, each furnace will have a definite capacity and efficiency, and will require a certain chimney draft to draw the air for combustion through the fuel bed. Test data gathered on the basis of the register design temperature of 175 F, permit rearranging the formula for Equation 1 into the following form:

$$G = \frac{144 \times H}{7.5 \times 12{,}790 \times 0.55 \times 0.75 \times 0.866\,[1 + 0.02\,(R\text{-}20)]} \tag{2}$$

$$G = 0.004205\,\frac{H}{[1 + 0.02\,(R\text{-}20)]} \tag{3}$$

As used in Equations 2 and 3, H equals the hourly heat loss of the entire building, in Btu per hour, including the heat required to warm any outdoor air introduced through a fresh air duct.

Figs. 6 and 7 show recommended practice as given in the *N.W.A.H. &A.C. Assn.* Standard Code Application Manual.

CHAPTER 19

Mechanical Warm Air Furnace Systems

Furnaces, Fans and Motors, Filters, Air Distribution, Automatic Controls, Design of Heating System, Selecting the Furnace, Selecting the Fan, Heavy Duty Fan Furnaces, Humidification, Cooling Methods, Cooling System Design

IN mechanical warm air or fan furnace heating systems[1], the air circulation is effected by motor-driven fans instead of by the difference in weight between the heated air leaving the top of the casing and the cooled air entering the bottom, as in gravity systems described in Chapter 18. The advantages of mechanical systems, as compared with gravity systems, are:

1. The furnace need not be centrally located but may be placed in any part of the basement.

2. Basement distribution piping can be made smaller and can be so installed as to give full head room in all parts of the average basement, or be completely concealed from view where desired.

3. Circulation of air is positive, and in a properly designed system can be balanced in such a way as to give a greater uniformity of temperature distribution.

4. Humidity control is more readily attained.

5. The air may be cleaned by sprays or filters, or both.

6. The fan and duct equipment may be utilized for a complete cooling and dehumidifying system for summer, using either ice, mechanical refrigeration, or low temperature water for cooling and dehumidifying, or adsorbers for dehumidifying.

7. The use of the fan increases the volume of air which can be handled, thereby increasing the rate of heat extraction from a given amount of heating surface and insuring sufficient air volume to obtain proper distribution in a large room.

8. Ventilation air may be positively introduced and heated.

FURNACES

Cast-iron furnaces are usually made in sections and are assembled and cemented or bolted together on the job. Steel furnaces are made with welded or riveted seams. The proper design of the furnace depends largely on the kind of fuel to be burned, and special units are being made for the use of coal, oil and gas. Each type of fuel requires a distinct type of furnace for highest efficiency and economy, substantially as listed herewith:

1. Coal Burning:
   a. *Bituminous*—Large combustion space with easily accessible secondary radiator or flue travel.
   b. *Anthracite or coke*—Large firebox capacity and liberal secondary heating surfaces.
   c. *Stoker firing*—Adequate firebox height, liberal heating surface.

2. Oil Burning:
   a. Liberal combustion space.
   b. Long fire travel and extensive heating surface.

3. Gas Burning:
   a. Low flue resistance.

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[1]Specifications for the furnace unit and the installed duct system are shown in The Yardstick for the Evaluation of a Forced Warm Air Heating System, and in Winter Air Conditioning—Forced Warm Air Heating, by S. Konzo, published by *National Warm Air Heating and Air Conditioning Association*, 145 Public Square, Cleveland, Ohio.

b. Extensive heating surface.
c. Close contact between combustion products and heating surface.

A combustion rate of from 5 to 8 lb of coal per square foot of grate per hour is recommended for residential furnaces. A higher combustion rate is permissible with larger furnaces for buildings other than residences, depending upon the ratio of grate surface to heating surface, firing period, and available draft.

In residential furnaces for coal burning, the ratio of heating surface to grate area will average about 20 to 1; in commercial sizes it may run as high as 50 to 1, depending on fuel and draft. Furnaces may be installed singly, each furnace with its own fan, or in batteries of any number of furnaces, using one or more fans.

Where oil fuel is used, care must be exercised in selecting the proper size and type of burner for the particular size and type of furnace used. The systems are usually designed for blow-through installations so that the furnace is under external pressure. The *National Warm Air Heating and Air Conditioning Association* has prepared a Tentative Code for Testing and Rating of Oil-Fired Furnaces. Compact fan-furnace-burner units are available, suitable for basement, closet, or even attic installations.

Gas fired forced air furnaces should conform in construction and performance to A.S.A. Approval Requirements[2].

Furnace Casings

Casings are usually constructed of galvanized iron or fireproof insulating boards, but they may also be constructed of brick. Galvanized iron casings should be lined with sheet iron liners, extending from the grate level to the top of the furnace and spaced from 1 in. to 1½ in. from the outer casing. Casings for commercial or heavy duty furnaces, if built of galvanized iron, should be insulated with fireproof insulating material at least 2 in. thick. In general, either brick or sheet metal casing should be equipped with baffles to secure impingement of the air to be heated against the heating surfaces. Brick furnace casings should be supplied with access doors for inspection.

Where a fan is to be used with a furnace casing sized for gravity air flow, some form of baffling must be employed to restrict the free area within the casing and to force impingement of the air against the heating surfaces. Where square casings are used, the corners must be baffled.

The hood or bonnet of the casing above the furnace should be as high as basement conditions will allow, to form a plenum chamber over the top of the furnace. This tends to equalize the pressure and temperature of the air leaving the bonnet through the various openings. It is generally considered advisable to take off the warm air pipes from the side of the bonnet near the top, as this method of take-off allows the use of a higher bonnet and thus provides a larger plenum chamber.

FANS AND MOTORS

Centrifugal fans are the type most commonly used, and these may be equipped with either backward or forward curved blades. Motors may be mounted on the fan shaft or outside of the fan for belt drive. Adjustable pulleys are desirable to provide a factor of safety and to allow for increased air circulation, provided the motor is adequate to carry the

[2]American Standard Approval Requirements for Central Heating Gas Appliances Z 21.13 (*American Standards Association*).

load imposed at maximum fan speed. Two-speed motors have given successful operating results and are recommended.

Special attention should be given to the problem of noise elimination. The metal duct connection to and from the furnace casing and fan housing should be broken by strips of canvas. Motors and their mountings must be carefully selected for quiet operation. Electrical conduit and water piping must not be fastened to, nor make contact with fan housing. The installation of a fan directly under a cold air grille is usually not recommended. (See also Chapter 32.)

FILTERS

Several types of filters are available for mechanical warm air furnace applications and are discussed in Chapter 28. For maximum efficiency and life under operating conditions, filters should not be subjected to a temperature in excess of 150 F. Filters should have at least 80 per cent average efficiency on an 8 hr test. Filter resistance rises rapidly with the accumulation of dirt, and may reduce the air circulation over heating surfaces.

For domestic furnaces, the maximum velocity should not exceed 300 fpm based on nominal filter area.

AIR DISTRIBUTION

The conditions of comfort obtained in a room are greatly influenced by the type of register used and the locations of the supply registers and return grilles. In general it has been found that changes in the type, air velocity, and location of the supply register affect the room conditions much more than the changes in the location of the return grilles. One method is to locate the supply register near the floor so that the warm air from the register *blankets* a cold wall, and mixes with the cold air dropping off from the exposed walls and glass. Another method is to locate the supply openings near the floor on the inside wall and the return openings near the greatest outside exposure. Tests in the Warm Air Research Residence[3] have indicated that continuous fan operation provided better results than intermittent fan operation.

Register and Grille Openings

Supply registers located in the floor require attention to keep them clean and are usually avoided.

Tests conducted in the Warm Air Research Residence[4] have indicated that comparable results are obtainable with either high side wall or baseboard registers, providing proper registers and air velocities are selected. Baseboard registers should be of a deflecting-diffuser type which throw the air downward toward the floor and diffuse it at the same time. For baseboard registers air temperatures under 125 F and air velocities over 500 fpm should be avoided as they may cause drafts.

High side wall registers must be of such type that the air is delivered horizontally or in a slightly downward direction, and must be so located as to avoid impingement of air on ceiling or wall. Directional flow diffusing type should be used to insure best results. Register air velocities should be such that the air stream carries to the opposite exposure. Velocities under 500 fpm are not recommended. In general better air

[3]Performance of a Forced Warm-Air Heating System as Affected by Changes in Volume and Temperature of Air Recirculated, by A P. Kratz and S. Konzo (A.S H.V E. TRANSACTIONS, Vol. 48, 1942, p 393)

[4]Loc. Cit. Note 3.

and temperature distribution is obtainable by using baseboard registers for heating and high side wall registers for cooling.

Registers should be well proportioned and decorated to harmonize with the trim. Air supply registers should be equipped with dampers and all registers should be sealed against leakage around the borders or margins. The register sizes shown in Table 1 have been recommended as standard by the *National Warm Air Heating and Air Conditioning Association.*

Velocities through registers may be reduced by the use of registers larger than the connecting ducts. Diffusers should be used to spread the air uniformly over the register face.

Return air grilles may be located in hallways, near entrance doors, under windows, in exposed corners, or inside walls, depending on location of supply registers. Baseboard returns are preferable to floor grilles.

Dampers

Suitable dampers for air direction or volume control are essential to any trunk or individual duct system. Special care must be used in the design

TABLE 1. RECOMMENDED REGISTER SIZES

Warm Air Registers, In. Sidewall and Baseboard Types				Return Air Grilles or Intakes, In. Sidewall and Baseboard Types			
8 x 4 10 x 4	10 x 5	8 x 6 10 x 6	10 x 8	8 x 4 10 x 4	10 x 5	8 x 6 10 x 6	10 x 8
12 x 4 14 x 4	12 x 5 14 x 5	12 x 6 14 x 6	12 x 8 14 x 8	12 x 4 14 x 4	12 x 5 14 x 5	12 x 6 14 x 6	12 x 8 14 x 8
				24 x 4 30 x 4	24 x 5 30 x 5	24 x 6 30 x 6	

of any system to avoid turbulence and to minimize resistance. Sharp elbows, angles, and offsets should be avoided. Three types of dampers are commonly used. *Volume dampers* are used to completely cut off or reduce the flow through pipes. *Splitter dampers* are used where a branch is taken off from a main trunk. *Squeeze dampers* are used for adjusting the volume of air flow and resistance through a given duct. It is essential that a damper with positive locking device be provided for each main or duct branch. Labels placed on ducts should indicate the room being served. Damper positions should be marked for summer and winter operation, and to avoid tampering.

Ducts

The ducts may be either round or rectangular in cross section. The radii of elbows should preferably be not less than one and one-half times the pipe diameter for round pipes, or the equivalent round pipe size in the case of rectangular ducts. Warm air ducts passing through cold spaces, or where located in exposed walls, should have ½ to 2 in. of insulation.

AUTOMATIC CONTROLS

Air stratification, high bonnet temperatures, excessive flue gas temperatures, and heat overrun or lag in a properly designed system can be

largely eliminated through proper care in the planning and installation of the control system[5].

Controls which are considered desirable for this system are:

1. A *thermostat* located in a living room where maximum fluctuation in temperature can be expected, in order to secure frequent operation of fans, drafts, and burners. The thermostat location should not be on an outside wall, in a bed room, bath room or sun room, or in a location where it will be affected by direct radiant heat from the sun or from a fireplace, or by direct heat from any warm air duct, register or chimney.

2. A *fan switch* control located in the bonnet to start blower operations at temperatures between 120 and 150 F, and to stop the blower at about 25 to 30 F below the cut-in point. The lower settings are used for high side wall register installations, and the higher settings for baseboard register installations. For most satisfactory results these settings should be as low as is feasible.

3. A *protective high limit control* located in the bonnet to stop the system independently of the thermostat if the bonnet temperature exceeds 200 F.

4. On oil and gas burner installations, a protective control should be included which will stop the system if the fire is extinguished or if there is a failure of the ignition system.

5. On automatic stoker installations, a control is usually included which will start the operation regardless of thermostat settings whenever the bonnet temperature indicates that the fire is dying, or a time interval contactor is used that will start the stoker to run a few minutes out of each hour.

Table 2. Factors Corresponding to Register Temperature for Equation 2

Register Temperature Deg F	Factor
110	0.0221
120	0.0184
130	0.0158
140	0.0140
150	0.0125
160	0.0114
170	0.0105

6. A *humidistat* to regulate the moisture supplied to the rooms, located either in one of the rooms or in the main return duct near the furnace.

METHOD OF DESIGNING FORCED-AIR HEATING SYSTEMS[6]

1. Determine heat loss from each room in Btu per hour. (See Chapter 6.)

2. Locate warm air registers and return registers on plans of house, beginning with the upper story rooms.

3. Sketch in duct layout to connect all registers and grilles with the central unit.

4. Determine equivalent length of duct for each register, allowing at least 10 diameters of straight pipe as equivalent to each 90 deg elbow having an inner radius not less than the diameter of the round pipe or the depth of the rectangular pipe.

5. Select a value for temperature of the air at the furnace bonnet. It is customary to use some value between 145 to 165 F. Use lower value if larger number of air recirculations is desired. The number of air recirculations should range from three to eight per hour.

6. Determine approximate value of temperature reduction in each duct caused by heat loss from the ducts. A value of from 0.3 to 0.6 F per foot of duct has been obtained from tests conducted in the Research Residence installation for uninsulated duct lengths up to approximately 60 ft.

[5]Automatic Controls for Forced-Air Heating Systems, by S. Konzo and A. F. Hubbard (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 37).

[6]Technical Code, Fourth Edition, January 1, 1942, published by the *National Warm Air Heating and Air Conditioning Association* 145 Public Square, Cleveland, Ohio.

7. Subtract this temperature reduction from the assumed bonnet air temperature to obtain an approximate value of the register air temperature for each register.

8. Determine the required air volume for each room from Equation 1, or from the values listed in Table 2:

$$Q = \frac{H}{60 \times 0.24 \times d\ (t_r - 65)} \tag{1}$$

where

Q = required air volume, cubic feet per minute.
H = heat loss of room, Btu per hour.
d = density of air at register temperature, pounds per cubic foot.
t_r = register temperature, degrees Fahrenheit.
0.24 = specific heat of air.
65 = return air temperature, degrees Fahrenheit.

For any given register temperature the solution of this equation simplifies to:

$$Q = H \times \text{Factor} \tag{2}$$

in which the values of the Factor may be obtained from Table 2.

9. Determine register size from the air volume delivered to each room:

TABLE 3. APPROXIMATE DESIGN VELOCITIES THROUGH DUCTS AND REGISTERS

DESCRIPTION	LOW VELOCITY SYSTEM (FPM)	MEDIUM VELOCITY SYSTEM (FPM)	HIGH VELOCITY SYSTEM (FPM)
Main ducts	500	750	1000
Branch ducts	450	600	750
Wall stacks	350	500	600
Baseboard registers (Down deflecting)	300	400	500
Wall registers above 5 ft (min.)	500	550	600

$$\text{Gross area of register, square feet} = \frac{Q}{VR} \tag{3}$$

where

Q = required air volume, cubic feet per minute.
V = velocity at register face, feet per minute.
R = ratio of free area to gross area of register.

Allowable register velocities to be used in Equation 3 are given in Table 3.

In residential applications it is not advisable to handle more than 150 cfm through any single register.

10. Duct systems for forced-air installations may consist of either trunk systems or individual duct systems.

Trunk Systems. Determine duct sizes and friction losses as outlined in Chapter 31, except that for residence applications the velocities in the main duct and in the various parts of the system should approximate the values recommended in Table 3.

Individual Duct Systems. An individual duct system is one having separate ducts extending from the heating unit to each register. In designing such a system select first the duct having the greatest equivalent length. Select a reasonable velocity using Table 3 as a guide. From friction chart in Chapter 31 determine unit friction loss per 100 ft of run, and from this the total friction loss in the duct selected. If this total friction loss exceeds a reasonable value a lower velocity should be used.

The remaining ducts are proportioned so that the total pressure in each duct is the same as that calculated for the longest duct. The added resistance necessary in the shorter ducts is accomplished by increasing the velocity in these ducts. No duct should be less than 6 in. in diameter, nor should the velocity in any duct exceed approximately 1200 fpm. The final adjustment in a duct system may be made by employing dampers.

Instead of proportioning the ducts as outlined in the preceding paragraph it is more usual in practice to proportion all the ducts so that they have the same velocity as that used in the longest duct and to balance the system by employing dampers in the shorter ducts.

Return duct systems are designed making use of the same principles as those used in the design of supply duct systems. In this case the design may be based on the volume of air corresponding to the density of air existing in the return ducts, or in order to provide a factor for air leakage, it may be based on the same volume as used for the supply ducts.

11. Determine frictional resistance in:

a. Supply side of system as outlined in Item 10.

b. Return side of system as outlined in Item 10.

c. Furnace units, casing or hood, which is usually considered as equivalent to 0.03 to 0.10 in. of water.

d. Accessories such as washers or air filters, from manufacturer's data.

e. Inlet and outlet registers and grilles[7].

f. Other accessory equipment such as cooling coils, from manufacturer's data.

The requirements in the Tentative Code for Testing Oil-Fired Fan-Furnace Units give static pressure loss requirements, *external* to the units as:

0 to 800 cfm = 0.12 in.
800 to 1600 cfm = 0.20 in.
1600 to 3000 cfm = 0.24 in.
3000 to 6000 cfm = 0.30 in.

Choose a fan which, according to its manufacturer's rating, is capable of delivering a volume of air, expressed in cubic feet per minute, against a frictional resistance, expressed in inches of water, computed by adding together the items listed in the preceding discussion. In practice it is recommended that liberal allowances should be made so that the fan will be capable of delivering air against pressures that may not have been foreseen during the design of the duct system.

12. Select a furnace capable of delivering heat at the register outlets equal to the total heat loss of the structure to be heated. Equation 4 may be used for coal burning furnaces:

$$G = \frac{H}{f \times p \times E_1 \times E_2 \,[1 + 0.02\,(R - 20)]} \tag{4}$$

where

G = required grate area, square feet.
H = total heat loss from building, Btu per hour.
f = calorific value of coal, Btu per pound.
p = combustion rate, pounds of fuel per square foot of grate per hour.
E_1 = furnace efficiency based on heat available at bonnet.
E_2 = efficiency of transmission based on ratio of heat delivered at register to heat available at bonnet.
R = ratio of heating surface to grate area.

Equation 4 is not applicable to ratios less than 15 to 1. Higher ratios as a rule provide lower flue gas temperatures and higher efficiencies. In practice it is customary to use these constants: f = 12,000 (for specific values, see Table 5, Chapter 8); p = 7.5 lb; E_1 = 0.65 (lower efficiency must be used with highly volatile solid fuel); and E_2 = 0.85.

The foregoing procedure for determining the size of the furnace to be used applies to continuously heated buildings.

13. Although intermittently heated buildings usually have their heat losses computed according to the standard rules for determining such losses, these rules do not take into account the heat which will be absorbed by the cold material of the building after the air is raised in temperature. This heat absorption must be added to the normal heat loss of the building to determine the load which the heating plant must carry through the warming-up process. It is customary to increase the normal heat loss figure by from 50 to 150 per cent depending upon the heat capacity of the construction material, the higher percentage applying to materials of high heat capacity such as concrete and brick.

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[7]Pressure Losses in Registers and Stackheads in Forced Warm Air Heating, by A. P. Kratz and S. Konzo (University of Illinois, *Engineering Experiment Station Bulletin* No. 342).

14. Follow the same methods for an oil furnace as for coal where a conversion unit is to be used, making sure that the ratio of heating surface to grate area exceeds 20 to 1. If it does not, a size larger furnace should be selected. Use the manufacturer's Btu ratings of furnaces designed for exclusive use with oil.

15. The selection of the proper size gas furnace for a constantly heated building can be easily made by using Equation 5:

$$R = \frac{H}{0.85} \tag{5}$$

where

H = total heat loss from building, Btu per hour.

R = output rating of the furnace, Btu per hour.

In the case of converted warm air furnaces a slightly different procedure is necessary, as the Btu input to the conversion burner must be selected rather than the furnace output. The proper sizing may be done by means of Equation 6:

$$I = 1.68\,H \tag{6}$$

where I = Btu per hour input.

The factor 1.68 is the multiplier necessary to care for a 15 per cent heat loss in the distributing ducts and an efficiency of 70 per cent in the conversion burner.

16. Specify location and type of all dampers in both supply air and return air sides of system. Specify controls including location of all thermostats. Arrange for proper control of humidifying equipment.

HEAVY DUTY FAN FURNACES

Fan furnaces for large commercial and industrial buildings, churches, schools, etc., are available in sizes ranging from 300,000 to 3,000,000 Btu per hour per unit. Heavy duty furnace heaters may be arranged in battery combinations of one or more units.

Most manufacturers of heavy duty furnaces rate their furnaces in Btu per hour and also in the number of square feet of heating surface. Conservative practice indicates that at no time in the heating-up period should the furnace surface be required to emit more than an average of 3500 Btu per square foot. A higher rate of heat emission tends to increase the heat loss up the chimney, and raise fuel consumption, to shorten the life of the furnace, and to overheat the air. The ratio of heating surface to grate area on furnaces for this type of work should never be less than 30 to 1 and as indicated previously may run as high as 50 to 1.

Control of temperature is secured through (1) controlling the quantity of heated air entering the room, (2) using mixing dampers, or (3) regulating the fuel supply.

The design of heavy duty fan furnace heating systems is in many respects similar to that of the central fan heating systems described in Chapter 20. Ducts are designed by the method outlined in Chapter 31.

HUMIDIFICATION

Temperatures and relative humidities should be governed within the limits of the generally accepted standards. (See Chapters 2 and 26.)

Water evaporating pans are usually located in air which has been heated by contact with the heating surfaces. To change water into vapor capable of being carried in an air stream as part of the mixture, about 1000 Btu per pound are required. There is a trend in present practice toward heating the water in addition to heating the air. Equipment for doing this may make use of sprays, or it may take the form of water circulating coils placed within the combustion chamber and connected by pipes to the humidifier pans where a constant water level is maintained by some separate float device. (See Chapter 26.)

Sprays are usually controlled by solenoid valves wired in parallel with the fan motor. The water supply may, in turn, be controlled by a humidity-controlling device located in one of the living rooms, so that the washer will operate at all times when the fan is in operation, unless the relative humidity should rise beyond a desirable percentage. Sprays used in connection with commercial or heavy duty plants should be a regulation type of commercial spray. In all cases provision must be made to flush out accumulation of lime and dirt.

Principles underlying humidity requirements and limitations for residences have been summarized in a bulletin[8] and are enumerated herewith:

1. Optimum comfort is the most tangible criterion for determining the air conditions within a residence.

2. An effective temperature of 66 deg[9] represents the optimum comfort for the majority of people. Under the conditions in the average residence a dry-bulb temperature of 72 F with relative humidity of 30 per cent is the most practical for the attainment of 66 deg effective temperature.

3. Evaporation requirements to maintain a relative humidity of 40 per cent in zero weather depend on the amount of air inleakage to the average residence, and vary from practically nothing to 24 gal of water per 24 hours.

4. Relative humidity of 40 per cent indoors cannot be maintained in rigorous climates without excessive condensation on the windows unless tight-fitting storm sash or the equivalent is installed.

5 The problems of humidity requirements and limitations cannot be separated from building construction, and the latter should receive serious attention in the installation of humidifying apparatus.

6. None of the types of gravity warm air furnace water pans tested proved adequate to evaporate sufficient water to maintain 40 per cent relative humidity in the Research Residence except in moderately cold weather.

7. The water pans used in the radiator shields tested did not prove adequate to maintain 40 per cent relative humidity in a residence similar to the Research Residence when the outdoor temperature approximated zero degrees Fahrenheit.

COOLING METHODS

A slight cooling effect may be obtained under certain conditions by the use of the cooler basement air. A more positive cooling effect may be obtained by the use of an air washer where the temperature of the city or well water is sufficiently low (55 F or lower), and where a sufficient volume of water can be provided. Unless the temperature of the leaving water is below the dew-point temperature of the indoor air at the time the washer is started, both the relative and absolute humidities will be somewhat increased.

Coils of copper finned tubing through which cold water is pumped are available for cooling. They require less space than air washers and have the advantage that no moisture is added to the air when the temperature of the water rises above the dew-point. Ample coil surface and fan capacity are necessary with this type of cooling.

It is thoroughly feasible to use ice or mechanical refrigeration in connection with a warm air system and to cool the building by this method, provided the building is reasonably well constructed and insulated. Windows and doors should be tight, and awnings should be supplied on the sunny side of the building. (See also Chapters 20, 23 and 24.)

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[8]Humidification for Residences, by A. P. Kratz (University of Illinois, *Engineering Experiment Station Bulletin* No. 230).

[9]The optimum winter effective temperature is 66 deg as recommended by the A.S.H.V.E. Committee on Ventilation Standards. (See Chapter 2.)

Conclusions drawn from studies[10] conducted in the University of Illinois Research Residence, subject to the limitations of the test are:

1. An uninsulated building of ordinary residential type may require the equivalent of three tons of ice in 24 hours on days when the maximum outdoor temperature reaches 100 F if an effective temperature of approximately 72 deg is maintained indoors.

2. The use of awnings at all windows in east, south, and west exposures may result in savings of from 20 to 30 per cent in the required cooling load.

3. The cooling load per degree difference in temperature is not constant but increases as the outdoor temperature increases.

4. The heat lag of the building complicates the estimation of the cooling load under any specified conditions and makes such estimates, based on the usual methods of computation, of doubtful value.

5. The seasonal cooling requirements are extremely variable from year to year, and the ratio between the degree-hours of any two seasons occurring within a 10-year period may be as high as 7.5 to 1. Hence an average value of the degree-hours cooling per season is comparatively meaningless.

6. The duct system in a forced-air heating installation can be successfully converted to a system for conveying cool air for the purpose of cooling the structure. No condensation of moisture was observed when the duct temperatures were not less than 65 F.

7. Cooling by means of water at a temperature of 60 F is not satisfactory unless an indoor temperature of less than 80 F is maintained.

8. In the selection of cooling coils, the additional frictional resistance of the coil to flow of air must be given consideration.

9. Cooling the structure by introducing large quantities of air from outdoors at night tended to reduce the amount of cooling required on the following day and was a practical means of providing more comfortable conditions in those homes where cooling systems were not available.

METHOD OF DESIGNING COOLING SYSTEM

The general procedure which may be used for the design of a summer cooling system in a forced-air installation is:

1. Calculate heat gain for each room or space to be conditioned. (See Chapters 4 and 7.) Allowance for addition of outside air must be included in this calculation.

2. Select a temperature of air leaving supply inlets. In Research Residence tests a value of from 65 to 70 F was found satisfactory.

3. Determine indoor conditions to be maintained. In Research Residence 80 F dry-bulb and 45 per cent relative humidity were found satisfactory.

4. Determine the quantity of air to be introduced into each room. (See Chapter 20)

5. Estimate heat loss in duct system between cooling unit and supply registers.

6. Calculate the sensible and latent heat to be removed by the cooling unit.

7. Determine size of ducts in duct system and size of registers, as explained in this chapter under the heading of Method of Designing Forced-Air Heating Systems.

8. Determine pressure loss in duct system and select fan as also explained in the same section.

9. Select cooling unit from manufacturer's data. Specify temperature and pressure of available cooling water, voltage and characteristics of electrical supply, and method of control of apparatus.

10. Select cooling coils from manufacturer's data to take care of latent heat load and to give required drop in air temperature with the weight of air flowing. (See Chapter 25.)

11. If system is to be used for both winter heating and summer cooling, duct sizes must be checked to insure that velocities and friction losses are reasonable for both conditions of operation. Adjustable dampers will be necessary to make changes in air distribution for the two seasons. Provision must also be made for changing fan speeds for summer and winter operation.

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[10]Summer Cooling in the Research Residence, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick (University of Illinois *Engineering Experiment Station Bulletins* Nos 290, 305 and 321). A.S.H.V.E. RESEARCH REPORT No 1177—Summer Cooling in the Research Residence with a Gas-Fired Dehydration Cooling Unit, by A. P Kratz, S. Konzo and E. L. Broderick (A S H V.E. TRANSACTIONS, Vol. 47, 1941, p. 203).

CHAPTER 20

Central Systems for Comfort Air Conditioning

Types of Systems for Ventilating, Heating, Air Conditioning, Factors Involved in Use and Design of Systems, Design Procedure

THE purpose of this chapter is to present a discussion of types of central systems usually encountered, together with a discussion of the factors involved in use and design and an outline of design procedure.

Insofar as this chapter is concerned, a central system is defined as a field assembled apparatus, comprising such elements of equipment as are necessary to fulfill the purpose for which it is designed, and serving one or more conditioned spaces. A factory produced unit, including all the essential items of equipment may be employed as a central system. Unitary equipment is discussed in Chapter 22. Further, this chapter is confined to comfort air conditioning systems as such, and ventilating systems, warm air heating systems, together with central systems of a special nature, are excluded from the discussion.

This chapter assumes a knowledge of all the component parts of a system and the reader is referred specifically to other chapters covering design conditions and physiological principles, cooling and heating load; spray equipment, heat transfer surface coils, cooling, dehumidification and dehydration, fans, air cleaning devices, refrigeration, air distribution and air duct design, automatic controls and instruments. In addition, the engineer should refer to the Code of Minimum Requirements for Comfort Air Conditioning[1] prepared by the joint committee of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *American Society of Refrigerating Engineers*, and to national, state or local codes that may apply.

CLASSIFICATION OF SYSTEMS

The generally accepted method of classifying systems is with regard to their function. A given type of system may be changed by the omission of certain of its functions or by the inclusion of others. As an example, a winter air conditioning system, by the omission of humidifying sprays, air cleaning devices, etc., will become a simple warm air heating system, and by further omissions will become a simple ventilating system. On the other hand, with the inclusion of a dehumidifier with its source of cooling, the system can become a year 'round air conditioning system. The three major types of systems are:

1. *Winter Air Conditioning Systems* The function is to ventilate, heat and humidify in winter the spaces under consideration, and provide the desired degree of air motion and air cleanliness. The equipment required normally consists of a preheater, filters, humidifying sprays or air washer, reheater, fan, distributing ducts, and the necessary manual or automatic means of control. See Fig. 1.

2. *Summer Air Conditioning Systems*. The function is to ventilate, cool and dehumidify the spaces under consideration and to provide the desired degree of air motion and cleanliness. The normal complement of equipment includes filters, dehumidifier with its source of cooling, reheaters or by-pass if required, fan, distributing ducts, and the necessary manual or automatic means of control. See Fig. 2.

3. *Year 'Round Air Conditioning Systems*. The function is to ventilate, heat and humidify in winter and cool and dehumidify in summer the spaces under consideration,

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[1]Code of Minimum Requirements for Comfort Air Conditioning (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 27). Reprints of this code are available at $ 10 a copy.

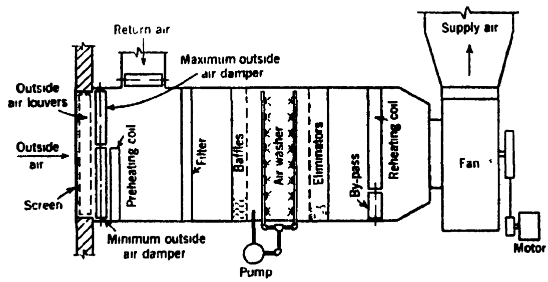

FIG 1. CENTRAL SYSTEM FOR WINTER AIR CONDITIONING

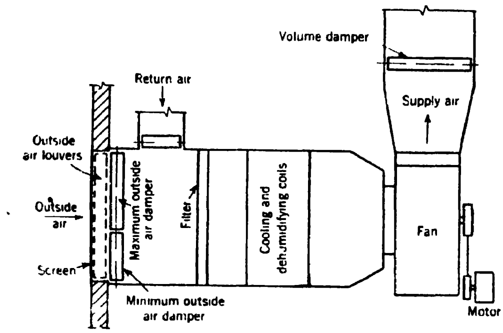

FIG 2. CENTRAL SYSTEM FOR SUMMER AIR CONDITIONING

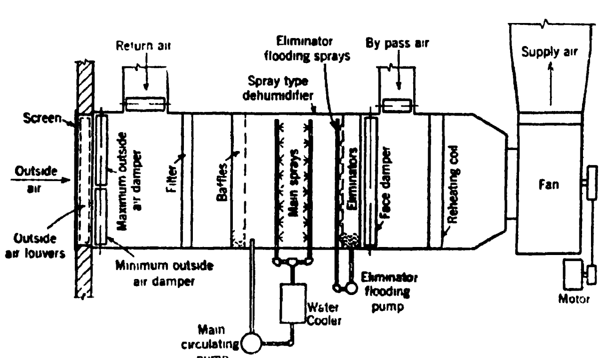

FIG. 3. CENTRAL SYSTEM FOR YEAR 'ROUND AIR CONDITIONING

and to provide the desired degree of air motion and cleanliness. The equipment usually comprises preheater, filters, spray or surface dehumidifier, reheaters and by-pass if required, fan, system of distributing ducts, and necessary means of manual or automatic control. See Fig. 3.

Items of equipment in the foregoing may, of course, be replaced with others fulfilling the same purpose. As an example, an absorption type dehydrator with an aftercooler may replace a dehumidifier.

Modifications

All of the general types of central systems may be modified in various ways. These modifications do not affect the functions of the system. In general, these consist of changes in or additions to the normal complement of equipment, or variations in arrangement and design of equipment and distributing ductwork in order to provide better control of conditions, greater flexibility, or improve the over-all economy and performance of the system. Some of these are applicable to certain systems only, while others are applicable to all types. The most commonly encountered modifications are:

1. Zoning.
   - *a.* Separate equipment.
   - *b.* Reheating or recooling. (See Figs. 4 and 5).
   - *c.* Multiple fans with individual by-pass or reheat (See Figs. 6 and 7).
   - *d.* Volume control. (See Fig. 8).
   - *e.* Dual duct system. (See Fig. 9).
2. Induction units. (Low pressure type). (See Fig. 10).
3. Induction units. (High pressure type). (See Fig. 11).
4. Evaporative cooling.
5. Precooling.
6. Sensible cooling with dry coils.
7. The run-around system.

There are many other possible modifications and the special requirements of some installations may warrant consideration of these, but space limitations prevent a discussion of all. The modifications mentioned are discussed further in this chapter.

USE OF CENTRAL SYSTEMS

Several factors must be considered in deciding on whether or not to use a central system and in deciding on the type of central system and modifications required. These factors are:

1. Comparative effectiveness.
2. Characteristics and requirements of the load.
3. Space requirements.
4. Initial cost.
5. Operating costs and maintenance.

The comparative effectiveness of the type of system and modification plays a major part in the choice and is to a large extent affected by the other factors. One great advantage provided by the central system lies in its ability to diffuse odors and smoke which may occur in parts of the system, so that the outside air is determined by the average instead of the sum of the peak requirements. However, caution must be used where odors are apt to be objectionable even if greatly diluted. In such cases a positive exhaust to the outdoors or a separate treatment for the particular locality is recommended. The *averaging ability* both as to odors and

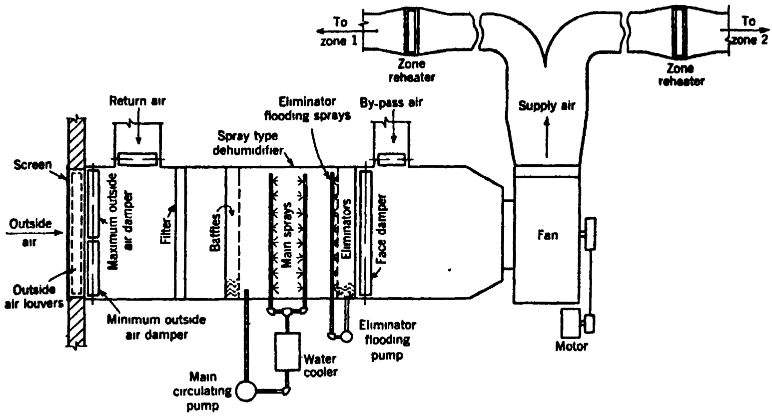

FIG. 4. CENTRAL SYSTEM WITH ZONING BY REHEATING

thermal effects of the system is one item to be considered in studying the comparative effectiveness, particularly where adequate zoning is to be provided. Full advantage of diversity and non-simultaneous peak load requirements can be taken in determining the dehumidified air quantity where adequate zoning is provided.

The characteristics and requirements of the load frequently are the deciding factors. Wide, non-simultaneous variations in load between spaces or parts of the same space indicate the necessity of zoning. Isolated spaces having a short time occupancy or brief load duration may be handled by units advantageously at times. The occurrence of simultaneous heating and cooling requirements in spaces having the same exposure or on the same zone presents a problem to be studied. Ability to maintain conditions during the intermediate seasons without the use of refrigeration must be considered at all times, particularly in those applications where a high internal load exists.

The matter of space requirements may rule out one type of system or another. The avoidance of the use of rentable and usable space for equipment and ductwork, in office buildings, stores, etc., is most im-

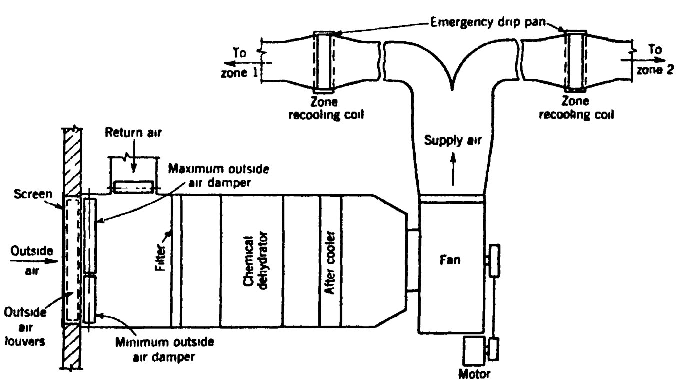

FIG. 5. SUMMER CENTRAL SYSTEM WITH ZONING BY RECOOLING

portant since the loss of revenue is directly chargeable to the operating cost of the system. Consideration of the spaces available with respect to the type of system and method of distribution used and their effect on over-all cost is essential.

Initial cost is given first consideration all too frequently. While elaborately designed installations are seldom justifiable, proper consideration must be given to operating costs and maintenance, performance required, and the life of the system. The increase in cost incurred by suitable zoning can be offset partially by reduced quantities of dehumidified air, and partially by reduced refrigeration requirements. In the end it may prove less expensive than an unsatisfactory single zone system. A small increase in initial cost to provide better access to equipment, better airflow and distribution, and better zoning, usually will pay for itself.

Operating costs often receive too little attention. Proper relationship between equipment selected and the load to be carried must be obtained.

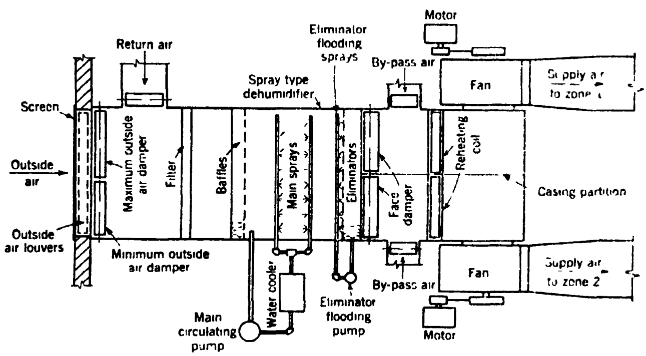

Fig. 6. Central System Using Multiple Fans with By-passes and Reheating for Zoning

Cooling systems in general operate at maximum capacity less than 20 per cent of the time and heating systems operate under design conditions but a few days in the year. Good partial load performance is essential for low operation costs. Proper zoning, good arrangement and selection of equipment and type of system all tend to reduce operation costs. Central systems having the bulk of equipment in a central location properly arranged, and having only the equipment required for zoning distributed through the conditioned space or spaces, will have relatively low maintenance costs. Care must be used to provide ease of access to important equipment and those items requiring servicing.

DESIGN OF SYSTEMS

Various factors are to be considered in the design of a central system, some of which have been touched upon in the foregoing as a modification or economic factor, while others are a part of the normal design procedure. These, in the order of usual occurrence are discussed herewith. For practical purposes these factors cover *Year 'Round Air Conditioning Systems*. Those directly applicable to summer only or winter only systems are to be viewed accordingly.

Design Conditions

Physiological principles and design conditions are covered in Chapter 2. A detailed study of these is beyond the scope of this chapter other than certain recommendations with regard to their application.

Where extreme or unusual outdoor conditions prevail for long periods of time, such as in the tropics, at high altitudes, in regions of extremely low humidities, etc., due allowance must be made for the fact that the people have become acclimatized, to a certain degree at least, to these conditions and the inside conditions should be selected accordingly.

A change in the inside conditions from a set standard sometimes is warranted from an economic standpoint. It is possible at times to make substantial savings in initial and operating costs by maintaining a lower

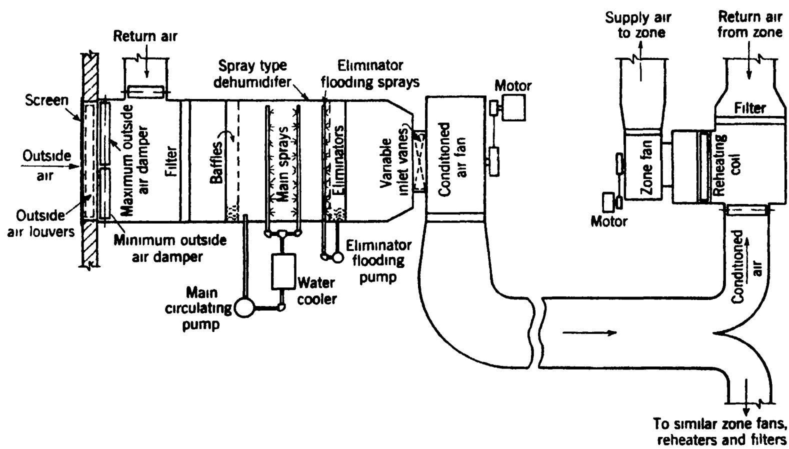

FIG. 7. CENTRAL SYSTEM WITH CENTRAL FAN AND CONDITIONER AND WITH INDIVIDUAL ZONE FANS

temperature and higher humidity in the conditioned space while maintaining the same effective temperature.

In winter, the matter of condensation on windows, walls, etc., is of extreme importance. Humidities low enough to avoid this should be maintained, and when it is necessary to carry the higher humidities, double glass should be used or suitable means of handling the condensation should be provided.

During intermediate seasons the use of refrigeration as a source of cooling may be undesirable from an operating cost standpoint depending on the local energy rate structure and demand charges. As a result the use of outdoor air, either directly or indirectly, as a source of cooling may be required and the system may have to operate on an unusual basis with regard to the temperatures and humidities that can be maintained. This situation should be carefully investigated. As a rule, provision should be made for introducing all outdoor air into the system during intermediate seasons.

With regard to outside conditions it must be noted that where systems are to operate at certain hours of the day only, the outside conditions that

occur during these times can be used provided the cumulative effect or heat gain lag is taken into consideration in the estimate.

Outdoor Air

Standards affecting the quantity of outdoor air have been established in Chapter 2. These standards relate the minimum amount of outside air to be introduced into the conditioned space for both the number of occupants and the type of occupancy, *i.e.*, people smoking, etc. This, of course, is the common-sense approach to the problem, but there are some cases, such as those spaces having a very low occupancy with regard to the cubical content where mustiness may develop unless a sufficient air change is provided. Where such a condition exists in a few spaces out of several which are being handled by a central system, the *averaging* ability of the central system may cope with the situation satisfactorily. This is

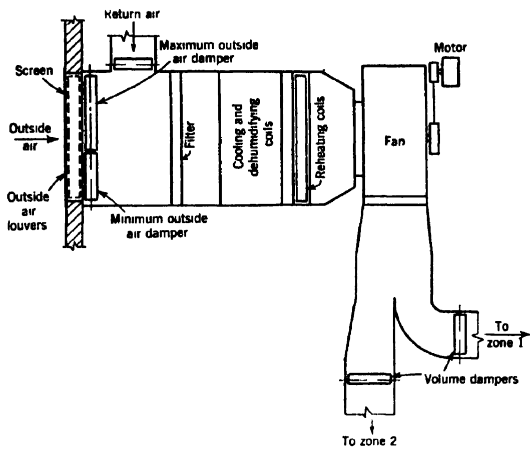

FIG. 8. CENTRAL SYSTEM WITH ZONING BY VOLUME CONTROL

due to the fact that the return air from the particular space is mixed thoroughly with the return air from all the other spaces and all the outside air supplied.

It should be noted that the minimum quantity of outdoor air is affected by infiltration and leakage. Infiltration will reduce the quantity to be introduced by the system while leakage may have to be offset by an increase in the quantity of outdoor air.

Most summer and year 'round air conditioning systems should be so designed and arranged that a quantity of outside air at least equal to the quantity of cooled or dehumidified air can be taken in when desired. In the intermediate seasons and in the cooling season it is more economical to take all outside air into the dehumidifier when the outdoor air wet-bulb temperature is lower than the inside wet-bulb temperature to be maintained. Also, where using a spray type dehumidifier or air washer, whenever the outdoor wet-bulb temperature is below the apparatus dew-point temperature required, the system can be operated on an evaporative cooling basis, dispensing with the need for refrigeration though cooling may be required. Automatic controls are available for accomplishing this and on reasonably large systems are justifiable from an economic standpoint.

Cooling Load

The method of determining the cooling load for a conditioned space or spaces is outlined in Chapter 7. As pointed out therein, many of the items of heat gain are variable and do not reach their maximum values simultaneously. Proper consideration of these *peaks* and the avoidance of pyramiding these peaks in the cooling load calculations are stressed. Maximum solar heat gain on an east exposure is seldom coincident with the maximum outdoor wet-bulb.

A large difference in the incidence of the peaks between various spaces or parts of the same space indicates the necessity for zoning. In a building having an east and west exposure where solar heat gain forms a fair share of the cooling load, the times of their individual peaks are apt to be hours apart, and the peak load of one plus the off peak load of the other will be

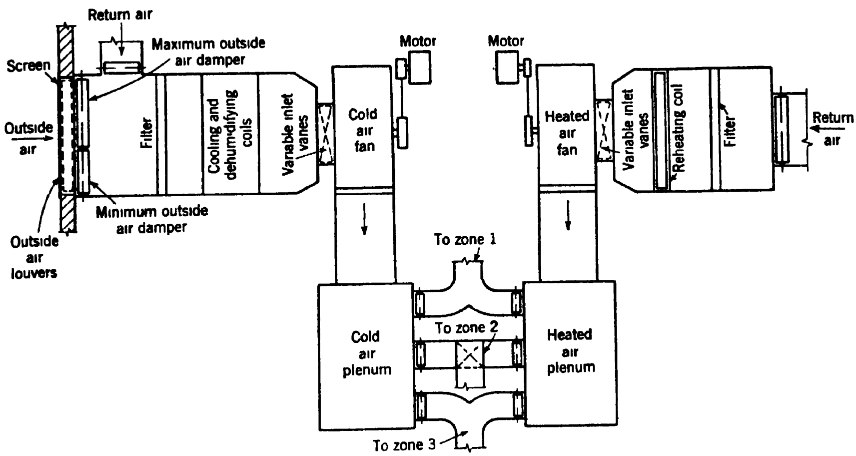

FIG. 9. CENTRAL SYSTEM USING DUAL DUCT METHOD OF ZONING

substantially less than their combined peak loads. Proper zoning will permit taking full advantage of this condition or similar conditions of non-simultaneous peaks and will result in a lower total load, reflecting itself in savings in equipment.

A factor, similar in effect and closely related to the non-simultaneous occurrence of peak loads is *diversity*. Typical of this is the case of a large department store where the air handling equipment serving a certain space must be sufficient to handle the load created by the throngs of people attending sales in that space. Under such a condition the number of people in other spaces is usually normal or below. While this means that the air handling equipment for certain departments must be large enough to cope with the situation, the refrigeration equipment must be only large enough to handle the average maximum. If a system employing zone recirculating fans and a single central fan and dehumidifier were used, the saving would be reflected in the capacity of the central fan and dehumidifier. Another example of this diversity is found in an office building having restaurants and stores of certain types on the first floor and basement. At noon, when the restaurants and stores are crowded, the offices are below normal occupancy.

Heat lag should be carefully considered in the cooling load calculations.

In certain types of building construction the effect of solar radiation is still apparent hours after the sun has shifted from that exposure. In other types having a much lighter construction, the heat gain due to solar radiation decreases markedly with the passing of the sun. Some walls, having been warmed by the sun, may radiate heat long after the passing of the sun, thus requiring lower inside temperatures to offset the radiant energy.

Storage effect is usually present in some degree. Often it can be utilized to great advantage and it has more than once provided an unknown safety factor. If a space is kept below the design inside temperature for a period of time, the interior walls, floors, furniture and fixtures begin to assume the temperature of the space. Where the period of time is sufficient, the entire mass may reach the room temperature, rather than just the skin or surface of the item. Thus, when a space has been precooled below the

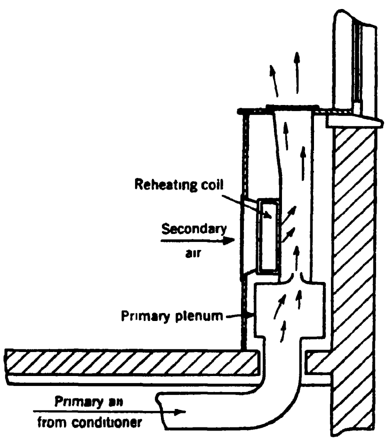

FIG. 10. INDUCTION UNIT (LOW PRESSURE TYPE)

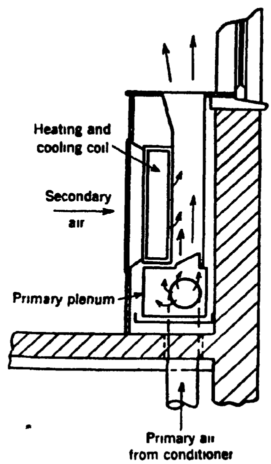

FIG. 11. INDUCTION UNIT (HIGH PRESSURE TYPE)

design maximum temperature for a period of time prior to the advent of the peak load, and the heat gain begins to increase to peak conditions, some of the increase is used in raising the temperature of the furniture, fixtures, etc., to the design conditions and the cooling load can be reduced accordingly. However, unless very accurate data with regard to the mass, surface, specific heat, etc., of the items within the space are available, due caution must be used in discounting the cooling load for this storage effect. In the absence of reliable data it is often a matter of experience rather than calculation.

Where air conditioning supply and return ducts pass through unconditioned spaces there will be a transfer of heat from these spaces to the air in the ducts, even though these ducts are well insulated. An allowance should be made for this heat gain and included in the heat estimate so that air can be supplied at a temperature low enough to offset the rise caused by this heat gain (see Chapter 31). There will also be some heat gain to the air in ducts passing through conditioned spaces, but since a cooling effect is produced in the space through which the duct passes,

this is not a loss and usually can be compensated for by adjustment of air quantities between the various spaces.

Heating Load

Methods of calculating the heating load are shown in Chapter 6. Many of the factors outlined previously under Cooling Load, such as zoning, non-simultaneous peaks, and diversity, apply in the reverse manner due to the heating requirement instead of the cooling requirement. However, these factors enter into the heating load picture from a standpoint of control of inside conditions, over-all performance and economy of operation more than from a capacity of equipment standpoint. It is not only necessary to heat a building or space to its design conditions when there is but the merest fraction of normal occupancy, practically no lights, internal heat, or solar radiation, but it is also necessary to provide capacity to heat the building quickly after a shut-down such as when a sudden cold snap follows relatively warm weather, or after a week-end or holiday. However, in normal operation during week-ends and holidays, buildings are usually kept at a *holding* temperature to prevent the freezing of services and conserve fuel. In many cases it requires less fuel to keep a building or space at a temperature of 50 to 65 F for some time than to shut the system down and then bring the temperature up again.

Apparatus Dew-Point

The method of locating the condition line for a given air conditioning problem has been explained in Chapter 1, Examples 19 and 20. Briefly, the method consists in estimating the net energy gain (or loss) per hour and the net moisture gain (or loss) per hour from data on location, exposure, construction, appliances, occupants, ventilation requirements, inside and outside design conditions. In computing the quantities of energy and moisture introduced and displaced by the ventilating air, only that portion of the ventilating air admitted directly to the conditioned space is considered. With this understanding, the ratio of the net energy gain (or loss) to the net moisture gain (or loss) determines the slope of the condition line through the state point of the inside air on the Mollier Chart.

The condition line may or may not cross the saturation curve. If it does, the intersection is called the *apparatus dew-point*, with application to summer cooling in mind. Thus, if the air conditioning apparatus can be set to take inside air, process it, and return it to the conditioned space completely saturated at the apparatus dew-point, the cooling load requirements can be exactly met both as to removal of energy and simultaneous removal of moisture.

In actual practice with commercial apparatus, it is rarely possible to obtain complete saturation and there may be several degrees difference between the dry-bulb and wet-bulb temperatures of the air returned to the conditioned space. This causes no difficulty. In fact, the only special significance of the apparatus dew-point is that, when it exists, it provides a convenient control point at which to regulate the operation of the apparatus, provided complete saturation is attainable.

In order to illustrate the effect of incomplete saturation in the conditioning apparatus, consider the cooling load problem of Chapter 1, *Example 19*. In this problem, 114,600 Btu of energy and 15.92 lb of moisture per hour are to be removed simultaneously. The slope of the condition line is determined by the ratio $q = 114{,}600 \div 15.92 = 7205$ Btu per pound of water and the apparatus dew-point is 58.02 F, as shown in Fig. 12. (Point *B*).

But suppose that the saturation efficiency of the apparatus is only 95 per cent, by which is meant that the air delivered by the apparatus is only 95 per cent saturated. Then the temperature at which the condition line crosses the 95 per cent saturation curve is the proper temperature at which to regulate the apparatus. This temperature is easily found to be 59.6 F dry-bulb temperature (Point *A* in Fig. 12). Of course, a somewhat larger quantity of air will have to be recirculated, namely 114,600 ÷ (31.41 − 25.51) = 19,400 lb of dry air per hour, where the enthalpy of the air at 95 per cent saturation on the condition line is 25.51 Btu per pound of dry air and h = 31.41 for the inside air. Only 18,056 lb of dry air had to be recirculated per hour with complete saturation.

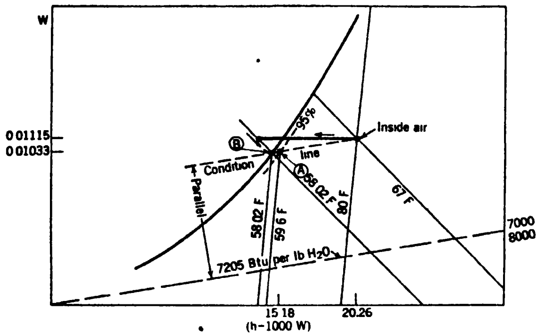

FIG. 12. DIAGRAM OF MOLLIER CHART ILLUSTRATING EXAMPLE 1

Many practicing engineers prefer an approximate method of calculating apparatus dew-point which employs what is called the *sensible heat factor, SHF*. The so-called *room sensible heat*, H_s, which is proportional to the quantity of supply air, Q is first estimated:

$$H_s = Q \times d \times C_p \times 60\,(t_r - t_e) \tag{1}$$

where

Q = quantity of supply air, cubic feet per minute.
d = density of air, pounds per cubic foot.
C_p = constant pressure specific heat of air.
t_r = room dry-bulb temperature, degrees Fahrenheit.
t_e = apparatus dew-point or supply air temperature, degrees Fahrenheit.

The so-called *room latent heat*, H_l, which is also proportional to the quantity of supply air, Q, is estimated:

$$H_l = Q \times d \times h_{fg} \times 60\,(W_r - W_e) \tag{2}$$

where

Q = quantity of supply air, cubic feet per minute.
d = density of air, pounds per cubic foot.
h_{fg} = an appropriate value of the latent heat of vaporization, Btu per pound.
W_r = room humidity ratio, pounds water per pound dry air.
W_e = apparatus dew-point or supply air humidity ratio, pounds water per pound dry air.

Finally the ratio

$$SHF = \frac{H_s}{H_s + H_l} \tag{3}$$

is computed. But, from Equations 1, 2 and 3

$$SHF = \frac{0.241\ (t_r - t_e)}{h_r - h_e} \tag{4}$$

where

0.241 = an appropriate value of C_p, Btu per pound.
t_r = room dry-bulb, degrees Fahrenheit.
t_e = apparatus dew-point, degrees Fahrenheit.
h_r = enthalpy at room conditions, Btu per pound dry air.
h_e = enthalpy at apparatus dew-point, Btu per pound dry air.

Thus, having estimated the value of the sensible heat factor, *SHF*, the problem of finding the apparatus dew-point reduces to solving Equation 4. Since the relation between h_e and t_e is only available in tabular or graphical form, a trial-by-error or graphical solution is required. Special charts have been devised for this purpose, but these are really unnecessary since the Mollier Chart of Chapter 1 is adaptable. Thus the quantity

$$q = \frac{h_{fg}}{1 - SHF} \tag{5}$$

where h_{fg} is an appropriate value of the latent heat of vaporization, determines, on the border scale of the Mollier Chart, the direction of the condition line which intersects the saturation curve at the apparatus dew-point.

Example 1. From a cooling load analysis of the room as determined in *Example 19*, Chapter 1, the net energy gain of 114,600 Btu per hour would be called *room total heat;* the net moisture gain of 15.92 lb per hour, multiplied by an appropriate value of the latent heat, say 1065 Btu per pound, would be called the *room latent heat*, namely, 16,950 Btu per hour; and 114,600 − 16,950 = 97,650 Btu per hour is the *room sensible heat.* Determine the apparatus dew-point, if the room conditions to be maintained are 80 F dry-bulb and 67 F wet-bulb temperature.

Solution. From Equation 3 the sensible heat factor is $SHF = 97{,}650 \div 114{,}600 = 0.852$. Using the latent heat of vaporization at 51 F, namely 1065 Btu per pound as an appropriate value in Equation 5, $q = 1065 \div (1 - 0.852) = 7205$ Btu per pound water.

The state point of the inside air is easily located on the Mollier Chart, as shown in Fig. 12. Through this point draw a line having a slope $q = 7205$ Btu per pound water as determined from the border scale. This line crosses the saturation curve at 58.02 F which is the apparatus dew-point temperature (Point B in Fig. 12).

From the point of view of satisfying the given cooling or heating load requirements, the effect of air passing through the apparatus without being *contacted* is merely to increase the quantity to be passed through. Thus, if 20 per cent of the air admitted is uncontacted, then 25 per cent more air must be admitted than would be necessary if all of it were contacted. The failure to achieve complete saturation in commercial apparatus may be explained as failure to contact all the air admitted. Or, some of the air may be deliberately by-passed. But whatever the explanation, the only thing of importance is the end result. Of course, a low saturation efficiency or a large proportion of uncontacted air means higher capacity for the conditioning apparatus and, perhaps, an excessive number of air changes for comfort.

In winter, a degree of saturation in excess of 30 per cent is seldom required and a low saturation efficiency may be desirable or even necessary where the full summer circulation air through the conditioner is maintained. With a spray type dehumidifier the main sprays may be shut off and an eliminator flooding pump provided which may give sufficient saturation. In other cases, such as where cooling coils are sprayed, the spray water may be throttled.

When it does become necessary to increase the saturation efficiency of the sprays, the spray water may be heated. The amount of heat put into the spray water by open or closed water heaters will be equal to that required to bring the dew-point temperature of the air entering the sprays up to that required before entering the reheater. It is possible where clean steam is available, to introduce the steam directly into the air stream to produce the desired dew-point temperature of supply air. However, the steam must be exceptionally clean or objectionable odors will result. This precaution should be observed also where open water heaters or ejector water heaters are used.

Present day practice, for spray type dehumidifiers of good design, assumes that the air leaves the dehumidifier at 1 to 2 F higher than the temperature of the spray water leaving the dehumidifier. A spray dehumidifier having sufficient length of spray chamber and density of spray together with a proper arrangement of nozzles may closely approach complete saturation.

Air Quantity and Effective Temperature Difference

The difference between the room air temperature and the supply air temperature at the outlet to the room is known as the *effective temperature difference*. In the theoretical case of a dehumidifier having a 100 per cent saturating efficiency and where this air is delivered directly to the room without temperature increases due to heat gain, then the *effective temperature difference* is the difference between room temperature and apparatus dew-point temperature. If duct heat gains are considered a part of the room load, this still holds true. The apparatus dew-point, as outlined previously, is fixed by the latent and sensible loads of the space, but in many cases, it is desirable to deliver more air to the spaces than is indicated by the difference between the room temperature and the apparatus dew-point.

It has been indicated that where a percentage of air is passed through the dehumidifier without being treated the relationship was modified in direct proportion, and that if room air passed through untreated no effect on the heat balance resulted Similarly, if room air is passed around the dehumidifier and mixed with the treated air the heat balance is not adversely affected. Therefore, if the quantity of air passed through the dehumidifier is determined by the usual methods, room air can be passed around the dehumidifier and mixed with the dehumidified air, increasing the supply air quantity and temperature and decreasing the *effective temperature difference*. Thus if the difference between the room temperature and the apparatus dew-point indicates that 10,000 cfm at 30F below room temperature will be required to hold conditions, that quantity can be passed through the dehumidifier and cooled to 30 F below the room, then mixed with 10,000 cfm of room air resulting in a supply air quantity of 20,000 cfm and an *effective temperature difference* of 15 F instead of 30 F. Supply air outlets and grilles that have a high induction ratio (that is, a large amount of room air is mixed with the air leaving the outlet within a short distance of the outlet through the induction effect of the air stream) are available as well as induction units. A proper selection of outlets or units may make it possible to introduce air at low temperatures and high velocities without causing objectionable drafts or cold spots, but care must be used to see that too little air motion is not a result. Lower *effective temperature difference* may be required for this reason. While the use of a high *effective temperature difference* results in a saving in initial cost of fans and ducts and in the operating cost of fans,

this difference should be carefully considered. If the sensible heat load of a space is subjected to substantial variations, lower *effective temperature differences* should be considered, since systems employing a *low effective temperature difference* will be less exacting in control requirements.

Assume that a space has a sensible heat load so that 10,000 cfm of air supplied at a 30 F *effective temperature difference* would be required to maintain a room temperature of 80 F. If the load is suddenly reduced 50 per cent with the air supplied at the same temperature, the resultant room temperature would become 65 F. On the other hand, if 20,000 cfm were supplied at an *effective temperature difference* of 15 F and the load suddenly reduced 50 per cent the resultant room temperature would be 72.5 F. This, while a rather extreme example, indicates the less exacting demand on the controls brought about by the use of the lower *effective temperature difference.* Of even greater importance is the case where two or more spaces are controlled from an average condition such as by a thermostat located in the return air stream. From the previous example, it can be seen that a large variation in the load in one of the spaces will not reflect itself in such a large change in room temperature. Thus in the long run the larger *effective temperature difference* may not be the most economical.

The analysis in the foregoing applies largely to summer air conditioning. The same analysis will apply to some extent in winter. The mathematical relationship is revised due to a heating requirement rather than cooling. Present practice indicates a high temperature difference in winter in comparison to that for summer. It is due to the fact that the heat losses in winter in Btu per hour far exceed the summer heat gains in Btu per hour, and particularly to the fact that in winter, reheating the air to produce desired room conditions is not reflected as a load on the system as it is in the summer, but merely accomplishes the necessary work.

In line with the latter, reduction of air quantity by slowing down the fans for the winter season and increasing the temperature difference often is feasible, creating a saving in fan horsepower at no expense to the final heat balance, providing the air distribution is not seriously affected.

Extremes should be avoided in all cases. For summer air conditioning low supply air temperatures will result in larger heat gains to the air passing through the ducts, poor control, etc. Too high a supply air temperature may result in excessive initial and operating costs. Suggested limits for the *effective temperature difference* are from 12 to 20 F, the actual selection being based on the requirements of the particular case. For winter air conditioning too high supply air temperatures result in excessive heat losses from the ducts and stratification within the room unless thorough mixture is insured, while too low supply air temperatures may cause drafts, high operating costs, etc. Suggested limits are from 15 to 35 F. There can be no set rule and each case should be judged according to its particular requirements.

By-Pass

The by-pass, in its accepted form, consists of an arrangement of ducts and apparatus connections with the necessary dampers which will permit air to pass around the dehumidifier or conditioner without being treated. It has two functions which may be employed separately or simultaneously.

The first of these is to provide a means of temperature control at a substantially constant total air quantity. If in summer, the load within the conditioned space is reduced and the temperature begins to fall, this

drop in temperature can be offset by passing some of the air around the conditioner instead of through it, while the total quantity of air in circulation remains unchanged. When used for this purpose, it is termed an *adjustable* or *automatic* by-pass. The second function is to maintain a lower effective temperature difference between the air supplied to the room and the room temperature than could be obtained if air at the apparatus dew-point were supplied, and when so used is called a *fixed by-pass.* As discussed previously, if return air from the conditioned space is passed around the conditioner and mixed with the conditioned air, the effect on the heat balance is the same as if the air were removed from the space and immediately reintroduced. This is not strictly true, due to the fact that when ducts pass through unconditioned spaces, there is a heat gain by this air, and an additional gain is imposed by the heat of compression of the circulating fan in moving the air against the resistance of the system. However, the heat gain, where the by-pass is used to lower the effective temperature difference, usually favors its use due to the fact that the increased volume and the resultant higher temperature of the mixture of conditioned air and room air may show a lower net duct heat gain with a smaller temperature increase per unit of volume of supply air. The advantages thus obtained may offset the increased fan power.

The adjustable or automatic by-pass can be made to serve the purpose of the fixed by-pass by providing a stop on the by-pass damper so that it cannot close completely. In some cases this stop is unnecessary since commercial dampers will permit an air quantity of 10 to 20 per cent of the conditioner capacity, depending on the resistance of the conditioner, to leak through even when fully closed.

Where a reduction in room sensible heat is not accompanied by a reduction in room latent heat, the by-pass is to be used with care. If the dew-point temperature of the air leaving the dehumidifier is controlled at a fixed value the reduced quantity passed through the conditioner may be sufficient to handle the sensible heat load but insufficient to handle the latent heat load, resulting in humidities that are too high. If the dew-point is not controlled, as when cold water is supplied at a constant temperature or where direct expansion cooling coils are used, the reduced loading on the conditioner brought about by the reduced air quantity will result in a lower dew-point temperature which often is entirely adequate. This condition should be investigated in each case.

The by-passing of outdoor air is to be avoided in general. While the sensible heat requirements of the space may be such that the by-passing of high temperature outdoor air will aid in controlling room temperature, high moisture content air introduced to the space may raise the humidity to an objectionable amount. If return ducts from which the by-pass air is taken run through unconditioned spaces and there is an inward leakage of outdoor or moist air, the effect, in a lesser degree, is that of by-passing outdoor air. Therefore the location of such return ducts and the points from which such air is taken are of importance. Exceptions to this are where the moisture content of the outdoor air is lower than that of the room, and where the sensible heat to latent ratio increases at partial loads.

The foregoing applies largely to summer air conditioning. In winter the by-pass usually is kept closed and room temperature control obtained by means of regulating the amount of heat supplied to the air. In some instances when the by-pass is located after the reheaters, the operation of the by-pass damper may be reversed and the by-pass still used as a means of control with the reheaters full on or operated in sequence with

the by-pass. In other cases the reheater may be located in the by-pass as described further.

The principle of the by-pass may be applied to items of equipment other than conditioners, such as humidifiers, dehydrators, heaters, etc., in much the same manner. Where a heater has a capacity such that the temperature rise of the air is higher than desired, a smaller heater may be used and a portion of the air by-passed around the heater. Throttling of the sprays in a humidifier is, in effect, a by-pass since it increases the portion of air passing through without being contacted.

Reheating

Reheating the supply air is necessary in winter where this air is used to offset heat losses. Tempering of the supply air (merely reheating to a lesser degree) is required where other means of heating, such as direct radiation or panel heating, are used to carry the main heating load. Supply air at the required apparatus dew-point would add to the load to be carried by the direct radiation and in addition may create a movement of cold air that while desirable in summer may be undesirable in winter. Modulation of the amount of steam supplied to the coils can be used to control temperature, or air can be by-passed around these heating or tempering coils.

Since reheating or tempering coils are required for winter and year 'round air conditioning, their use as a means of summer as well as winter temperature control is indicated. Where heat is available in summer, as the room sensible heat falls off, the low temperature of the supply air can be raised by means of these coils to maintain the desired room temperature while still providing adequate air at the proper dew-point.

Reheating presents an excellent method of accurate temperature control since the quantity of air passed through the conditioner is not changed as in the case of the by-pass, and since the distribution and circulation are not affected as when the volume of supply is reduced. However, reheating in summer has one disadvantage. It has the effect of maintaining a constant internal sensible heat load on the system. This means that when an effective temperature difference of 15 F is being maintained at the maximum room sensible heat load, this temperature difference must be reduced by means of heating to 7.5 F at half of the sensible heat load. The room sensible heat usually is about 35 to 45 per cent of the total cooling load and thus the penalty imposed on the refrigeration cycle is not extremely large. When the outdoor wet-bulb temperature is less than the desired room wet-bulb temperature, all outdoor air should be passed through the conditioner and under these circumstances reheating does not impose a load on the refrigeration cycle since the heated air is not returned to the conditioner. Further, where the volume of outdoor air introduced to the system, which for all practical purposes is wasted after its work has been done, is sufficiently large in relation to the amount of reheat used (that is, if the heat required does not exceed that necessary to raise the outdoor air only from dew-point to room temperature) the use of reheat does not impose a load on the refrigeration cycle. This, of course, assumes that there is no economic penalty involved in the use of heat itself.

One of the best applications of reheating for summer purposes is in combination with other means of temperature control for the purpose of levelling off accentuated demands for temperature control. As an example, the by-pass can be applied to a certain extent, or the volume of supply air throttled to a limited degree, or both of these used in sequence, and reheating can be used as the final step in temperature control.

In the foregoing it has been assumed that summer reheating is derived from an extraneous source of heat such as steam, electric heaters or hot water. The economics of reheating as outlined may be improved by the use of sources of heat that are available within the equipment used.

Where certain types of refrigeration cycles are used, auxiliary refrigerant condensing coils can be placed in the air stream. At partial loads these can be used as additional refrigerant condensing surface and improve the performance of the refrigeration plant. This is generally known as hot gas reheating. Less effective is the use of coils in the air stream through which liquid refrigerant from the condenser is passed before being delivered to the evaporator, sub-cooling the liquid refrigerant and improving the performance of the refrigeration cycle. The use of refrigeration condenser water as a source of reheating provides definite economics. If the condenser water is passed through coils in the air stream before being used for refrigerant condensing purposes, the lowering of the condenser water temperature accomplished by reheating the air will result in savings in the refrigeration plant power consumption and increased refrigeration capacity. The latter two methods are at some disadvantage in that the amount of reheat available decreases as the need for reheating increases, particularly where evaporative condensers or cooling towers are used.

Zoning

Zoning consists of an arrangement of equipment or a division of equipment into sections that will permit individual control of the temperature and humidity of those spaces or groups of spaces that do not have simultaneous variations in sensible or latent heat load. The equipment is so arranged or divided that air can be supplied to spaces or groups of spaces in accordance with the individual load requirements of that space or group of spaces.

Solar heat gain is one of the major causes of the zoning requirement since its effect and amount vary with the season, time of day and exposure. Other sources of heat gain, subject to variations, such as large changes in the number of occupants in one space with a constant occupancy in another indicate the necessity of zoning. Some of the various methods of zoning are:

1. Separate equipment.
2. Reheating or recooling. (See Figs. 4 and 5.)
3. Multiple fans with individual by-pass. (See Figs. 6 and 7.)
4. Volume control. (See Fig. 8.)
5. Dual duct system. (See Fig. 9.)
6. Combinations of the above methods.

Zoning by separate equipment represents the extreme in zoning. Individual conditioners, fans, heaters, controls, distributing duct work, etc., are provided for each zone and are separate from those of other zones. Each assembly of equipment is arranged to operate at full or partial load according to the requirements of that zone. In extreme cases an individual refrigeration plant may be provided for each zone. In general, this method of zoning is uneconomical since each piece of equipment is large enough to handle the full load requirements of that zone and no advantage can be taken of the fact that while one zone is at its full load others may be operating at considerably less than full capacity. This applies both to the initial cost of a system and the operating cost. There are, however, many cases where this method of zoning when used to a limited degree or combined with other methods is most desirable.

Zoning by reheat is one of the more simple methods of approaching the problem of differing variations in load. Reheating has been discussed in the foregoing. Heating coils located in the distributing duct work or apparatus connections which supply only those spaces having substantially the same load variations will, by adding heat when the cooling load falls off (or the reverse in the case of a heating load), maintain the desired temperature conditions. It has been shown previously that where the amount of reheating required is not excessive and that heat for this purpose is available from an economic standpoint this method of zoning may be entirely practical and even highly desirable.

Zoning by recooling is literally the reverse of zoning by reheating. It is not used often but there are many cases where it is desirable. Its most practical application is limited to a sensible heat removal function where the latent heat requirements are handled by another source such as a dehumidifier or dehydrator, and where the recooling equipment (usually cold water cooling coils) can be utilized for reheating (usually with hot water) in the heating season. In using this method of zoning, when the cooling load of a given space is reduced, the cooling effect produced by the recooler is also reduced.

The use of *multiple fans with individual by-passes* presents a simple and sometimes inexpensive method of zoning. Two or more fans may be arranged so that each draws its treated air from the same conditioner. The connection between the conditioner and fans is divided or partitioned in such a manner that a by-pass connection can be made to each fan. See Fig. 6. In this way the amount of by-pass air can be regulated according to the requirements of the zone served by that fan. In this application a face damper must be used for each segment of the conditioner, arranged to close as the individual by-passes open or an unbalanced system may result. A much better adaptation of this principle, though slightly higher in initial cost, is obtained by using a single conditioner or dehumidifier, having a central conditioned air fan delivering the treated air to the necessary zone fans where the conditioned air is mixed with return air according to the requirements of the zone. See Fig. 7. With this method of zoning, the zone fans are provided with casings to which both return air and conditioned air are delivered, their proportions being regulated by dampers working in opposite directions. If conditioned air is delivered to the casing at slight positive pressure the return air damper may be omitted. Reheating can be effectively combined with either of these for year 'round use or for winter use only. The reheater is usually located in the air stream to the zone fan. Where a central conditioned air fan is used to deliver conditioned air to a number of zones, a static pressure regulator controlling a volume damper or inlet vanes on this fan should be installed to prevent unbalancing the system, when one or more zone fan conditioned air dampers are throttling. This is one of the better methods of zoning and is particularly effective when used in combination with reheating for winter or year 'round conditioning as previously outlined.

Volume control is the least expensive and most frequently used method of zoning. It is usually obtained by placing a throttling damper in the supply duct feeding a particular zone and operating the damper to restrict flow of air as the heating or cooling load is reduced. The damper may be operated manually or automatically. Volume control, however, has two serious disadvantages. The first of these is that any large reduction in the quantity of air supplied may impair the ventilation. The second is that a large reduction of the air supply may entirely upset the distribution from the room outlets causing dead pockets, stratification,

lack of air motion, or the reverse—undesirable drafts. Where the degree of volume control is large or the system extensive and where volume control is applied to one portion of a system and not to another, the use of a static pressure regulator controlling a fan discharge damper or fan inlet vanes is indicated. Its best application is in combination with some other method of zoning such as reheat or by-pass where it is used as one step in a control sequence, and the reduction in volume limited to a proper amount.

The *dual duct method* of zoning, sometimes referred to as the schoolhouse system, can be successfully applied to comfort air conditioning though certain precautions must be observed. Essentially this method employs a source of warm air and a source of cold air both of which are delivered to a common point where either or a mixture of both are delivered to a particular zone according to the requirements of that zone. See Fig. 9. Several variations of this method are possible, some of which do not employ dual ducts as denoted by the name but which utilize the principle. An example of this is found in a blow-through system where the fan is located on the entering side of the conditioner and the conditioned air passes from the conditioner into a plenum from which distributing ducts for the various zones are taken. A by-pass connection around the conditioner from the fan to the zone duct is made and dampers provided so that conditioned or untreated air, or a mixture of both is passed into the zone supply duct. In this method of zoning and in most of the variations of this method, the matter of by-passing untreated or outdoor air presents itself. This has been discussed earlier in this chapter. If a return air fan is used and the by-pass connection made from the return fan to the zone supply duct, then the by-passing of outdoor air does not need to be considered. Complication of ducts and connections should be avoided since it may result in difficult sheet metal work with accompanying leakage of air and waste of cooling or heating effect.

Combinations of these several methods of zoning usually provide the most effective zoning. It is then possible to use each method to its greatest advantage without incurring operative or economic penalties which may be inflicted by the exclusive use of any one. Thus, where wide variations in load occur, volume control can be used to reduce the air quantity a limited amount, then as the load continues to fall off, reheat, or by-pass, or both can be used. By-pass and reheat can be used in series very effectively. Many combinations are possible and each case should be considered with regard to its particular requirements when deciding on the method of zoning.

Induction Units—Low pressure type

Induction units are essentially induction type convectors. These units utilize a jet of conditioned air (or primary air) to induce into the unit a flow of room or secondary air which mixes with the primary air. The mixture is discharged into the room through a grille at the top of the unit. Heating coils are located in the secondary air stream for use in heating. Control is obtained by either manually or automatically throttling the jet, and, in addition, heat may be supplied to the secondary coils in summer as well as winter to provide control by reheating. The use of these induction units presents several advantages. Since the secondary air stream is thoroughly mixed with the high velocity low temperature air stream before leaving the discharge of the unit, the resultant temperature of the mixture is satisfactory even though the primary air is introduced at a temperature too low for ordinary methods of distribution. A unit is

usually provided under each window in place of the customary direct radiation, and combines the air distribution system with the heating system. With a conventional system it may be necessary to provide supplementary heating in the form of direct radiation. These induction units may be selected so that their heating coils will have sufficient capacity under gravity conditions (that is, with the fan system off and no primary air entering the unit) to maintain the building or spaces at a reasonable temperature. The use of low temperature dehumidified air which has not been reheated or mixed with room air before delivery to the room results in a reduction in fan capacity and smaller sized duct work. In some cases the use of the by-pass may be desirable in order to keep the primary air volume up and provide additional control. This system can provide a degree of zoning that is usually impossible with conventional systems since each unit can be put under manual or automatic volume control and reheat control. The selection of units should be made with regard to noise level when related to the noise level of the spaces. The inductive capacity of the unit increases with the jet velocity but too high jet velocities result in a high noise level.

Induction Units—High pressure type

A recent development of the induction unit as previously outlined is the high pressure type of unit. This unit employs nozzles which produce a high velocity jet quietly. The term, high pressure type of unit, is to some extent inaccurate since the pressure at the nozzles, while several times that of the low pressure unit, is still less than the total resistance pressure of a conventional central system. The high velocity jet of primary air induces a flow of secondary room air through coils located in the secondary air stream. The coil in the secondary air stream is supplied with chilled water in summer and hot water in winter and thus handles a large portion of the room sensible heat gain in summer and of the room sensible heat loss in winter. The primary air is supplied at a sufficiently low dew-point to take care of room latent heat gain in summer. In winter, it is supplied at a sufficiently high dew-point to take care of room latent heat losses. Control of temperature is obtained by throttling the water quantity supplied to the secondary coils. The quantity of primary air is greatly reduced due to the fact that a portion of the room sensible heat load is carried by the secondary air stream coil. Since the primary quantity is small, very high velocities can be carried in the supply ducts without requiring fan power in excess of that required for a conventional system. This means that the supply ducts or pipes can be very small and can be run in chases, or furred in at columns with the water pipes. The primary air is treated in the usual manner to provide air at the required dew-point and either a surface or spray dehumidifier or a dehydrator may be used. The primary air quantity is sufficient for ventilation purposes, and frequently consists entirely of outdoor air. The water piping for the units can be so arranged and valved that hot water can be supplied to one zone that may require heating while cold water may be supplied to a zone that requires cooling.

This system is usually limited in application to hotels, apartments, office buildings and other multi-room installations having a large perimeter with relation to the floor area. The units are usually installed beneath the windows, replacing direct radiation or convectors. Where the spaces to be conditioned extend a large distance from the outer wall into the interior of the building, a separate system or zone for the conditioning of the interior portions may be required.

Evaporative Cooling

In climates where on design maximum days, the outdoor wet-bulb depression is relatively high it may be possible to dispense with refrigeration or other cooling sources by use of the evaporative cooling effect. A well designed air washer using recirculating sprays will reduce the entering dry-bulb temperature to within a degree or two of the entering wet-bulb condition. Thus, it may be possible that with air entering at 100 F dry-bulb and 60 F wet-bulb temperature a leaving condition of 62 F dry-bulb, nearly saturated can be obtained. Under some conditions of latent and sensible heat load the results may be entirely satisfactory.

Under those conditions when the outdoor wet-bulb temperature is not quite low enough to permit the use of straight evaporative cooling it is possible to use precooling coils with refrigeration, well water or a cooling tower as the basic source of cooling to lower the wet-bulb temperature (by sensible heat removal) of the air before it enters the air washer. Where internal heat loads are high, this may be more economical than using return air. Under other conditions where the required supply air dew-point is too low to permit straight evaporative cooling and the sensible heat load not too great, intentional partial saturation may be employed. That is, the low dew-point of the outdoor air is utilized by permitting some of it to pass through the humidifying sprays untreated or pass around the humidifier. All of these remarks with regard to evaporative cooling are based, as indicated, on the assumption that the supply air will consist entirely of outside air. Provision should be made for the return of air from the conditioned spaces for control purposes as well as for winter use in all cases.

Precooling

Where sufficiently cold water from wells or streams is available a saving in the refrigeration cycle may be obtained by the use of precooling. Cooling coils are placed ahead of the dehumidifier or conditioner and the cold water from a well or stream circulated through the coils. The resultant cooling of the air decreases the load to be carried by the dehumidifier and refrigeration plant. In normal practice the water after passing through the precooling coils is delivered to the refrigeration plant for condensing purposes. The economic advantages of this scheme are apparent and it is frequently used.

Sensible Cooling with Dry Cooling Coils

Under certain atmospheric conditions where a large wet-bulb depression exists and the dew-point of the outdoor air is sufficiently low at all times, proper inside conditions may be obtained by removing sensible heat only from the outdoor air and supplying it to the spaces. Under this condition of a high wet-bulb depression a cooling coil may be located in the air stream and this coil supplied with water from a cooling tower of one type or another. When humidity control is desired sprays to saturate or partially saturate the air may be used after the dry cooling coil. Saturation or partial saturation after the dry air cooler will further reduce the dry-bulb temperature of the supply air and reduce the supply air quantity required. This system has very definite application in hot dry climates and in general is most economical.

Run-Around System

An interesting method of control is found in the use of combined reheating and precooling usually termed the *run-around system*. Coils are placed in the air stream before and after the conditioner and water or

brine is circulated around the conditioner from one coil to the other. The water passing through the reheating coil is cooled by the air leaving the dehumidifier and the air is heated by the water. The cooled water is then circulated through the precooling coil where the entering air is cooled by the water and the heated water sent back to the reheating coil. The run-around has the advantage of permitting a higher supply air dew-point temperature than would be possible otherwise. This is due to the fact that continual reheating is available which is not a large penalty on the refrigeration plant since it provides precooling at the same time. This reheating at peak load creates an artificial sensible heat gain which increases the ratio of room sensible heat to room total heat and for a given room temperature results in a higher apparatus dew-point. Thus, while the volume of supply air is increased, the low side temperature level of the refrigeration plant is raised and this may effect savings in initial and operating costs. This system has the disadvantage of providing a decreasing amount of heat for reheating as the demand for reheating increases.

RELATION TO BUILDING TYPE

Few buildings or spaces are physically identical and those that are similar in this respect may have marked differences in internal loading, zoning requirements, and economic limitations. Consequently it is virtually impossible to establish fixed rules governing the type of system to be used. Each case must be considered on its own merits with due regard to all engineering and economic factors. However, some generalizations are possible.

In small single spaces the use of an elaborate system is undesirable from an initial cost standpoint. Zoning may be often eliminated. This also applies to large single spaces except that in very large spaces the necessity of providing adequate zoning is encountered more frequently. If the spaces are extremely large, physical and economic limitations such as the size of equipment, size and length of ducts, may require the division of the space into sections. Whether these sections are to be made according to zones or whether each section is to be zoned will depend on the particular case.

Where groups of spaces or small buildings are encountered, simple systems still prove the most economical. A single central station with zoning by means of volume control and reheat combined may be entirely satisfactory. If the perimeter of the building is large with regard to the area, induction units of the low or high pressure type may be employed, particularly if it is a multi-room application. Occasionally the dual duct system may be considered, but is infrequently used due to its complications.

Low buildings with large floor areas, such as large department stores and large general offices, may have to be divided into sections for treatment. In the case of large department stores it may be possible to provide a single conditioner with a fan delivering the conditioned air to recirculating fans which supply the various departments or spaces. This application is limited by practicability of running the large conditioned air ducts to the various recirculating fans. In some cases the use of separate systems for each section will be indicated with the added necessity of dividing into horizontal as well as vertical sections. If the latter is required, each vertical section may be handled by a separate system consisting of a single conditioner and fan delivering conditioned air to recirculating fans supplying the horizontal sections. Zoning is obtained by proper allocation of the recirculating fans or other conventional methods

used in conjunction with the recirculating fans. For general offices in particular, or types of buildings having a large perimeter, the use of one of the induction type units for perimeter treatment, combined with a conventional system for the treatment of the interior portions, offers possibilities.

High buildings having large floor areas may be successfully handled in many ways. Horizontal or vertical sectionalizing or both may be required, as determined by economic factors and physical limitations. Where vertical sections only are required, the use of a single conditioner and fan for each section delivering conditioned air to zone fans at various floors may be used. These zone fans can be so arranged that one recirculating fan can handle similar zones on several floors, thus reducing the number of fans required and providing a degree of vertical zoning. Where vertical sectionalizing is not indicated, the building may be divided into horizontal groups, each group handled by a central system and adequately zoned. In some extremely large buildings apparatus rooms for the systems may be located in the basement and in the attic and on intermediate floors.

In high buildings having small floor areas the treatment required may be the same as for that of a vertical section of one having a large floor area. A single conditioner and fan can be used to deliver conditioned air to recirculating fans located at various floors.

In all cases of high buildings the necessity for horizontal sectionalizing is indicated by the economical size of air supply and return risers and by the extent to which they encroach upon usable space. In all buildings the necessity for vertical sectionalizing is indicated by the economical size of horizontal supply and return ducts and the space requirements of these ducts.

Usually the most simple systems are best adapted to theaters, auditoriums, and similar applications. Zoning is seldom required other than in connection with auxiliary spaces served by the same system. A conventional central system with by-pass control possibly augmented by reheat usually will suffice. The auxiliary spaces may be supplied with air from the same system controlled by volume reduction and reheat. Balconies and large lobbies frequently justify the use of separate zoning fans.

The foregoing are merely generalizations and suggestions. It is the responsibility of the engineer to explore thoroughly the possibilities of all types of systems and employ that best suited to the purpose from a standpoint of maintained economy, maintenance, life, operation and physical applicability.

EQUIPMENT SELECTION

Other chapters cover in detail most of the items of equipment used in a central system. Each item must be selected not only on its own merits, but in relation to all the other items that go to make up the complete central system. Each item should be considered from the standpoint of both initial cost and operating costs. Consideration must be given to performance at partial loads since most systems operate at full load but a small percentage of time. Many items of equipment have been well standardized and are manufactured in certain definite sizes. The fullest advantage of this should be taken. One item that may be oversized of necessity may permit the use of a smaller piece elsewhere.

Fans operate at full capacity continually in many systems and therefore should be selected for good efficiencies. In winter where higher tem-

perature differentials are used it is possible to use lower air quantities and a two-speed motor may be provided for the fan, resulting in a power saving. In such cases the air distribution under the reduced volume should be investigated before providing this feature.

In selection of the dehumidifier or conditioner the relation of this item to the refrigeration plant is to be given careful consideration. Frequently, it is possible to make a saving in the refrigeration plant by providing more surface in the dehumidifier or conditioner. On the other hand, an excess of capacity in the refrigeration plant can be used to lower the apparatus dew-point (if the lower room humidity is satisfactory) resulting in a reduced quantity of dehumidified air and smaller dehumidifier.

In winter, where preheating coils in outside air intakes are required and subjected to entering air temperatures below freezing, the coils should be selected for operation at full capacity whenever the entering air temperature is below 35 F or where throttling of the steam supply to the coils is desirable. A type of coil that is designed especially for this must be used. In many cases the use of preheaters is not justified since the temperature of the mixture of outside and return air may be entirely satisfactory.

Where reheating coils are used after the supply fan, either as zone control reheaters or boosters, and are relatively near outlets, care is to be used to install these coils so that any stratification of temperature produced by the throttling of the steam or water supply does not result in cold air being delivered to one outlet and warm air to another. Some types of coils that do not produce stratification under throttled conditions are commercially available.

The selection of the refrigerating plant is in itself an economic study. The availability, consumption, and costs of condenser water are to be compared to the initial costs involved and water savings produced by the use of cooling towers or evaporative condensers. Whether or not a direct expansion or flooded system, cold water or brine type of plant will be used is not only a matter of initial cost but one of over-all performance, operating economy, compactness, and in some cases, one of safety. Where water or brine is used as a cooling medium, the possibilities of using lower temperatures and decreased quantities of water or brine, with resultant savings in pumping power and line sizes, are to be compared with the increased power consumption and probable increased cost of the refrigeration machine.

Air cleaning devices are to be selected according to the particular requirements of the project as well as to existing atmospheric conditions. In some applications such as certain types of stores or departments in large stores, lint screens should be provided for the return air as well as filters. Whether or not return air is to be filtered will depend upon the individual case and will be related to the amount of dirt or dust generated in or brought into the conditioned space from various sources. The type of filter or cleaning device to be used depends largely on economic considerations. Obviously an expensive high efficiency cleaning device is not warranted where atmospheric dust or dirt is of such a nature that a less expensive, less efficient device will remove the more objectionable matter. Interior cleaning costs, dust and dirt damage, and hazard due to an accumulation of inflammable dust or dirt within the system are most important factors.

Automatic instruments are nearly always used in present practice for the control of temperature and humidity and to an increasing extent in the control of large refrigerating plants as well as small ones. Whether

electric or pneumatic controls are to be used is a function of the particular requirements of the application and with certain exceptions a function of economy. Either electrically or pneumatically operated controls can be made to serve the same purpose though the individual case may favor one or the other. A detailed discussion of automatic controls may be found in Chapter 33. One point is to be emphasized. In general, the simpler the control system the better it will perform. Control systems seldom receive the maintenance they deserve and the fewer the instruments and devices the better the care. Further, it is poor policy to provide, at additional expense, instruments of extreme sensitivity to devices incapable of responding to the control demands.

Insulation is an important factor in air conditioning systems. Its economics with regard to steam and water or brine piping are well known and need no comment. The insulation of duct work is not merely a matter of economics but sometimes is a necessity from the standpoint of limiting the temperature rise of the air even when the ducts are in conditioned spaces. This temperature rise of the air should always be taken into account when apportioning the air and sizing the ducts and will indicate the necessity for insulation. In some cases, leakage of air from the duct where the duct is in a furred space or chase will eliminate the need for insulation by maintaining a reasonable temperature surrounding the duct.

There are practically no items of equipment associated with central air conditioning systems that are not subject to economic limitations as well as those of performance and duty and all of them should be considered in the selection.

ARRANGEMENT OF EQUIPMENT

A proper arrangement of equipment is essential for the proper functioning of any system. Where systems are to be installed in existing buildings the arrangement of equipment may be limited by structural or space considerations, but no compromises that may prevent satisfactory operation or maintenance should be considered.

The location of the apparatus room is often determined by building construction or available space. The closer the apparatus room to the conditioned space, the less expensive is the duct-work. On the other hand if the equipment generates noises it may be necessary to locate the room some distance from the spaces or provide adequate sound and vibration treatment. The scattering of wet apparatus throughout a building is to be avoided unless suitable precautions are taken. It must be remembered that encroachment on spaces that are otherwise usable can be charged against the system as an operating cost.

In general the apparatus should be arranged to have straight line air flow. This is desirable but not always possible. Each change in direction is the source of air resistance, and in addition may cause eddy currents resulting in stratification. The usual order of equipment, beginning at the outside air intake is: outside air screen, outside air louvers, maximum and minimum outside air dampers, preheaters, return air connection, filters, conditioner, by-pass connection with or without reheaters, reheaters, fan and distributing ductwork. See Figs. 1, 2 and 3 for typical arrangements. Use of one or more of the methods of zoning may require a modification of this order but usually only after the dehumidifier or conditioner.

Outside air screens prevent the entry of large foreign matter, birds, etc. The use of louvers or a hood at the outside air intake prevents the

entry of rain and snow. Both of these should be used on all systems. As pointed out earlier, the louvers and screens should be of sufficient size to permit the passage of the entire conditioned air quantity.

The minimum outside air damper usually covers the entire face of the preheater, which in turn is selected for the minimum outside air quantity. The maximum outside air damper is designed for the difference between the dehumidified air quantity and the minimum outside air. Its size can be such that its air resistance when open will equal the air resistance of the open minimum outside air damper plus that of the preheater if used. Where the spaces conditioned are very tight against air leakage, some type of relief or positive exhaust may be necessary when all outdoor air is introduced to the system to provide some means of egress for the air. Such reliefs require back-draft dampers to prevent infiltration when all outdoor air is not being used. A return fan properly dampered as indicated later is sometimes used.

The return air from the conditioned spaces usually is brought into the apparatus between the preheater and the filter. Just as it is necessary to make provision for using all outdoor air it is necessary to make provision for using all return air. Heating an unoccupied building or cooling it after a shut-down is easier if all the air used is return air. Consequently the return connections should be ample for this. In many cases higher velocities can be carried in the return ducts during such times, since the fan will exert a pull at the return connection nearly equal to the resistance of the outside air connection and the return ducts need not be larger than for normal return air quantities at normal velocities. In other cases it is possible that a reduced quantity of air due to the increased resistance of the return duct system may be satisfactory during the starting period. Where the return air system is extensive or complicated a return air fan is desirable. This fan can serve as a combination exhaust and return fan by arranging dampers in its discharge so that the necessary return air can be delivered back into the system and the remainder discharged outdoors. When all outdoor air is passed through the conditioner the entire return air quantity is discharged outdoors.

The by-pass connection normally connects the return air duct system into the apparatus casing between the conditioner and fan. Usually it is sized to handle about 50 per cent of the fan capacity where a variable by-pass is used though extreme load variations may require a greater amount. It is at times good design to locate the reheater in the by-pass connection using a certain amount of by-pass air when heating is required. Since the relatively high resistance of the conditioner is to be balanced by the heating coil and by-pass connection, enough heating surface can be provided to raise the temperature of the by-pass air to the point where the mixture of by-pass air and conditioned air will have the required temperature. When a variable by-pass is used a damper working in opposition to the by-pass damper should be placed across the face of the dehumidifier, for unless the resistances of the two are most carefully balanced at all operating points the proper mixtures of air will not be obtained. The avoidance of by-passing outside air is again stressed. Where the by-pass is made a part of the dehumidifier or conditioner and located on the top or side of it, the return air connection should be made in such a way that stratification of return air is insured, baffles being provided to accomplish this purpose if necessary. Where return air and by-pass air connections are taken off a return duct system it may be necessary to install a back-draft damper between the return air connection and the by-pass connection, if the return duct system is extensive

and the connections simple. In this instance, when the by-pass damper is at maximum opening it may be much easier for outside air to pass through the return damper, into the return duct connection and through the by-pass than for return air to pass through the by-pass connection into the fan. Air always takes the easiest path and if the dehumidifier resistance is high, and the return duct resistances low, this situation is apt to occur unless precautions are taken. A return fan instead of a back-draft damper may be required for this case if the failure of return air to reach the dehumidifier or conditioner is a serious matter under reduced load conditions.

The location and arrangement of the dehumidifier, humidifier, or conditioner with reference to each apparatus assembly are more or less standardized. In general, the outside air intake, preheaters, and return air connections precede the conditioner while the by-pass, reheaters and fan follow the dehumidifier. In the cases of the *blow through* system, where the fan is located ahead of the conditioner, the leakage of air at the conditioner is outward instead of inward and may be accompanied by water leakage unless the proper precautions are taken.

The location of the complete apparatus assembly including the dehumidifier will be dependent on the type of building, spaces available, structural characteristics, etc. The type of conditioner used may limit the location under certain conditions. Where cooling coils employing chilled water or brine as the cooling agent are used there are few restrictions with regard to location other than those of pumping power, working pressures, line costs, etc. Where open spray dehumidifiers are used very definite limitations present themselves, and these may require certain extraneous equipment to make the system workable. If several spray type dehumidifiers are located on different levels, a surge or storage tank to which the return water from each dehumidifier can be taken is required Should the water level in the pan of the dehumidifiers be low in relation to that of the surge tank, return water pumps will be required, and these pumps will have to be operated until the water supply lines are drained in order to prevent flooding of the lower dehumidifiers. Where spray dehumidifiers are on the same level, equalizing lines between the pans may be required if a storage tank is not provided.

All of the various pieces of equipment from the outdoor air intake through the fan usually are connected together by sheet metal casings. Frequently the building structure or specially constructed walls or partitions may be used to form all or a portion of the casing. In any case the casing or connection must be sufficiently sturdy for the required duty. Sheet metal work must be well braced not only to prevent bellying or vibration under pulsations in air flow but to withstand the abuse of normal usage. Casings should be adequately braced wherever access doors are installed and all large panels should be adequately reinforced by angle iron.

Each apparatus layout is to be made with accessibility in mind. Where cooling coils are used space for removing and repairing or replacing the coils should be provided. Adequate space is to be provided for the servicing and replacement of eliminators. Filters must be so located that the proper cleaning, replacement or routine servicing can be accomplished without difficulty. Free access to the bearings of all moving machinery is a necessity. Provisions should be made for the complete removal and replacement of any part of the system that is subject to wear, deterioration or damage, whether it may be filter, fan wheel, motor, rotor, pump impeller or heat transfer surface.

DESIGN PROCEDURE

The customary design procedure is outlined herewith. For simplification the procedure is set up on the basis of a year 'round system. For summer only or winter only systems, the unrelated parts are to be omitted.

1. Selection of design conditions (inside and outside).
   a. Summer.
   b. Winter.
2. Determination of outside air requirements.
3. Determination of cooling load.
   a Room sensible heat gain.
   b. Room latent heat gain.
   c. Room total heat gain.
   d. Grand total heat gain.
4. Determination of heating load.
   a. Room sensible heat loss.
   b. Room moisture loss.
   c. Humidification requirement.
   d. Total heating requirement.
5. Determination of apparatus dew-point and dehumidified or humidified air quantity.
   a. Summer (full load and part load).
   b. Winter.
6. Supply air temperature difference and quantity.
   a. Summer.
   b. Winter.
7. Equipment selection.
8. Equipment layout.

The foregoing steps are merely typical. Many applications will require at least a preliminary investigation of some of the latter steps before proceeding with the earlier steps.

CHAPTER 21

Unit Heaters, Unit Ventilators, Unit Humidifiers

Unit Heaters; Classification; Electric, Gas, Oil, Coal Fired Types, Air Velocities and Distribution; Ratings; Temperatures; Location; Application; Steam Heater Connections and Boiler Capacity; Unit Ventilators; Ratings; Heat Requirements; Application; Window Ventilators, Unit Humidifiers

DESCRIPTIONS of heating, cooling, ventilating, humidifying, and dehumidifying systems are given in other chapters. This chapter deals with unit heaters, unit ventilators, and unit humidifiers. Cooling units, unit air conditioners, and attic fans are described in Chapter 22.

Definitions

The generally accepted meaning of the word *unit* in the terms unit heaters, unit ventilators and unit humidifiers is that of a factory made, encased assembly of, the functional elements indicated by its name. Such units can be shipped complete or in sections so that the only field work necessary is the assembling of the sections, providing proper supports and connecting the unit to sources of heat (or fuel), power and water supply, and, if necessary, to vent pipes for combustion gases.

The term *unit heater* describes an assembly of elements whose principal function is heating. The essential elements of a unit heater are a fan, a heater, a housing, and outlet vanes or diffusers.

The term *unit ventilator* describes an assembly whose principal function is to ventilate. It may serve to circulate air within the space, or to introduce air from without the space, or may accomplish both purposes. The essential elements of a unit ventilator are a fan, a heater, a set of dampers, a housing and outlet vanes or diffusers.

The term *unit humidifier* describes an assembly of elements whose principal function is to humidify. The essential element of a unit humidifier is an atomizer or evaporator. To this may be added a fan, a heater, outlet vanes or diffusers, and a housing to enclose the various parts.

Additional generally accepted definitions[1] of various units are given as follows:

1. A *Heating Unit* is a specific air treating combination consisting of means for air circulation and heating within prescribed temperature limits.

2. A *Heating Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for heating and maintaining humidity within prescribed limits

3. A *Humidifying Unit* adds water vapor to and circulates air in a space to be humidified.

4. A *Free Delivery Type Unit* takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.

5. A *Pressure Type Unit* is for use with one or more external elements which impose air resistance.

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[1]Standard Method of Rating and Testing Air Conditioning Equipment, prepared by a Joint Committee of the *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electrical Manufacturers' Association* and *Air Conditioning Manufacturers' Association*, *ASRE Circular 13-42*

UNIT HEATERS

Classification

The various types of unit heaters which are at present available can usually be classified according to one of the three following methods:

1. *By type of heater.* Under this classification there are three types of heating elements to be considered: (*a*) *the steam or hot water type*, (*b*) *the electric type*, and (*c*) *the direct fired type* which may be *gas*, *oil*, or *coal fired*.

2. *By type of fan.* Under this classification there are two types of fans to be considered: (*a*) the *disc or propeller type* which may be equipped with a horizontal or vertical shaft, and (*b*) the *centrifugal type* which may be designed for *horizontal* or *vertical blow*.

3. *By arrangement of elements.* Under this classification there are two types of heaters to be considered: (*a*) the *draw-through type*, in which the fan draws air through, and (*b*) the *blow-through type*, in which the fan blows air through the heater.

Unit heaters, except during the emergency, are available in any combination of the three preceding general classifications. For example, the steam or hot water type may be secured with either the disc or centrifugal types of fans, and in either the draw-through or blow-through types.

Unit heaters also vary in other minor respects. For example, steam and return inlets and outlets may be located on the top and bottom respectively or on the same side of the unit. Some small units are supported by the piping and some have independent supports. The heating surface of steam or water types may be constructed of non-ferrous or of steel pipe with, or without, extended surfaces; or it may be made of cast-iron sections, or of built-up sections of the cartridge or automotive type.

There are a variety of applications which are favorable to the use of *electric unit heaters*. For supplemental heat in residence bath rooms, for the heating of ticket booths, watchmen's offices, factory offices, locker rooms and other isolated rooms scattered over large areas, their use is peculiarly adaptable. They are particularly useful in isolated and unattended pumping stations or pits where they may be thermostatically controlled to prevent freezing temperatures from occurring.

Gas fired unit heaters find application in industrial plants, offices, stores, garages, in fact in almost every location where steam type units are used. The installation cost of gas fired units is usually much less than that of any other type.

Oil fired unit heaters are used in industrial plants, garages and commercial buildings.

Coal fired unit heaters are of finned, welded steel, or cast-iron construction and equipped with centrifugal blowers. They are usually stoker-fired to insure proper firing of fuel. They are used principally in large industrial plants such as foundries or assembly plants, and provide a very economical source of heat, both from the installation and the operating standpoint.

Outlet Velocities

Outlet velocities of unit heaters vary from about 400 to 2500 fpm depending upon the type of unit and the distance to which it is desired to project the air. Noise and drafts must be considered in the choice of air velocities, since both increase with air velocity.

Velocities and decibel ratings for the various types of unit heaters are given in Table 1.

The *centrifugal fan type*, as illustrated by Figs. 1 and 2, delivers air at high velocity (1500 to 2500 fpm) and is equipped with outlets adjustable to deliver air in several directions. It is able to project the heating effect over distances of from 30 ft to as much as 200 ft from the unit. However, it is more suited to long than short blow, except when mounted high above the floor with discharge at a sharp angle downward. As a rule it can be located at considerable distances from other units thus reducing the piping and conserving floor space usually allotted to heating equipment.

The *propeller fan type* is divided into two classes of *horizontal* and *vertical blow*. The horizontal blow type, illustrated by Fig. 3, discharges air at from 400 to 1000 fpm. It takes care of medium distances of blow, up to approximately 100 ft where mounting height is not too great and where other conditions, such as final temperature and mass of air handled by the unit, are favorable.

The *vertical blow* unit, illustrated by Fig. 4, discharges air at from 1200 to 2200 fpm. It is mounted on the ceiling of a room or up in the roof trusses and blows vertically downward. Various diffusers are available to take care of the height and space conditions. These units are applied to elevations of from approximately 10 to around 50 ft above the floor. Where mounted in roof trusses, as in industrial buildings, they do not interfere with traveling cranes.

TABLE 1. OUTLET VELOCITIES, DISTANCE OF BLOW AND DECIBEL RATINGS FOR VARIOUS TYPES OF UNIT HEATERS

Type of Unit Heater	Outlet Velocities FPM	Distances of Blow—Ft	Decibel Rating
Centrifugal Fan	1500–2500	20–200	34–90
Horizontal Disc Fan	400–1000	30–100	44–84
Vertical Disc Fan	1200–2200	10–50	50–82

A code[2] governing the number of sizes of propeller fan units as well as standardization of fan motors and the method of specifying outlet velocities has been adopted.

Air Outlets

In order to direct the air to points desired and to diffuse the air to avoid drafts, it is common practice to equip unit heaters with directional outlets, adjustable louvers or fixed types of diffusers.

RATINGS OF UNIT HEATERS

It is standard practice to rate unit heaters on the basis of the amount of heat delivered by the air in Btu per hour above an entering air temperature of 60 F. This applies to all types of unit heaters, the steam or hot water type, the electric type and the direct fired type. There are, however, other factors which must be taken into account, especially when an attempt is made to compare one type of heater with another. These are the temperature of the heating element and the velocity of air through it. Consideration is given to these factors in the discussion of ratings for each type of unit heater in the following paragraphs.

[2]Standards for Propeller Type Unit Heaters prepared and adopted by the *Industrial Unit Heater Association*, June, 1938.

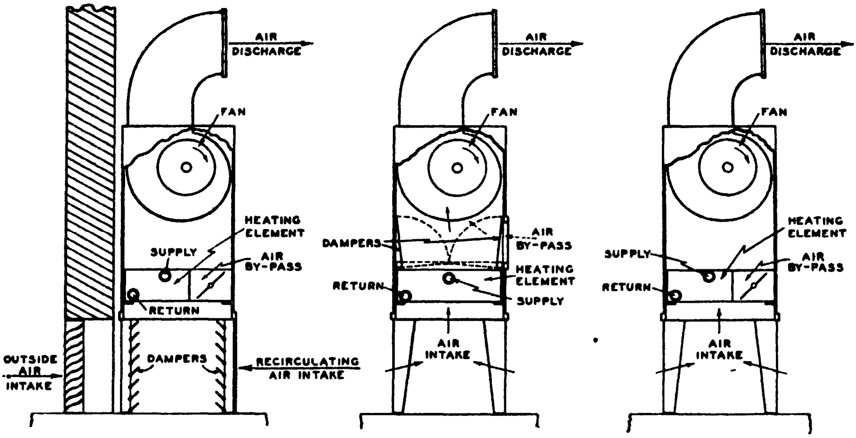

FIG. 1. FLOOR MOUNTED UNIT HEATER, HOUSED TYPE FAN

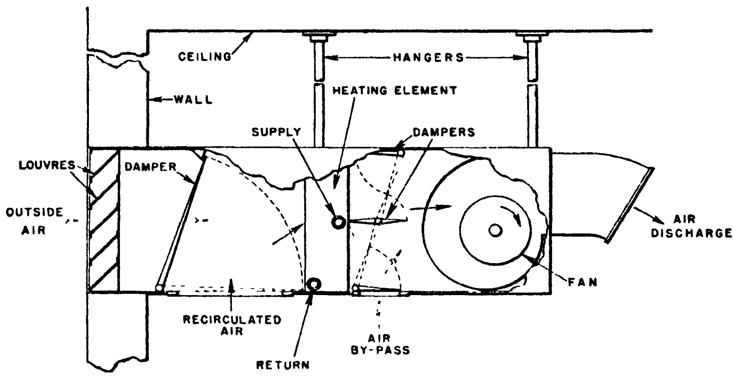

FIG 2. SUSPENDED TYPE UNIT HEATER, HOUSED TYPE FAN

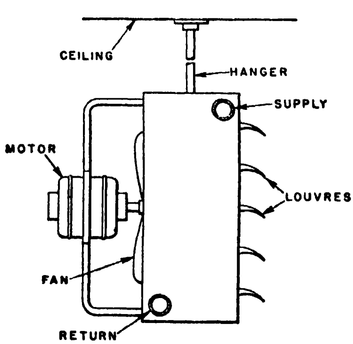

FIG. 3. SUSPENDED UNIT HEATER

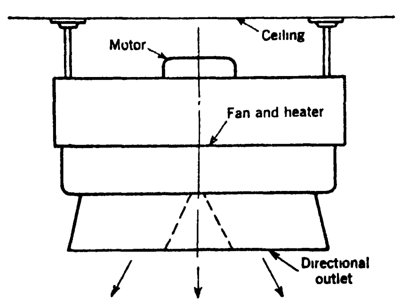

FIG. 4. VERTICAL BLOW UNIT HEATER

Ratings for Steam or Hot Water Type

The rating of steam type unit heaters has been standarized by a code[3] in which steam at 2 psi and air entering at 60 F are used as the standard basis of rating. The capacity of a heater increases as the steam pressure increases, and decreases as the entering air temperature increases. The heating capacity for any condition of steam pressure and entering air temperature other than standard may be calculated approximately from any given rating by the use of factors in Tables 2 and 3. Table 2 is used for the blow-through type and Table 3 for the draw-through type of unit.

Ratings for Electric Type

Electric type unit heaters are usually limited in size to a maximum of about 9 kw capacity. They consist of resistance type heating elements either of the radiant or convector type. The convector type may have a fan for forced circulation or may be of the gravity type. Electric heaters are made in the built-in wall model, and the free standing or portable types.

Electric unit heaters are rated on the energy input to the heater, expressed in terms of kilowatts, Btu or EDR. Quite often all three ratings are given in parallel columns in the catalogs.

Ratings for Gas Fired Type

The gas fired type of unit heater can be secured either with the disc or propeller type fans or with the centrifugal type. They may be secured in a wide range of sizes from 45,000 to more than 1,650,000 Btu output capacity and in floor or suspended models.

Gas fired unit heaters are usually rated in terms of both input and output according to the approval requirements of the *American Standards Association.*

Ratings for Oil Fired Type

The oil fired type of unit heater is usually equipped with the centrifugal type of fan only and can be secured in sizes ranging from 125,000 to 1,650,000 Btu per hour output capacity in standard units. It is furnished in either the floor mounted or in smaller sizes in the suspended type. Greater capacities are available in coal fired units converted to oil burning.

Ratings for Coal Fired Type

The stoker fired type of unit heater can be secured in ranges of from 300,000 to 6,000,000 (or more) Btu per hour output capacity. Ratings are based upon the delivered output at the heater outlet.

Effect of Resistance Upon Capacity

Unit heaters are customarily rated as free delivery type units. If outside air intakes, air filters, or ducts on the discharge side are used with the unit a reduction in air and heating capacity will result because of this added resistance. The percentage of this reduction in capacity will depend upon the characteristics of the heater and on the type, design and speed of the fans so that no specific percentage reduction can be assigned for all heaters at a given added resistance. In general, however, disc or propeller fan type units will experience a larger reduction in capacity than

[3]Standard Code for Testing and Rating Steam Unit Heaters (A.S.H.V.E. TRANSACTIONS, Vol 36, 1930, p. 165), prepared by a Joint Code Committee of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *Industrial Unit Heater Association* and adopted 1930.

TABLE 2. CONSTANTS FOR DETERMINING THE CAPACITY OF BLOW-THROUGH TYPE UNIT HEATERS FOR VARIOUS STEAM PRESSURES AND TEMPERATURES OF ENTERING AIR

(Based on Steam Pressure of 2 lb Gage and Entering Air Temperature of 60 F)

Steam Pressure Lb per Sq In	Temperature of Entering Air											
	−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
0	1.538	1.446	1.369	1.273	1.191	1.110	1.034	0.956	0.881	0.809	0.739	0.671
2	1.585	1.495	1.405	1.320	1.237	1.155	1.078	1.000	0.926	0.853	0.782	0.713
5	1.640	1.550	1.456	1.370	1.289	1.206	1.127	1.050	0.974	0.901	0.829	0.760
10	1.730	1.639	1.545	1.460	1.375	1.290	1.211	1.131	1.056	0.982	0.908	0.838
15	1.799	1.708	1.614	1.525	1.441	1.335	1.275	1.194	1.117	1.043	0.970	0.897
20	1.861	1.769	1.675	1.584	1.498	1.416	1.333	1.251	1.174	1.097	1.024	0.952
30	1.966	1.871	1.775	1.684	1.597	1.509	1.429	1.346	1.266	1.190	1.115	1.042
40	2.058	1.959	1.862	1.771	1.683	1.596	1.511	1.430	1.349	1.270	1.194	1.119
50	2.134	2.035	1.936	1.845	1.755	1.666	1.582	1.498	1.416	1.338	1.262	1.187
60	2.196	2.094	1.997	1.902	1.811	1.725	1.640	1.555	1.472	1.393	1.314	1.239
70	2.256	2.157	2.057	1.961	1.872	1.782	1.696	1.610	1.527	1.447	1.368	1.293
75	2.283	2.183	2.085	1.990	1.896	1.808	1.721	1.635	1.552	1.472	1.392	1.316
80	2.312	2.211	2.112	2.015	1.925	1.836	1.748	1.660	1.577	1.497	1.418	1.342
90	2.361	2.258	2.159	2.063	1.968	1.880	1.792	1.705	1.621	1.541	1.461	1.383
100	2.409	2.307	2.204	2.108	2.015	1.927	1.836	1.749	1.663	1.581	1.502	1.424

Note: To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering and 2 lb pressure.

TABLE 3. CONSTANTS FOR DETERMINING THE CAPACITY OF DRAW-THROUGH TYPE UNIT HEATERS FOR VARIOUS STEAM PRESSURES AND TEMPERATURES OF ENTERING AIR

(Based on Steam Pressure of 2 lb Gage and Entering Air Temperature of 60 F)

Steam Pressure Lb per Sq In.	Temperature of Entering Air											
	−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
0	1.483	1.405	1.329	1.253	1.178	1.105	1.032	0.962	0.892	0.822	0.754	0.688
2	1.520	1.442	1.363	1.290	1.215	1.141	1.069	1.000	0.930	0.861	0.792	0.728
5	1.565	1.485	1 410	1.334	1.260	1 187	1.114	1.045	0.975	0.906	0.838	0.771
10	1.637	1.558	1.480	1.403	1.328	1.253	1.182	1.112	1.042	0.973	0.903	0.838
15	1.688	1.610	1.533	1.458	1.382	1.310	1.239	1.168	1.099	1.028	0.960	0.895
20	1.728	1.649	1.572	1.498	1.421	1.350	1.278	1.208	1.138	1.070	1.002	0.936
30	1.803	1.725	1.648	1.572	1.497	1.423	1.352	1.281	1.212	1.145	1.078	1.010
40	1.864	1.787	1.710	1.637	1.563	1.491	1.420	1.350	1.282	1.215	1.148	1.081
50	1.927	1.850	1.773	1.700	1.628	1.554	1.483	1.416	1.347	1.278	1.211	1.145
60	1.973	1.897	1.820	1.748	1.673	1.601	1.531	1.463	1.394	1.325	1.260	1.194
70	2.018	1.943	1.869	1.795	1.722	1.651	1.582	1.512	1.443	1.377	1.310	1.243
75	2.043	1.970	1.895	1.822	1.750	1.680	1.609	1.540	1.471	1.402	1.333	1.268
80	2.064	1.988	1.914	1.841	1.770	1.698	1.629	1.560	1.491	1.422	1.354	1.288
90	2.102	2.028	1.951	1.878	1.804	1.732	1.661	1.590	1.523	1.457	1.387	1.321
100	2.150	2.071	1.994	1.919	1.845	1.770	1.700	1.630	1.560	1.492	1.425	1.359

Note: To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering and 2 lb pressure.

housed centrifugal fan units for a given added resistance and a given heater will have a larger reduction in capacity as the fan speed is lowered.

Also in disc or propeller type units, not only will the capacity be reduced, but it is possible that the motor may be overloaded if duct or other restrictions increase and it is therefore advisable to check the resistance carefully when designing a unit heater system. The heat output to be expected under other than free delivery conditions should be secured from the manufacturer.

Inlet, Outlet and Space Temperatures with Unit Heaters

In the selection of unit heaters for any particular design, consideration should be given to the temperature of air entering the heaters as well as the temperature to be maintained in the working zone of the space. In general, the temperature differences per foot of elevation, when using unit heaters are less than corresponding variations when using direct radiation[4]. High velocity units will maintain slightly lower temperature differences than low discharge velocity units. Correspondingly, units with lower discharge air temperature will maintain lower temperature differences than units with higher discharge temperatures.

When some *outside air* is introduced, the temperature of the mixture of outside and recirculating air must be calculated and used as the entering air temperature at the heater. Unit heaters connected in this manner perform the function of unit ventilators. For a discussion of this function see the section of this chapter entitled *Unit Ventilators.*

For recirculating heaters located *at the floor* or with *intakes at the floor*, the temperature of air entering the heater should be assumed to be the same as that to be maintained in the room itself. Temperature differences to be expected under these conditions should be less than 0.5 F per foot of elevation when the heaters are operating at their maximum capacity.

For recirculating heaters located *above the floor*, the temperature of air entering the heater may be calculated on the basis of a temperature difference per foot of elevation of 1 F. This temperature difference is based on average operation of unit heaters located above the floor and operating at their maximum capacity.

Location of Unit Heaters

Unit heaters should not be located in a corner or placed close to a wall, since the full effect of the unit will not be available under such conditions if the circulation of air is impeded. The best arrangement is to locate units so that they discharge air parallel to exterior walls, and in a direction which will produce a rotational circulation around the room. This is preferable to directing the discharge against the outside walls.

Various types and makes of unit heaters are illustrated in the *Catalog Data Section* of this edition. As hot blasts of air in working zones are usually objectionable, heaters mounted on the floor should have their discharge outlets above the head line and suspended heaters should be placed in such manner and turned in such direction that the heated air stream will not be objectionable in the working zone. In the interest of economy, however, the elevation of the heater outlet and the direction of discharge should be so arranged that the heated air shall be brought as close above the head line as possible, yet not into the working zone. In

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[4]A.S.H.V E. Research Report No. 958—Temperature Gradient Observations in a Large Heated Space, by G. L. Larson, D. W. Nelson and O. C. Cromer (A.S H.V.E. Transactions, Vol. 39, 1933, p. 243). A.S.H.V.E Research Report No. 1011—Tests of Three Heating Systems in an Industrial Type of Building by G. L. Larson, D. W. Nelson and John James (A.S.H.V.E. Transactions, Vol. 41, 1935, p. 185).

general, the higher the elevation of the unit, the greater the volume and velocity required to bring the warm air down to the working zone, and consequently, the lower the required temperature of the air leaving the unit.

Determining Unit Heater Requirements

The formulae given in the section on Unit Ventilators may be used to determine unit heater capacity requirements.

Application of Unit Heaters

Unit heaters are used principally for commercial and industrial applications such as garages, factories, laboratories, etc. They may also be

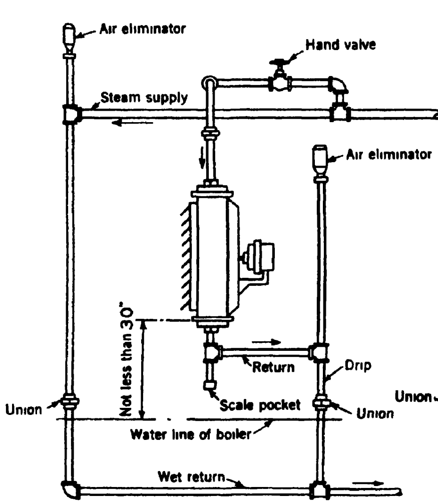

Fig. 5. Unit Heater Connection to One-Pipe Gravity Steam System

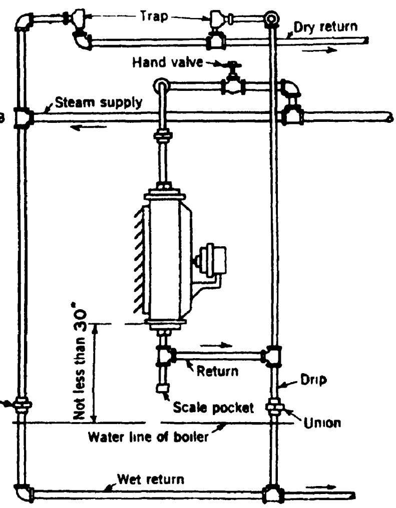

Fig. 6. Unit Heater Connection to Gravity System with Wet and Dry Returns

used for heating finished rooms if properly applied and concealed, and if some consideration is given to the problem of noise.

Unit heaters may also be adapted to a number of industrial processes, such as drying and curing, with which the use of heated air in rapid circulation with uniform distribution is of particular advantage. They may be used for moisture absorption, such as fog removal in dye houses, or for the prevention of condensation on ceilings or other cold surfaces of buildings in which process moisture is released. When such conditions are severe, it is necessary that the unit heaters draw air from outside in enough volume to provide a rapid air change and that they operate in conjunction with ventilators or fans for exhausting the moisture-laden air. See discussion of condensation in Chapter 4.

There are three major factors to consider in the application of unit heaters, namely: (1) location of unit, (2) air distribution, and (3) heating medium.

Piping Connections for Steam Unit Heaters

Piping connections for steam unit heaters are similar to those for other types of fan blast heaters. The piping of unit heaters must conform

strictly to the system requirements while at the same time permitting the heaters themselves to function as intended. The basic piping principles for steam systems are discussed in Chapter 14.

Rapid condensation of steam, especially during heating-up periods, is characteristic of this type of equipment. The return piping must be planned to keep the heating surfaces free of rapid condensation, while the steam piping must be ample to carry a full supply of steam to the surfaces to take the place of that condensed. An adequate size of pipe is therefore essential for all heating surfaces over which there is a flow of forced air. Especially is this true where the fan is operated under start-and-stop control and where all or part of the air is taken from the outside. In such

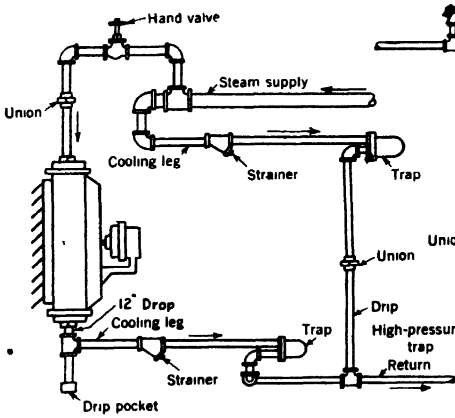

Fig. 7. Unit Heater Connection for Vacuum or Vapor System Discharging Condensation into Dry Return

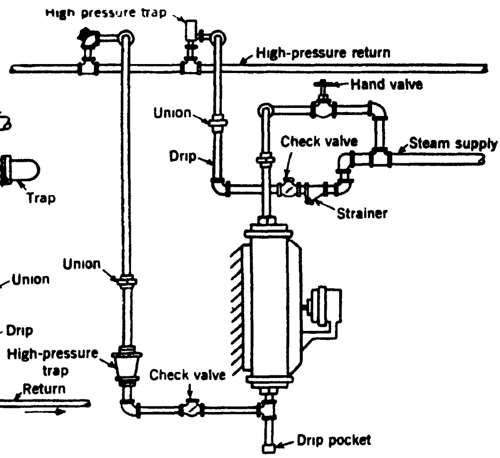

Fig. 8. Method of Connecting Unit Heater to High Pressure Return

installations the condensation rate may vary rapidly and the necessity for ample pipe capacity is particularly important.

A method of connecting a unit heater to a one-pipe gravity system is illustrated in Fig. 5. In cases where the return main is located above the boiler water line, an artificial water line must be created by providing an equalizing loop to prevent steam passing into the return and thus into other units.

Where there is a wet and dry return, a method of pipe connection is shown in Fig. 6. In this case the condensate from the heater and the drip from the supply main drop to the wet return by gravity, while the air passes upward through traps to the dry return and is vented from the entire system by a master trap in any suitable location.

A piping arrangement where both the air and condensate pass through a common return to a boiler, with vent trap or condensate pump and receiver, is shown in Fig. 7. The traps must pass air and condensate rapidly to keep the return piping partially full of water.

Since unit heaters are often constructed with sufficient strength to resist high pressures, use of high pressure steam in them is a common practice. In Fig. 8 the condensate and air reach the return overhead through traps, and check valves are located in the return piping.

For two-pipe closed gravity return systems, the return from each unit

should be fitted with a heavy duty or blast trap, and an automatic air valve should be connected into the return header of each unit heater. Provisions must be made for compensating for the pressure drop by elevating the unit heater above the water line of the boiler or of the receiver.

In pump and receiver systems the air may be eliminated by individual air valves on the heaters, or it may be carried into the returns as in vacuum systems and the entire return system be free-vented to the atmosphere, provided all units, drip points, and radiation are properly trapped to prevent steam entering the returns.

On vacuum or open vent systems the return from each unit should be fitted with a large capacity trap to discharge the water of condensation and with a thermostatic air valve for eliminating the air, or with a heavy-duty trap for handling both the condensation and the air, provided the air finally can be eliminated at some other point in the return system.

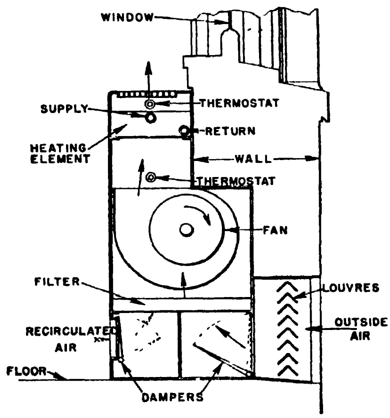

FIG. 9 TYPICAL UNIT VENTILATOR SHOWING ONE OF MANY ARRANGEMENTS OF DAMPERS AND HEATING COILS

For high pressure systems the same kind of traps may be used as with vacuum systems, except that they must be constructed for the pressure used. If the air is to be eliminated at the return header of the unit, a high pressure air valve can be used; otherwise the air may be passed with the condensate through the high-pressure return trap, with its elimination at some other point in the system.

Boiler Capacity for Steam Unit Heaters

The capacity of the boiler should be based on the rated capacity of the unit heaters at the lowest entering air temperature and highest fan speed that will occur, plus an allowance for pipe line losses. It is unwise to install a single unit heater as the sole load on any boiler, particularly if the unit heater motor is started and stopped by thermostatic control. The wide and sudden fluctuations of load that occur under such conditions would require closer attention to the boiler than is usually possible in a small installation. Where oil or gas is used to fire the boiler, it is possible by means of a pressurestat to control the boiler, in response to this rapid fluctuation. In most cases, and particularly where the boiler

is coal-fired, it is advisable to use two or more smaller unit heaters instead of one large unit.

Steam pressures below 5 lb can be used with safety for recirculating unit heaters when their heating surfaces are designed for those pressures, and when proper provision is made for returning the condensate. If units admit air that may be at a temperature below freezing, a steam pressure of not less than 5 lb should be maintained on the heating element, or a corresponding differential in pressure between the supply and returns should be maintained by means of a vacuum.

UNIT VENTILATORS

Unit ventilators are similar in principle to unit heaters, except that they are provided with an arrangement of dampers for introducing outdoor and recirculated air in varying quantities. In general, they are obtainable only in the steam type but unit heaters of all types may be adapted to the unit ventilator principle by the addition of outdoor air and

TABLE 4. TYPICAL CAPACITIES OF UNIT VENTILATORS FOR AN ENTERING AIR TEMPERATURE OF ZERO

Cubic Feet of Air per Minute		Total Capacity in Square Feet, Equivalent Direct Radiation	Capacity Available for Heating the Room, Square Feet Equivalent Direct Radiation	Final Air Temperature Deg F
Anemometer Rating	Condensate Rating			
750	500	214	56	95
1000	750	320	84	95
1260	1000	427	112	95
1560	1250	534	141	95

recirculating dampers. Unit ventilators are used to supply air with a discharge temperature at or slightly higher than room temperature. Also they are provided with an arrangement for introducing outdoor and recirculated air in varying quantities. If the unit is only used for circulating air, then radiators or some other equipment must be provided for heating the room. Unit ventilators are intended primarily for schools, offices, and semi-commercial establishments. A typical unit ventilator is illustrated in Fig. 9. A roof ventilator used for exhausting air is sometimes termed a unit ventilator. For information on roof ventilators, see Chapter 42.

Ratings

Unit ventilators of the steam type are customarily cataloged with two ratings, one the input and the other the output capacity. The first is the heat input to the unit which is determined by measuring the temperature and quantity of condensate and the pressure and quality of the steam. The second is the heat output of the unit which is determined by measuring the quantity of air delivered and the temperature of the air to and from the unit. Table 4 shows the air handling capacities by the two methods of rating[5] and also the approximate heating data. In accordance with the A.S.H.V.E. Standard Code for Testing and Rating Steam

[5]A.S.H V E. Standard Code for Testing and Rating Steam Unit Ventilators (A.S.H.V.E. TRANSACTIONS, Vol 38, 1932, p. 25).

Unit Ventilators the information to be supplied regarding ratings and the basis of rating is the following:

Rating Factors to Be Specified. The rating of the unit ventilator shall specify the following:

a. Final temperature at different entering air temperatures.
b. Total EDR at different entering air temperatures.
c. Air delivered by the unit in cubic feet per minute at the *standard basis of rating* with the fans operated at rated speed, with all air being blown through the heating unit and with the standard louver and grille on the outlet.

The Standard Basis of Rating shall be as follows:

a. Dry saturated steam at a temperature at the unit corresponding to an absolute pressure of 16.7 lb per square inch (218.5 F).
b. Entering air temperature of zero degrees Fahrenheit.
c. Volume delivered in cubic feet per minute converted to standard air at 70 F.

Rating Tables for unit ventilators shall contain the following data in addition to the standard rating, for entering air temperatures from -30 F to $+60$ F:

a. Inlet temperature, degrees Fahrenheit.
b. Final temperature, degrees Fahrenheit.
c. Total EDR at the specified entering temperature.
d Surplus or heating EDR at the specified entering temperature.

Surplus or Heating Equivalent Direct Radiation for the purposes of this code shall be construed to mean difference between the total EDR at a specified inlet temperature and the EDR required to heat the air from that temperature to 70 F.

If no direct heating surface (radiation) is installed to take care of the normal heat transfer losses, and the unit ventilator is to be used for both heating and ventilation, then the combined requirements must be taken care of by the unit ventilator.

Heat Required for Ventilating Only

When all of the air handled by the unit is taken from the outside, the total heat to be supplied is obtained by means of Equations 1, 2, 3, and 4.

$$H = 0.24\, W\, (t_y - t) \tag{1}$$
$$H_v = 0.24\, W\, (t - t_o) \tag{2}$$
$$H_t = 0.24\, W\, (t_y - t_o) = H + H_v \tag{3}$$
$$W = d\, 60\, Q \tag{4}$$

where

d = density of air, pounds per cubic foot.
H = heat loss of room, Btu per hour.
H_v = heat required to warm air for ventilation, Btu per hour.
H_t = total heat requirements for both heating and ventilation, Btu per hour $= H + H_v$.
Q = volume of air handled by the ventilating equipment, cubic feet per minute
t = temperature to be maintained in the room, degrees Fahrenheit.
t_o = outside temperature, degrees Fahrenheit.
t_y = temperature of the air leaving the unit, degrees Fahrenheit.
W = weight of air circulated, pounds per hour.
0.24 = specific heat of air at constant pressure.

From Equations 2, 3, and 4:

$$H_t = H + 0.24\, d\, 60\, Q\, (t - t_o). \tag{5}$$

Example 1. The heat loss of a certain room is 24,000 Btu per hour, and the ventilating requirements are 1000 cfm. If the room temperature is to be 70 F and all air is taken

from the outside at zero, what will be the total heat demand on the unit if it is required to provide for both the heating and ventilating requirements (combined system)?

Solution. Substituting in Equation 5:

$$H_t = 24{,}000 + 0.24 \times 0.075 \times 60 \times 1000\,(70 - 0) = 99{,}600 \text{ Btu per hour}$$

$$t_y = \frac{24{,}000}{0.24 \times 0.075 \times 60 \times 1000} + 70 = 92.2 \text{ F.}$$

If the heat loss of the room is to be taken care of by the direct heating surface, the unit ventilators will be required to warm the air introduced for the ventilating requirements. Therefore:

$$H_v = 0.24\,W\,(t_y - t_o) \tag{6}$$

In this case t_y should be equal to or slightly higher than t. If the unit ventilator were of such capacity as to provide exactly for the ventilating requirements, the direct radiation would be selected on the usual basis. However, it is necessary to employ a unit which may not exactly meet the ventilating requirements, since standard units are usually rated in terms of the volume of air that will be delivered at a certain temperature t_y for an initial temperature of t_o. Therefore a certain amount of heat (H_h) may be available from the unit ventilator for heating purposes, as previously stated, and the amount of equivalent direct heating surface may, if desired, be deducted from the amount required for heating the room.

Heat Required for Ventilating and Recirculating

When part of the air handled by the unit is taken from the room and the remainder from the outside,

$$H_t = 0.24 W_o\,(t_y - t_o) + 0.24\,W_1\,(t_y - t) \tag{7}$$

where

W_o = weight of air, pounds per hour taken from out-of-doors.
W_1 = weight of air, pounds per hour taken from the room.

$$W_o = d_o\,60\,Q_o \tag{8}$$

$$W_1 = d_1\,60\,Q_1 \tag{9}$$

where

d_o = density of air, pounds per cubic foot at temperature t_o
d_1 = density of air, pounds per cubic foot at temperature t.
Q_o = volume of air taken in from the outside, cubic feet per minute.
Q_1 = volume of air taken in from the room, cubic feet per minute.

$$t_y = \frac{H}{0.24\,(W_o + W_1)} + t \tag{10}$$

$$H_t = H + 0.24\,d_o\,60\,Q_o\,(t - t_o) \tag{11}$$

Equations 7, 8, 9, 10 and 11 may be used in the same manner as is illustrated previously for Equations 3, 4 and 5. It may be noted in Equation 11, representing the total heat requirements, that as the quantity Q_o is diminished the heat requirements for the unit diminish very materially.

In Example 1, if one third of its air volume is taken from the outside and two thirds from the room the total heat requirement would be $24{,}000 + \dfrac{99{,}600 - 24{,}000}{3} = 59{,}200$ Btu per hour. Units designed and operated on this principle show an average heat requirement and, there-

fore, a boiler capacity requirement of less than 50 per cent of that required for units taking all their air from the outside.

Heat Required for Recirculating Only

If all of the air is recirculated, the total heat required is the same as the heat loss of the room, or

$$H_t = H = 0.24\ W\ (t_y - t) \tag{12}$$

In *Example 1*, if the quantity of air taken in from the outside is reduced to zero, or all of the air handled by the unit is recirculated, the total heat requirements H_t reduced from 99,600 to 24,000 Btu per hour, or to about one-fourth.

Applications of Unit Ventilators

Items to be considered in the application of unit ventilators are: (1) combination with other means of heating, (2) location of units, and (3) method of venting or exhausting.

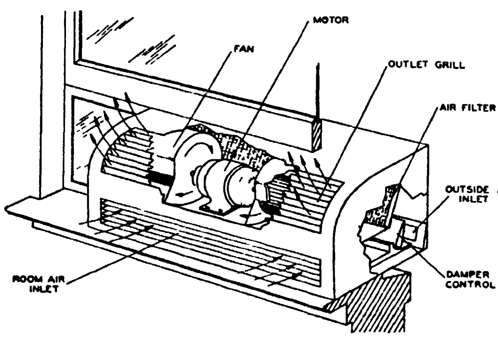

FIG. 10. TYPICAL WINDOW VENTILATOR

In a *split* system the unit is used primarily for ventilation. Air is delivered to the room at or slightly above room temperature, and enough radiation is installed in the room to take care of the normal heat transfer losses. Where the unit ventilator selected has a capacity more than sufficient to warm the air needed to meet the ventilating requirements, a corresponding reduction may be made in the amount of direct radiation installed. The greater the amount of excess capacity of the unit, the more efficient will be the temperature regulation of the room. The split system permits the heating of the room during failure of electric current, since the direct radiators will furnish heat, but it permits a careless operator to avoid operating the ventilating equipment.

The *combined* system employs a unit ventilator with sufficient capacity for both ventilation and the normal heat transfer losses. In such a case no direct radiation is required. It then becomes necessary for the fan to be running whenever the room is heated, but this also gives assurance of ventilation, especially if automatic dampers are used in the air intake from out-of-doors and in the recirculating intake arranged so as to give a certain quantity of air from the outside (commensurate with weather conditions) whenever the unit is operating and after the room is heated. The cost of installation of a combined system is usually less than that of a split system and there is less danger of overheating, but if the electric

energy fails there will be available only the heating effect of the units acting as convectors.

Location of Unit Ventilator

The location of the unit ventilator in a room is important. Wherever possible it should be placed against an outside wall. It is difficult to obtain proper air distribution if the unit is erected either on an inside wall or in a corner of the room. Standard units discharge the air stream upward, but for special cases units may be installed to discharge air horizontally. Units may be set away from the wall or partially recessed into the wall to save space without materially affecting the results. The air inlet may enter the cabinet at the back at any point from top to bottom.

Air Exhaust Vents and Flues

The size and location of the air exhaust vent[6] outlet is important. In many cases the sizes for public buildings are regulated by law. See table of state codes and standards in Chapter 48.

In cases where no codes govern, the location and size of vents is left to the discretion of the engineer.

Best results have been obtained with a velocity through the vent openings nearly equal to that at which the air is introduced into the room, thus maintaining a slight pressure in the room. Calculated velocities at the vent openings of from 600 to 800 fpm produce the best diffusion results from this system. Many states, however, have regulations that will not permit of velocities as high as 800 fpm. If a vent opening at or near the floor is near a desk or place where a person is seated, a velocity of 800 fpm in the vent opening will produce an objectionable draft. In such a case the velocity in the vent opening should not exceed 400 to 450 fpm, although duct velocities are maintained at 600 to 800 fpm if codes permit.

In school buildings provided with wardrobes or cloakrooms the vents may be so located that the air shall pass through these spaces, ventilating them with air which otherwise would be passed to the outside without being used to the best advantage. Many state codes for ventilation of public buildings make this arrangement mandatory.

WINDOW VENTILATORS

A window ventilator illustrated in Fig. 10 consists of filter and motor driven fans enclosed in a cabinet to be mounted on the window sill. These units accomplish ventilation, air cleaning, and air circulation, but have no means of heating the air. The direction of air discharge is manually adjustable for seasonal operation. Their operation is controlled by an electric switch.

UNIT HUMIDIFIERS

Unit humidifiers which may be procured in the market at the present time, fall into four general classifications, depending on the method of producing the humidity. These are as follows:

(1) *Nozzle Type*, (2) *Rotary Type*, (3) *Cascade Type*, and (4) *Heater Type*.

In the *nozzle type* of humidifier water is sprayed into the air and

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[6]A.S H.V E. Research Report No. 936—Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson and R. W. Kubasta (A.S.H.V.E. Transactions, Vol 38, 1932, p. 463). A.S.H.V.E. Research Report No. 1017—Air Supply to Classrooms in Relation to Vent Flue Openings, by F. C. Houghten, Carl Gutberlet and M. F. Lichtenfels (A S.H.V.E. Transactions, Vol. 41, 1935, p. 279).

evaporation is effected by adiabatic heat transfer. Units of this type in simplest form spray a fine mist of water directly into the air in a space. They are used to a great extent in the textile industry.

To the simple type of nozzle unit may be added the following accessories: a water heater whose function is to increase the vapor pressure, an air heater to heat the air either before or after humidification takes place, a screen or air filter over which the water is sprayed and through which the air is drawn for intimate contact with the water, a fan to create an air stream and to deliver air to the space to be humidified, and a housing to enclose all the elements. This type of unit is procurable in practically all of the variations mentioned. It has the disadvantage of clogging the nozzles and must be serviced continually.

Fig. 11 illustrates a nozzle unit having a humidifying capacity sufficient for a residence or small building. These units usually include air filters and in some cases provide ventilation air by means of an outside air duct

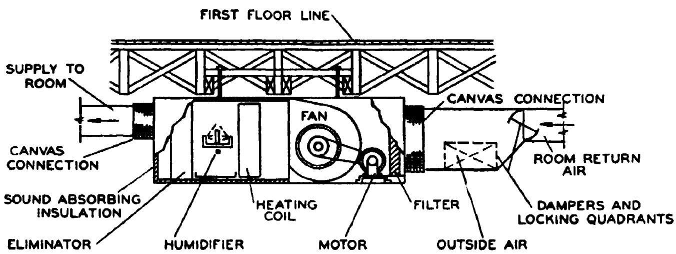

FIG. 11. TYPICAL UNIT HUMIDIFIER OF THE SPRAY TYPE WITH STEAM COIL TO PREHEAT THE AIR FOR RESIDENCES

connection to the unit. The units are available for either floor or ceiling mounting and are usually placed in a central location in the basement with short supply and return duct connections from the first floor. Room air is brought into the unit through the return duct connection and first passes over a tempering coil heated by steam or hot water, then is humidified by passing through some type of spray humidifier. Surplus moisture is removed by an eliminator and the humidified air is delivered to the room through a duct connection. Since a large percentage of the tempering coil capacity is transformed into latent heat during the humidifying process, the unit does not generally eliminate any existing steam radiation but does tend to improve comfort conditions by supplying heating during the off-period of furnace operation[7].

In the *rotary type* of humidifier the spray is created by rotating vanes or discs which throw the water by centrifugal force and in so doing break it up into a fine mist. In all other respects, this type of humidifier is similar to the nozzle type. It has the advantage over the nozzle type of being less liable to become clogged.

In the *cascade type* the humidification takes place by water falling in sheets over a series of baffles or trays. This type is usually furnished with a fan, air heater, and air filter all enclosed in a housing.

In the *heater type* of humidifier the water is heated either to the boiling point or to a temperature at which the water vapor readily passes into

[7]Estimating the Humidification Requirements of Residences, by W. H Severns (Papers Presented at the First Annual Conference on Air Conditioning, University of Illinois, *Engineering Experiment Station Circular*, No. 26, October. 1936).

the air stream. There are many variations of this type of humidifier. In the simplest form the heating element is placed in a pan or vessel of water and the vapor passes from the surface of the water to the stream of air. The heating medium may be steam, hot water, gas, oil or electricity.

A modification of this type of humidifier is the combination of spray nozzle and heater type in which the water is sprayed over a hot surface and evaporated. It has the disadvantage of accumulating scale on the surfaces of the vessel or heating surface.

For a complete discussion of the principles of the various methods of humidification refer to Chapter 26.

CHAPTER 22

Unit Air Conditioners, Unit Air Coolers, Attic Fans

Definition of Types, Unit Air Conditioners, Heating, Humidifying, Cooling and Dehumidifying, Filtering, Ventilating, Types of Units, Application, Ratings, Unit Air Coolers, Design and Performance, Types of Units, Ratings, Defrosting, Economics, Attic Fans

AN assembly of functional elements, as indicated by the name, comprises the unit air conditioner or unit air cooler. Such a unit when complete in itself, employing its own direct means of air distribution and source of refrigeration is known as a self-contained unit. When used in various combinations with remote sources of refrigeration, heat or air supply, it is termed a remote unit, indicating that the source of refrigeration is not contained within the unit cabinet. Either the self-contained or remote type units may be located within or without the conditioned area, and are a separate classification from the central plant type of system, as described in Chapter 20.

The code, Standard Method of Rating and Testing Air Conditioning Equipment[1], defines the various types of unitary equipment:

1. A *Cooling Unit* is a specific air treating combination consisting of means for air circulation and cooling within prescribed temperature limits.
2. An *Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for maintaining temperature and humidity within prescribed limits.
3. A *Cooling Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for cooling and maintaining temperature and humidity within prescribed limits.
4. A *Self-Contained Air Conditioning or Cooling Unit* is one in which a condensing unit is combined in the same cabinet with the other functional elements. Self-contained air conditioning units are classified[2] according to the method of rejecting condenser heat (water cooled, air cooled, and evaporatively cooled), method of introducing ventilation air (no ventilation, ventilation by drawing air from outside, ventilation by exhausting room air to the outside, or ventilation by a combination of the last two methods), and method of discharging air to the room (free delivery or pressure type).
5. A *Free Delivery Type Unit* takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.
6. A *Pressure Type Unit* is for use with one or more external elements which impose air resistance.
7. A *Forced-Circulation Air Cooler* is a factory encased assembly of elements by which heat is transferred from air to evaporating refrigerant[3].

UNIT AIR CONDITIONERS

This equipment takes the form of an encased assembly including the apparatus necessary to perform either some or all of the functions of cooling, dehumidifying, filtering, ventilation, air circulation, heating, and humidifying. Control of air conditions is provided by manual

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[1]Prepared by a Joint Committee of the *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electrical Manufacturers' Association*, and *Air Conditioning Manufacturers' Association* (*A S R E Circular* No 13-42).

[2]Standard Method of Rating and Testing Self-Contained Air Conditioning Units for Comfort Cooling prepared by a Joint Committee of the *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electrical Manufacturers' Association*, and *Air Conditioning Manufacturers' Association* (*A S R.E. Circular* No. 16).

[3]Defined in Proposed *A.S R E* Standard Methods of Rating and Testing Forced-Circulation Air Coolers for Commercial and Industrial Refrigeration (*A.S R E Circular* No. 25-43).

switches, manual dampers, and automatic devices, in combination. The controls are either mounted on the units, or in some suitable location in the conditioned area. See Chapter 33 for a discussion of controls.

The various conditioning elements and their functions, which produce the required effects on air, are discussed under separate headings.

Heating

Heating is usually accomplished by means of a heating coil in the unit, supplied with steam or hot water from an external source. Electric strip heaters may also be considered, where installation and operating costs justify their use. They are often used in special control applications. Reverse cycle heating as described in Chapter 24 has been developed as a feature in some unit air conditioning equipment, but this form of heating has not yet been generally adopted.

Humidifying

Adding moisture to the air involves the absorption of heat, by the humidifying water, for conversion to water vapor. Heat may be supplied by heating the humidifying water, or supplied from the air to be humidified. In the latter case the air may be warmed or the water finely divided to present a large evaporating surface to the existing air. The source of heat may be from electricity, steam or hot water coils, or from a heat transfer surface as in a direct-fired unit. Occasionally vapor is added directly by means of steam jets but this is usually confined to industrial applications because of the presence of some odor from the steam. Methods and types of humidifying apparatus are dealt with in detail in Chapter 26.

For unit application, some of the available methods are the spray nozzle, or the atomizing nozzle, the impact-jet, the drip screen and the evaporating pan. The first method is usually employed where humidification on a large scale is desired, as with large remote industrial units. Eliminator plates are necessary and the water supply may be recirculated by a pump or wasted to the drain. The drip screen, impact-jet, evaporator pan or small atomizing jets are used when humidification is desired in the smaller remote and self-contained units.

Wetted surfaces exposed to the air stream and utilizing the capillary action of water on porous substances such as fabrics and ceramics, are used for adding moisture to air. Frequent cleaning or replacement is necessary to avoid closing of the pores and to maintain freedom from odors or growths on the humidifying element.

Cooling and Dehumidifying

These two functions of air conditioning are usually performed simultaneously, although both may be done separately. For example, air may be dehumidified or dehydrated without sensible cooling by the process of adsorption. Sensible cooling of air may be accomplished without dehumidifying by maintaining the cooling surface temperature above the dew-point temperature of the air to be treated. Chapter 23 explains these fundamentals in detail.

Unit equipment commonly utilizes heat transfer surface such as pipes or coils, through which a cooling medium as cold water or a refrigerant, is circulated. Types and methods of generating cooling mediums are covered in Chapter 24. Brine or water sprays may be used if desirable, either separately or in combination with coil or pipe surface. However, these methods find their best application in the larger, remote type units.

Surface temperature and area of the coil or pipe, air volume and velocity, and spray temperature and volume are some of the controlling factors in unit air conditioner design and application. Ice as a cooling medium is practical but is seldom used in connection with units.

Filtering

Cleaning of outside or recirculated air discharged to the conditioned area is one of the important functions of air conditioning, and air filters should be included on all units which condition air for comfort. Protection is also afforded the cooling and heating coils as well as the condenser coil in the case of the air cooled type.

Means of filtering may vary from the lint screen to the electrostatic filter, with the degree of efficiency covering a wide range. Inexpensive throwaway type filters lend themselves well to the compact design of the unit conditioner. Air cleaning devices form the subject of Chapter 28.

Ventilating

Provision for the introduction of outside air should be an essential part of unit conditioner design. Odors and air vitiation are avoided and better load control is possible when a positive means of introducing outside air is available.

On the small, air cooled room units, it has been found practical to control the air in such a way as to permit variation from all recirculated air to 100 per cent outside air. An added feature is a dampering arrangement whereby it is possible to exhaust air from the room to remove smoke and generally ventilate the space.

A code[4] sponsored by a Joint Committee of the A.S.H.V.E. and the *American Society of Refrigerating Engineers* may be consulted, although it should be realized that individual applications may often show a need for ventilation in excess of these minimum requirements.

Types of Units

Unit air conditioners fall into two general classifications, depending on the location of the refrigeration source. Those units having the condensing unit completely enclosed in the same cabinet as the evaporotor are known as self-contained types, while those units having the condensing unit remotely located from the evaporator and requiring piping of refrigerant from the condensing unit to the evaporator and return are known as remote types.

Self-contained units are further divided into two groups in accordance with their condensing mediums, being either air-cooled or water-cooled. Evaporative cooled types are included in this latter class.

The air-cooled types are small in capacity, ranging from ⅓ to 1½ hp. Their principal application is for conditioning such spaces as hotel rooms, offices and residential living quarters. A duct connection between the unit and an outside window or ventilated air shaft is required to permit disposal of the heat extracted from the conditioned area. The unit may stand in front of the window or be mounted on the window sill. Various styles and types of windows are encountered which increase the difficulty of making the window connections. The evaporation of condensate on the condenser coils, as a means of disposing of this moisture, tends to increase the condensing capacity and reduce the operating head pressure.

[4]Code of Minimum Requirements for Comfort Air Conditioning (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 27). Reprints of this code are available at $0.10 a copy.

Some units add supplementary water so that increased capacity may be obtained from constantly wetted condenser coil surface. Connections to an electrical outlet may be by means of a conventional cord and plug or a permanent electrical connection, depending on local code rulings pertaining to the installation of small motors. The exterior finish of the unit in metal, wood or fabric is decorated to harmonize with office or bedroom furnishings.

A unit of the air-cooled condenser type for floor mounting is shown in Fig. 1. Of the two fans shown, the lower one acts as condenser air fan, and in some units this fan is arranged with slingers for discharging condensate on the condenser coil while the upper fan discharges air into the conditioned area. A feature of the design shown in Fig. 1 is that the condensate from the cooling coil is sprayed over the condenser surface and vaporized, thus eliminating the need for drain connections. A simple dampering arrangement is generally provided for exhausting some air from the room, in addition to introducing outside air and recirculating

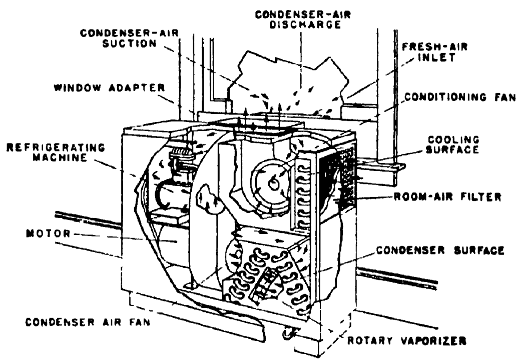

FIG. 1. SELF-CONTAINED AIR-COOLED UNIT AIR CONDITIONER

required amounts of air. It is possible to remove the equipment for winter storage or utilize the ventilating features for winter operation.

Water or evaporative-cooled units start at 1 hp in size and may be as large as 30 hp. Condensers requiring water use either city water, well water or recirculating water from a cooling tower. Evaporative condensers are seldom used on units under 5 hp.

The heat generated by the compression of refrigerant gases and that given off by the electric motor is removed from the compressor compartment in four ways: by the use of a water coil in the compressor compartment; by means of utilizing the cold suction gases; by drawing part of the return air through the compressor compartment and finally by circulating room air through the compressor compartment by means of a fan attached to the motor shaft.

The smaller water-cooled units have somewhat the same application as the air-cooled units. Water and drain facilities must be available and, although window connections are not required for heat disposal, it is desirable to have an outside air connection for ventilation purposes. Electrical wiring is generally permanently connected.

Up to 7½ or 10 hp in size the units are usually styled for locating directly in the conditioned area. Above this size the tendency is to

locate the equipment adjacent to the conditioned area, with supply and return duct connections. Compressor compartment heat is removed by the same methods described previously.

On water units of the vertical type of 5 hp and under, use is made of air distributor headers, equipped with directional louvers, on one or more sides, for discharging the air directly into the conditioned space in which the unit is located. Above 5 hp this method usually becomes impractical and a system of ducts is employed. Arrangements for outside air supply are similar to those in central plant design, although simpler, hand operated dampers are usually employed.

Accessory equipment for the self-contained unit conditioners includes heating coils, humidifiers, and controls for utilizing the unit for winter

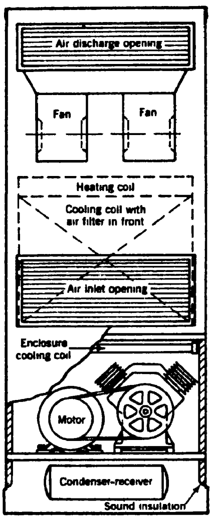

FIG. 2. SELF-CONTAINED WATER-COOLED AIR CONDITIONER

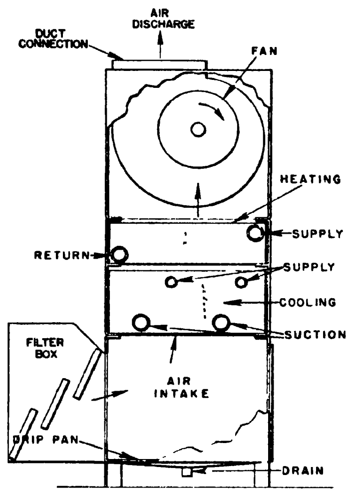

FIG. 3. VERTICAL REMOTE TYPE UNIT AIR CONDITIONER

heating and ventilating. Refinements of dust and odor control and constant temperature and humidity regulation are considered to be special application problems with this type of equipment.

A typical unit with water-cooled condenser is illustrated in Fig. 2. The header arrangement permits air distribution in several directions, and in such a way as not to fall on the room occupants. All side panels are removable for complete access to equipment. It is only necessary to bring water, drain and electrical service to the unit, and a source of heat if desired.

Remote type units cover a much broader range of size and application and generally are used in connection with or in lieu of a central plant system. They may be used individually or in groups in the place of self-contained equipment.

Without the weight of the refrigerating equipment, these units may be suspended from roofs or ceilings, or located wherever space permits on roofs and in basements, or in the conditioned area itself. All combinations of filtering, humidification, cooling and heating may be

employed, with control as elaborate or as simple as is required. The remote unit is particularly adaptable where a variety of application conditions is to be met from a single source of refrigeration, such as the modern industrial plant which may have a laboratory, executive offices, a cafeteria, together with various processing departments.

The cooling and heating mediums, consisting of a refrigerant, chilled water or brine, and steam or hot water, are piped to the units, and transfer takes place by means of coils, or in the case of air washers, by means of water or brine sprays in the air stream, or a combination of the two.

The floor mounted or vertical type and the suspended or horizontal type of remote units are respectively shown in Figs. 3 and 4. The cabinets are generally of sheet steel, insulated to prevent heat transfer, finished to

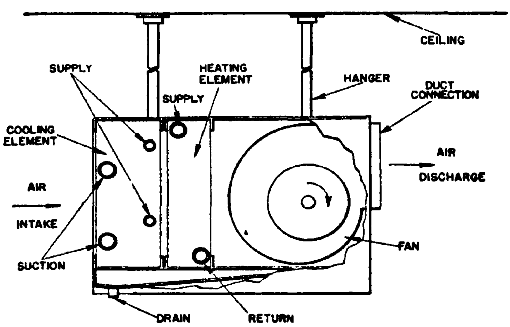

FIG. 4. HORIZONTAL REMOTE TYPE UNIT AIR CONDITIONER

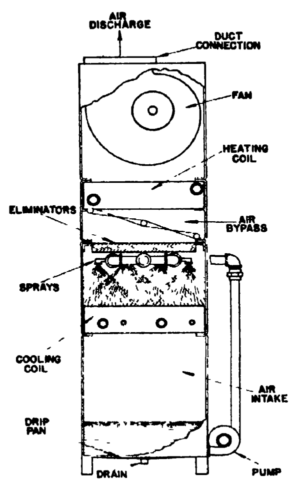

FIG. 5. (*right*) SPRAY TYPE REMOTE UNIT AIR CONDITIONER

prevent corrosion and suitable for applying additional decoration if desired.

A spray type remote unit is illustrated by Fig. 5. Many designers prefer the air washing and coil wetting features. Air is circulated by means of attached or built-in fans, delivering the conditioned air through a system of ducts which include outside air connections if desired.

Some types of units, mostly those suitable for suspension, employ a propeller fan such as in Fig. 6. These are located in the conditioned area, are without ducts and seldom do more than cool and dehumidify, due to the design of the propellor fan for moving air against low resistance. Generally these units are provided with a lint screen instead of filters to limit resistance to air flow.

Individual floor mounted type remote units are available for use in offices or hotel rooms. Similar in appearance but somewhat smaller in size than the self-contained room unit, this form of equipment may be grouped:

1. Mechanical, year 'round type containing blowers, filters, humidifiers and coils, with outside air connection. Manual or automatic controls provided with each unit.

2. Mechanical, semi-year 'round type with no outside air connection, containing

blowers (filters optional), and coils. Usually used where there is an existing radiation or ventilating system. Controls provided with each unit.

3. Non-mechanical type containing coils, using air ejected under pressure from a remote source for inducing circulation over coils (see Chapter 20). Manual or automatic control of air temperature only is provided. Both summer and winter air conditioning functions may be performed by the one unit.

One coil for cooling and heating may be provided or a single coil used, through which hot water in winter and cold water in summer is circulated. All three types of systems require remote sources of refrigeration, with the first group obtaining outside air from the window, the second group having no outside air unless used in connection with a central plant or ventilating system and in the case of the third group, all air delivered under pressure is outside air.

Various combinations or alterations of these room units are available. Different filtering, humidifying and air delivery methods are employed

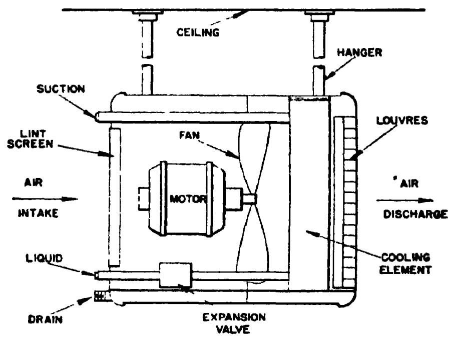

FIG. 6. SUSPENDED PROPELLER FAN TYPE UNIT AIR CONDITIONER

to achieve the desired conditions. A typical remote floor type room unit air conditioner is shown in Fig. 7.

Application

In the application of unit air conditioners it is important to consider several factors:

1. Location of equipment.
2. Air distribution.
3. Multiple units versus central station system.
4. Multiple remote unit system versus a self-contained unit system.
5. Methods of control.
6. Methods of conserving water.
7. Code limitations.

In choosing locations for air conditioning units, consideration must be given to the characteristics and use of the conditioned space; type of system contemplated; duct locations; sources of power, water, refrigeration, heating and drainage; and accessibility of equipment and system for maintenance.

Where units are placed within the conditioned area, particular attention must be given to air distribution, sources of outside air and convenience to service sources and facilities previously noted. Skill and ingenuity are required to produce a neat appearing, inconspicuous job

without sacrificing quality from an engineering point of view. Furring in of duct work, and building in and refinishing to match existing furniture and fixtures are ways this may be accomplished. Where location of equipment outside of conditioned space is possible, use may be made of storage rooms, halls, basements or any space less valuable than that to be conditioned. Less emphasis need be placed on the appearance of equipment.

However, choices of equipment location are frequently influenced by such economic factors as long runs of insulated ducts in order to locate units near refrigeration sources, versus short duct runs but long extensions of service facilities. Care must be taken that equipment will not be damaged by climatic conditions.

Air distribution from units located within the conditioned area is by means of grilles, either fixed or adjustable and mounted in air distribution

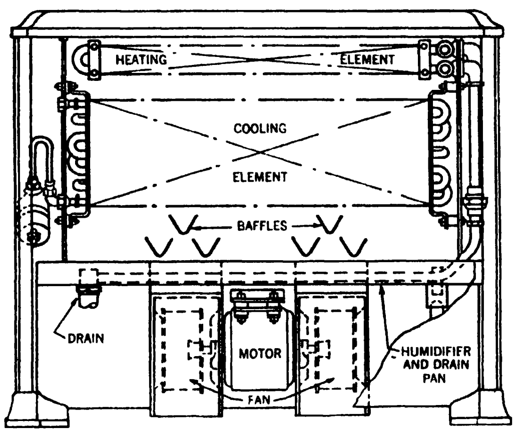

FIG. 7. REMOTE FLOOR TYPE ROOM UNIT AIR CONDITIONER

headers, either furnished with the unit or constructed at the installation to meet the needs of the application under consideration. A system of ducts and distribution grilles may also be used, *similar* to the arrangement used when equipment is outside of the conditioned area. The problem of securing a supply of outside air is often a difficult one. The proper design of ducts is of major importance for good air distribution. Chapters 30 and 31 are devoted to these subjects.

In the analysis of any large structure to be air conditioned which is divided into small spaces, such as an office building or hotel, a comparison should be drawn between the use of units and a central plant system. This study is one of economics, and should include such factors as first cost, installation costs, obsolescence, depreciation, maintenance costs, return on investment, flexibility, time of installation and possible loss of useful space during installation.

This same analysis should be extended to include a comparison of remote units with self-contained units. Self-contained units are possible where short term leases are involved, or where wiring can readily be brought to location or where existing water and drain facilities are adequate to handle the increased demand of water-cooled models. On

the other hand, remote type room units contain less mechanical machinery, avoid the operating cost penalty of an air-cooled condenser, as in the case of the self-contained, air-cooled room units, and seldom require heavy wiring to handle the fan load. In general, the smaller remote units are more suitable for use with new construction, where the units will probably remain during their useful life. This condition may change where larger, ceiling mounted units are compared with the floor mounted self-contained units. Controls are essential in unit application and range from the simple snap switch of a room unit to elaborate means of controlling temperature, humidity and air movement in laboratory and testing room application. The criterion of a well controlled installation is one which has neither too few nor too many controls. (See Chapter 33).

With the expansion of cities during the past decade, the problem of water supply has become a costly and ever present problem[5]. Consequently, laws are appearing designed to restrict the usage of water wherever possible, particularly when substitute means are available. Evaporative condensers and cooling towers (described in Chapters 24 and 26) are means devised to save large quantities of water in this connection. Most manufacturers furnish equipment designed for use in combination with these water savers.

Municipal plumbing, heating, electrical and refrigeration codes, as well as fire underwriter restrictions, are likewise having their effects on the application of unit air conditioners. Meeting municipal and national code requirements should be an important item in connection with the installation of any unit equipment.

Ratings

There are two codes governing the rating and testing of unit air conditioners. The first code, Standard Method of Rating and Testing Air Conditioning Equipment[6], covers all types of air conditioning units except the self-contained type. The latter is covered by the second code, The Standard Method of Rating and Testing Self-Contained Air Conditioning Units for Comfort Cooling[7]. The two codes are necessary because of the basic difference caused by the heat given up by the self-contained units. The standard rating conditions for self-contained unit air conditioners, as given in the code, are set forth in Table 1.

The standard rating of a self-contained unit for the conditions specified in Table 1 include all items which apply to the function of a unit as: (1) name of unit, (2) functions which unit performs, (3) data on cooling, (4) data on heating, (5) data on air flow, and (6) data on humidification.

The standard rating conditions for unit air conditioners, other than the self-contained type, are identical to those in Table 1 except the entering wet-bulb temperature for cooling is expressed as 50 per cent relative humidity (66.7 F wet-bulb) instead of 67 F wet-bulb temperature. In addition, the saturated suction refrigerant temperature for comfort cooling is specified at 40 F. This condition is omitted from Table 1 for self-contained units as immaterial in the rating of a unit that includes the evaporator and condensing unit.

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[5]Concerning Conservation of Underground Water with Suggestions for Control, by Noel E Porter (A.S.H.V.E TRANSACTIONS, Vol. 47, 1941, p. 309).

[6]Loc. Cit. Note 1

[7]Loc. Cit. Note 2.

TABLE 1. STANDARD RATING BASIS FOR SELF-CONTAINED AIR CONDITIONING UNITS

FUNCTIONS	TYPES OF UNITS	RATING CONDITION		
		Item	Description	Value
All	All	a	Barometric Pressure	29.92 in. Hg.
Cooling	Water-Cooled, Air-Cooled and Evaporatively Cooled Condensers	b	Unit Ambient and Air Entering Room—Air Inlet (1) Dry-Bulb (2) Wet-Bulb	 80 F 67 F
		c	Ventilation Air	See Note
	Water-Cooled Condensers	d	Water Temperature Entering Unit	75 F
		e	Water Temperature Leaving Unit	95 F
	Air-Cooled and Evaporatively Cooled Condensers	f	Air Entering Outside Air Inlet (1) Dry-Bulb (2) Wet-Bulb	 95 F 75 F
Heating	All Types Provided with Heating Function	g	Unit Ambient and Total Air Entering Unit	70 F
		h	Heating Medium, Pressure or Temperature (1) Dry Saturated Steam (2) Water In (3) Water Out	 16.7 lb per sq in. abs. 180 F 160 F
Humidifying	All Types Provided with Humidifying Function	i	Unit Ambient	70 F
		j	Total Air Entering Unit (1) Dry-Bulb (2) Wet-Bulb	 70 F 53 F
Air Circulation	All	k	Filters	New and Clean

Note: Rating shall be based on both ventilation and recirculated room air entering at 80 F dry-bulb and 67 F wet-bulb temperature (The *Note* as given in the code has been condensed in order to remove material not pertinent to this chapter).

UNIT AIR COOLERS

This type of unit is primarily intended to perform the main function of cooling air, with humidity control a secondary function within the limitations of the design. The main application of this equipment is in process and product refrigeration such as cold storage warehousing, fruit and vegetable packing, in breweries, and in wholesale and retail food markets; although some comfort cooling may be obtained by the use of a unit similar in design to this type of unit as previously explained and as illustrated by Fig. 6.

Application of the unit method of air cooling with mechanical circulation is comparatively recent, being an improvement over the pipe or finned coil, which depended on gravity for circulation. Bunkers were sometimes constructed around the coils to direct the air flow and sometimes fans were used for forcing air over the coils. The location of the unit air cooler is usually within the refrigerated area, but the larger, blower type models may be remotely located.

Design and Performance

Greater application and use of commercial refrigeration have resulted from the development of the unit air cooler. Flexibility of design has permitted almost any condition to be met. By varying such physical features as the method of introducing the refrigerant into the coils, the depth of coil rows and area of their surfaces, and the air volume over the coils, the designer is able to produce a wide range of performances and to offer many desirable features not obtainable with the coil and bunker method. More uniform temperatures, high relative humidity, moderate first cost and a minimum of installation expense are likewise factors in their development.

New uses have appeared for unit air cooler application in industrial and commercial processes involving both the raw materials and finished product, where the maintenance of low temperatures is a necessary part of these processes. Of particular interest is the new field of extreme low

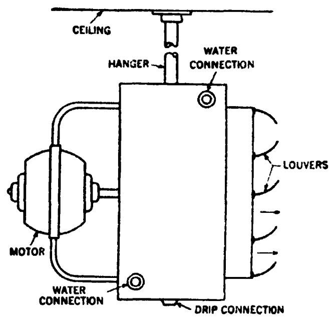

FIG. 8. CEILING TYPE UNIT AIR COOLER

temperature application where many new uses for refrigeration are being found.

Types of Units

The two standard types are the suspended or ceiling type, and the vertical or floor mounted type. There are variations of this such as the panel type which is wall mounted and arranged to take in air from the lower section and discharge it from the upper section.

The ceiling type has the appearance of a unit heater, with its propeller type fan blowing air through a bank of coils as shown in Fig. 8. Singly or in combination, they are easily installed and occupy little or no useful space. Alterations may be accomplished with little cost by relocating units or adding additional ones for increased capacity.

The floor mounted types employ blower type fans, as their air deliveries are higher and their locations may be remote from the space to be refrigerated. This type of unit is illustrated by Fig. 9. Air velocities and volumes must be designed for the individual application. This type of unit may employ a pump to spray a eutectic solution over the coils for the purpose of avoiding frosting as shown in Fig. 10.

Ratings

In order to rate and test equipment of this kind which normally operates below the frost line, a proposed code, Standard Methods of Rating and

Testing Forced-Circulation Air Coolers for Commercial and Industrial Refrigeration[8], has been issued. This proposed standard covers only the modifications of the Standard Method of Rating and Testing Air Conditioning Equipment[9], as it is related to the different applications of unit air coolers. In this standard, the gross cooling effects are taken since the motor power input equivalent is to be computed as part of the load.

From this code Table 2 is abstracted to show standard rating conditions for forced-circulation air coolers. Other modifications take into consideration the effect of frost formation on the coils and the change in length of test runs required to meet such conditions.

Defrosting

Unit air coolers are often required to operate in rooms where air and

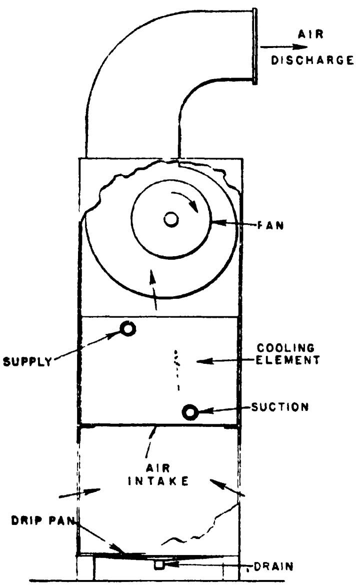

FIG. 9. SURFACE TYPE COOLING UNIT

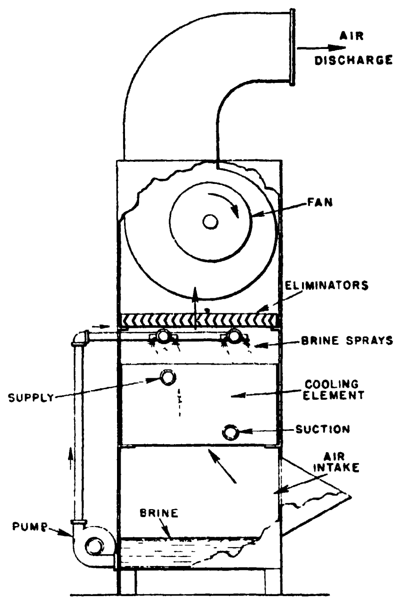

FIG. 10. BRINE SPRAY TYPE COOLING UNIT

refrigerant temperatures are below the freezing point. This results in the freezing of the condensation on the coils and this accumulation of frost builds up to such an extent that there is a loss of capacity. This deposit is removed from the coils by the process known as defrosting which may be accomplished by several methods:

1. Where the room temperature is above the freezing point, the flow of refrigeration to the coils is halted and the fan continued in operation until the coils are defrosted.

2. The hot gas defrosting method is accomplished by a valving arrangement whereby hot compressed gases from the compressor are pumped directly into the evaporator. This operation is continued until defrosting is complete, when the system is returned to normal operation.

3. Where brine is used as a refrigerant, hot brine may be circulated through the coils.

[8]Loc. Cit. Note 3

[9]Loc. Cit. Note 1.

TABLE 2. STANDARD RATING CONDITIONS FOR AIR COOLERS

GROUP NO.	ENTERING DRY-BULB TEMPERATURE, DEG F	ENTERING RELATIVE HUMIDITY, PER CENT	EVAPORATING TEMPERATURE, DEG F
I	45	85	30
II	35	85	25
III	35	85	15
IV	0	85	−10
V	0	85	−20

Where the frosting is particularly heavy it is sometimes more advisable to apply the source of heat externally. This principle of defrosting is accomplished by:

4. Defrosting by warm air is accomplished by dampering arrangements which permit the cooler fan to draw warm air from a source outside of the refrigerated space, pass it over the coils and discharge it outside of the refrigerated space.

5. Constant wetting of the coils with a brine or eutectic solution prevents the formation of frost. This is accomplished on a vertical unit air cooler as shown in Fig. 10.

6. Electric heating elements, placed in such a manner that the fan forces the heated air over the coils.

7. A simple means of defrosting is by means of water sprays placed so the coil is thoroughly wetted during the defrosting process. City water at ordinary pressures is used. The fan is not running during this operation.

Those systems using method 5 have the problem of removing condensation from the eutectic solution. One way is to waste the entire charge when dilution has rendered it ineffective. Another method employs a device which boils off the water and returns the eutectic solution to the system for further use.

ECONOMICS

In the planning and designing of a unit system or in comparing units with central plant application, a systematic approach to the problem should be made from the economic viewpoint.

First Costs. The question of first cost is but one factor in the economic approach of a skillful designer or an intelligent buyer.

1. *Equipment.* The use of a large percentage of factory fabricated equipment to maintain installation costs at a minimum, as represented by self-contained units, should be contrasted with the use of systems with a large percentage of installation labor and material, such as a central station system. The remote unit offers a compromise between these two.

2. *Installation costs.* The influence of existing codes and ordinances, and installation conditions, involving new construction or treating an existing structure deserve careful consideration. Methods which are suitable for new construction often may not be applied on buildings erected in the past. Sprinkler system rearrangements, fire doors and dampers, restrictions on multiple direct expansion units, rewiring or increasing service, cutting or reinforcing of ceilings and roofs are some items which should receive attention in this respect.

Operation and Maintenance. Tenants or occupants usually operate the self-contained equipment, which also lends itself readily to contract service for maintenance. Central plants require operating engineers in attendance who also frequently service the equipment. The remote units require a combination of these two methods of operation with maintenance by building personnel.

Water and power costs per season can be tabulated and compared for

the various systems and an estimate of the costs of replacement material such as filters, belts, oil, refrigerant and wearing parts should be included on an annual basis.

Such questions as obsolescence, depreciation and return on investment are subjects for special study and investigation. Due consideration should be given to the value of resale of equipment and its portability in the event of removal to new locations.

ATTIC FANS

Attic fans are used during the warm months of the year to draw large volumes of outside air through a house and offer a means of using the comparative coolness of outside evening and night air to lower the inside temperature.

Because the low static pressures involved are usually less than ⅛ in. of water, disc or propeller fans are generally used instead of the blower types. The fans should have quiet operating characteristics, and they should be capable of giving about 20 to 30 air changes per hour in northern areas. In the South the usual specification requires one air change per minute which provides appreciable air movement in addition to lowering the inside air temperatures[10].

Types

Open attic fans are units in which the fan is installed in a gable or dormer of the attic and one or more grilles are provided in the floor of the attic, permitting air to flow from the hall below. Outdoor air, which enters the house through open windows, is drawn into the attic through the grilles, and is discharged outside by the fan. An attic stairway may be used in place of the grilles. It is essential that the roof and the attic walls be free from air leaks.

Boxed-in fans are units in which the fan is installed within the attic in a box or housing directly over a central ceiling grille, or in a bulkhead enclosing an attic stair. The fan may be connected by a duct system to the grilles in individual rooms. Outdoor air entering through the windows of the rooms below is discharged into the attic space and escapes to the outside through louvers, dormer windows, or screened openings under the eaves.

Another version of the attic fan is the *window fan* for use when attic application is not feasible or no attic is available. Supplied with a perforated or expanded metal enclosure and mounted in either the upper or lower window section, this fan is easy to install or move to another location.

The locations of the fan, the outlet openings, and grilles should be selected after consideration of the room and attic arrangements in order to give uniform air distribution in the individual rooms served. If the outlet for the air is not on the side away from the direction of the prevailing wind as in the case of the boxed-in fan, openings should be provided on all sides. Kitchens should be separately ventilated because of the fire hazard, and to prevent the spread of cooking odors.

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[10]Comfort Cooling with Attic Ventilating Fans, by G B. Helmrich and G. H. Tuttle (A.S.H V E. TRANSACTIONS, Vol 40, 1934, p. 155). A S H V E. RESEARCH REPORT No 979—Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P Kratz and S. Konzo (A S.H V E. TRANSACTIONS, Vol. 40, 1934, p 167). A S H V.E RESEARCH REPORT No. 1198—The Effect of Attic Fan Operation on the Cooling of a Structure, by W. A. Hinton and A F Poor (A S H V.E. TRANSACTIONS, Vol 48, 1942, p 145). The Installation and Use of Attic Fans, by W. H. Badgett (Agricultural and Mechanical College of Texas, *Bulletin*, No. 52, 1940).

The window fan may be located in a hall or an unused bedroom. Noise of operation is more of a problem with the window fan than with the attic type, although care should be taken to locate either type of fan so that occupants are not disturbed.

These fans range in capacity from 3000 to 30,000 cfm. The window type usually does not exceed 8000 cfm, while the most generally used attic type ranges from 8000 to 16,000 cfm. Power consumption is under 50 watts an hour per 1000 cfm of rated output for the 8000 cfm fan and larger while the watts input for smaller fans is greater than this figure. Improved results can be secured with the window fan by closing off parts of the house where ventilation is not desired.

CHAPTER 23

Dehumidification

Definitions and Methods, Dehumidification by Sorption, Estimating Loads, Location and Use of Equipment, Equipment Auxiliaries, Equipment Performance, Comparison of Methods of Dehumidification

THE interchange of sensible or latent heat between air and a controlling medium requires (1) that the controlling medium be held at the necessary temperature or vapor pressure to produce a flow of heat or moisture in the desired direction, and (2) that sufficient contact be obtained between the air and the controlling medium to produce the desired final air condition under the temperature or vapor pressure head causing the heat flow. The medium may be brought into direct contact with the air by means of a water spray or steam humidifier or the heat transfer may be effected through a barrier surface such as that of a steam radiator or direct expansion cooling coil.

Heat interchange is obtained when molecules of air are brought into close proximity with the medium or heat transfer surface. These molecules then re-mix and transfer heat to other molecules in the air stream. For a given heat head the effectiveness of interchange is a direct function of the number of such successive contacts, and is a measure of the effiency of the surface[1]. The contacting surface may be that of the medium such as a finely atomized spray or the bed of a solid dehumidifying agent; or a chilled or warmed metal surface, as in a coil; or a combination of medium and surface, as in a packed tower, in which the medium produces the interchange and the surface provides the necessary contact area.

DEFINITIONS AND METHODS

There are several basic methods of producing the necessary difference in temperature or vapor pressure between air and the medium employed to achieve cooling or dehumidification, or both simultaneously:

Cooling of air refers to the reduction in temperature resulting from the removal of sensible heat. It is always a result of contact with a medium held at a temperature lower than that of the air. Cooling may be accompanied by moisture addition (humidification), by moisture extraction (dehumidification), or by no change of moisture content whatever. Moisture change, if present, is considered as a secondary or by-product effect. As previously stated, the medium may be directly in contact with the air (as water, brine, or ice), or indirectly through a barrier wall (as cooling surface). When the latter method is used, and the surface temperature is held above the air dew-point, only cooling occurs without moisture interchange.

Evaporative Cooling involves the adiabatic exchange of energy between air and a water spray or wetted surface. The water assumes the wet-bulb temperature of the air, which remains constant during its traverse of the exchanger. No heat is added or abstracted from the medium (water), which is continually recirculated. Cooling of the air occurs due to the temperature and vapor pressure difference between entering air and water at the wet-bulb temperature. Humidification occurs as a result of the vapor pressure exerted by the water which is higher than that corresponding to the entering air dew-point. Since this is an adiabatic exchange, the enthalpy of the air vapor mixture remains constant, while the dew-point rises and the dry-bulb falls, and the loss of sensible heat exactly equals the gain in latent heat (neglecting radiation losses). The maximum available temperature reduction is the difference between entering dry- and wet-bulb temperatures. Equipment delivering air at a dry-bulb temperature equal to the wet-bulb temperature is termed *completely* saturating or 100 per cent efficient, since the air

[1]The Contact Mixture Analogy Applied to Heat Transfer with Mixtures of Air and Water Vapor, by W. H. Carrier (*A.S.M.E. Transactions*, Vol 59, 1937).

leaves in a saturated state. Equipment delivering air at a dry-bulb temperature above the wet-bulb temperature is termed *partially* saturating.

Evaporative cooling is being used advantageously in many parts of the country. It is particularly applicable (1) in districts where the normal maximum wet-bulb temperature remains sufficiently low during the cooling season, and (2) in applications where the cooling load is principally sensible heat.

Dehumidification of air means to decrease the density of water vapor in the space. This may be accomplished by the condensation of water vapor from air due to its contact with a chilled medium (see Cooling). This type of energy exchange invariably includes temperature reduction due to removal of sensible heat. When moisture removal is the prime consideration it is frequently necessary to cool air to a greater extent than can be balanced by normal sensible heat gain in order to remove the required amount of water vapor by condensation. In this case it is necessary to restore sensible heat to the dehumidified air stream by re-heating.

Dehumidification may also be accomplished by contact with a dehumidifying agent. The primary distinction between these two methods is in the manner in which the required vapor pressure at the surface of the contacting medium is obtained. In the case of dehumidification by condensation this surface vapor pressure is always the same as that which would be exerted by a body of water (or ice) at that same surface temperature. In the case of a dehumidifying agent, the surface vapor pressure is always *lower* than that exerted by water at the same temperature, and the effectiveness of the medium as a dessicant is largely a function of the amount by which this vapor pressure can be lowered at the working temperature involved.

Thus it is evident that the primary function of a dehumidifying agent is to maintain a vapor pressure lower than that of the water vapor in the air in order to secure a removal of moisture from the air. In its simplest form the process is essentially an adiabatic one in which the latent heat lost by the mixture is converted to sensible heat which raises the temperature of the air and medium by an equivalent amount. This process is therefore an energy exchange, similar to, but the reverse of, adiabatic saturation or evaporative cooling.

Combination Methods. It is evident that two or more of the above processes—cooling, evaporative cooling, and dehumidification—may be combined by the proper application of interchangers in sequence. Such combinations are dictated by the availability of prime sources and cost of energy as well as the requirements of the particular application in question.

This chapter discusses in detail the engineering and economic principles involved in dehumidification by the use of sorbents. For similar discussion of the other processes, refer to the following material: Cooling and dehumidification by the use of surface interchangers (cooling coils), see Chapter 25. Cooling, dehumidification and evaporative cooling with air washers, see Chapter 26. For sources of cooling involving city and well water and cooling towers, see Chapter 26, while for mechanical refrigeration and ice, refer to Chapter 24. For the thermodynamics of evaporative cooling, see Chapter 1.

SORBENTS

Sorbents may be divided into two general classifications:

1. *Adsorbent*—A material which has the ability to hold water or other vapors on its internal surfaces without itself being permanently changed physically or chemically. Certain solid materials, such as silica gel, and activated alumina have this property. Adsorbents also show an affinity for vapors other than water and this action is frequently selective. Thus, in a mixture of water vapor and organic gases, silica gel will effectively remove the water vapor and charcoal will remove the organic gases. The selective property of adsorbents is used in the removal of some contaminating gases. (See Chapter 28.)

2. *Absorbent*—A material which has the ability to take up water vapor but which changes physically, chemically, or both, during the cycle. Calcium chloride is an example of a solid material while liquid materials include solutions of lithium chloride, calcium chloride, lithium bromide and the ethylene glycols.

Adsorbents

These substances contain a vast number of sub-microscopic pores which afford a tremendous internal surface to which water adheres or is

adsorbed. In spite of a porous structure these substances retain sufficient mechanical strength to resist the wear and handling to which they are subjected. To be suitable for dehumidification purposes such substances must fulfill the following requirements:

1. Possess suitable water vapor pressure characteristics.
2. Be available at a reasonable cost.
3. Adsorb sufficient moisture per pound of material to avoid excessive bed dimensions.
4. Be chemically stable, resisting contamination from impurities.
5. Be physically rugged to resist breakdown from handling, abrasion, etc.
6. Be able to withstand breakdown from indefinitely repeated reactivation cycles.
7. Be capable of reactivation at reasonable temperatures.

Activated Aluminum Oxide (Alumina) is a porous, amorphous form of aluminum oxide and is called by the common name Activated Alumina. Commercial activated alumina commonly used for dehumidification contains about 92 per cent of Al_2O_3 combined with hydrated aluminum oxide. It is available in granules ranging from a fine powder to pieces approximately 1.5 in. in diameter. It has high adsorptive capacity per unit of weight and is non-toxic. It may be repeatedly re-activated after becoming saturated with adsorbed moisture without practical loss of its adsorptive ability. In the grade frequently used for air drying the re-activation may be accomplished at temperatures between 350 and 600 F. Specific gravity is 3.25 and the pores are reported to occupy 51 per cent of the volume of each particle. For most estimating purposes the volume-weight relation on a dry basis may be taken as 50 lb per cubic foot although in the smaller sizes the packed weight may be as much as 64 lb per cubic foot.

Silicon Dioxide (Silica), in a prepared form obtained by suitable mixing of sulphuric acid with sodium silicate, is another solid adsorbent and is commonly called silica gel. Its capillary structure is exceedingly small, so small that its exact structure has to be deduced as it cannot be observed. It has high adsorptive capacity per unit of weight; it is non-toxic, and may be repeatedly re-activated at temperatures up to 600 F without practical deterioration. Re-activation is generally accomplished with air or other gases at temperatures not over 350 F. Volume of the capillary pores is reported to be from 50 to 70 per cent of the total solid volume. Silica gel is available commercially in a wide variety of sizes of graded granules ranging from 3 to 8 mesh granules to impalpable powder (through 325 mesh). The 6 to 16 mesh granules are generally used in dehumidification applications and will vary in density from 40 to 45 lb per cubic foot.

There are other solid substances having marked adsorbent properties, but details concerning them are not available. Most adsorbents have the ability to absorb some gases and condensible vapors other than water vapor, a property sometimes useful in air conditioning applications, and one which accounts for the extensive use of adsorbent materials in the field of chemistry.

Air to be dehumidified is drawn or blown through a screened bed of dry, solid adsorbent and the water vapor in the air-vapor mixture is caught and retained in the pores of the medium. The exact nature of the process which goes on during adsorption is not known but it is stated that the action is brought about by surface condensation, and also by a difference between the vapor pressure of the water condensing inside the pores and the partial pressure of the water vapor in the air-vapor mixture. The adsorbing process in the bed can continue until the vapor pressures reach equilibrium. The amount of vapor adsorbed will depend on the

characteristic of the adsorbent being used, the temperature of the air and the bed, the vapor pressure of the water vapor in the air being passed through it, and its final vapor pressure or moisture content.

As the bed of material adsorbs moisture, its vapor pressure approaches that of the contacting air and the efficiency of adsorption gradually decreases. Equilibrium throughout the whole bed will not be reached for an extended period depending on bed thickness. Because of this diminishing efficiency of adsorption, commercially designed systems do not approach the state of equilibrium, but generally operate on a predetermined cycle or contact time and at partial saturation.

As the process of adsorption goes on heat is liberated in the bed. The total heat so liberated consists of the latent heat of the water vapor condensed together with the so-called heat of wetting of the adsorbent. For silica gel, the total heat of adsorption will amount to approximately 1293 Btu per pound of water vapor adsorbed from the air-vapor mixture being passed through the bed. The heat of wetting varies with the substance used as adsorbent while the latent heat of condensation depends only on the temperature and pressure of the water vapor.

Since the adsorptive ability of an adsorbent depends on the temperature of the bed and on the vapor pressure difference between the pores and the air-vapor mixture, it is important to know the pressures and temperatures at which pressure equilibrium is reached.

Evidently the equilibrium conditions represent the limits beyond which adsorption of vapor cannot continue. This relationship for one commercial silica gel of 0.70 specific gravity is shown graphically in Fig. 1. These curves indicate the general manner in which adsorbents may be expected to perform, though the exact form and values of the curves will vary for each specific adsorbent and may vary even within the same generic group.

As an example in the interpretation of the chart consider the case when moist air at a temperature of 80 F and a partial vapor pressure of 0.5 in. of mercury flows through a bed of silica gel which is at a temperature of 80 F. The chart indicates that the equilibrium of pressure between the air-vapor mixture and the bed is reached when the dry bed has adsorbed moisture to the extent of 30 per cent of the weight when dry. When this happens the bed can adsorb no more moisture unless its temperature is decreased or the air vapor pressure increased.

The effect of temperature upon the adsorptive capacity of silica gel may be observed by following the 59 F dew-point line in Fig. 1. At 80 F gel temperature equilibrium is reached at 30 per cent moisture concentration while at 120 F and 200 F equilibrium is reached at 12 and 6.6 per cent moisture respectively.

Under normal reactivation temperatures the residual water content of silica gel is between 5 and 6 per cent.

In practice, the temperature rise in the dehumidified air caused by the adsorption heat is approximately 10 F for each grain of moisture removed per cubic foot of air at atmospheric pressure. This temperature rise occurs progressively through the adsorbent bed and is an important consideration in predetermining the performance of a given design of apparatus. Data such as these together with information covering other characteristics, such as specific heat, resistance to air flow, etc., are of value in the basic design of adsorption apparatus. In the solution of air conditioning problems, however, reference must be made to per-

formance data on established apparatus designs such as are presented later in this chapter.

Absorbents

Any absorbent substance may be used as a dehumidifying agent if it has a vapor pressure lower than the vapor pressure in the air-vapor mixture from which the moisture is to be removed.

Solid Absorbents. The substances used are in general the solid forms of the liquid absorbents Calcium chloride is frequently used because of low cost. At present they are used principally in small dessicating chambers, and in small dryers of the cartridge type, through which air is forced under pressure.

Liquid Absorbents. These are characteristically water solutions of materials in which

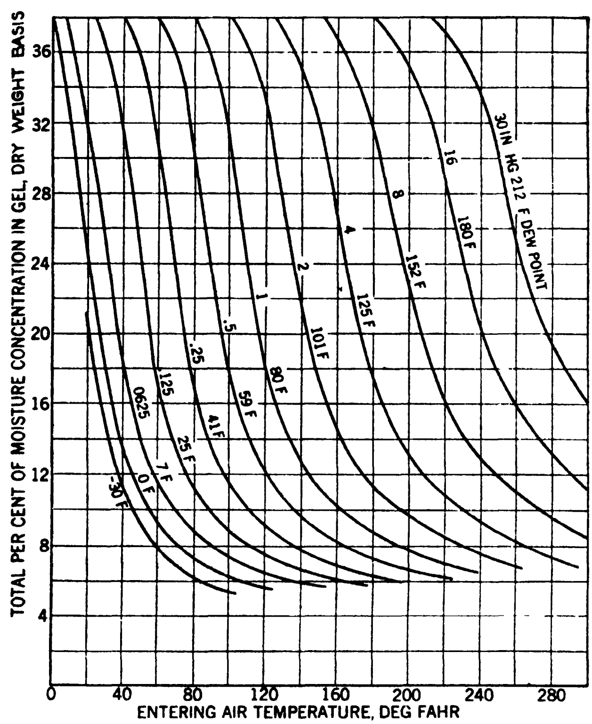

FIG. 1. TEMPERATURE—VAPOR PRESSURE—CONCENTRATION RELATION FOR A SILICA GEL BED AT CONSTANT TEMPERATURE

the vapor pressure is reduced to a suitable level by governing the concentration and temperature of the dehydrating solution. In addition to having suitable vapor pressure characteristics a practical absorbent must also be widely available at economical cost, be non-corrosive, odorless, non-toxic, non-inflammable, chemically inert against any impurities in the air stream, stable over the range of use and especially it must not precipitate out at the lowest temperature to which the apparatus is exposed. It must have low viscosity and be capable of being economically regenerated or concentrated after having been diluted by absorbing moisture.

Water solutions, or brines, of the chlorides or bromides of various inorganic elements such as lithium chloride and calcium chloride are the absorbents most frequently used in connection with air conditioning applications and detailed attention is confined to these two in this chapter.

The application consists of bringing the air-vapor stream into intimate

contact with the absorbent, permissibly by passing the air stream through a finely divided spray of the brine but more generally by passing the air over a contacting pack where the liquid absorbent presents a large surface to the air stream. The difference in vapor pressure causes some of the vapor in the air-vapor mixture to migrate into the brine. Here it condenses into liquid water and decreases the concentration of the absorbent.

As the water vapor is added to the absorbent and condenses, it gives up its latent heat of condensation. For every pound of water absorbed and condensed the heat released is obtainable from steam tables. For in-

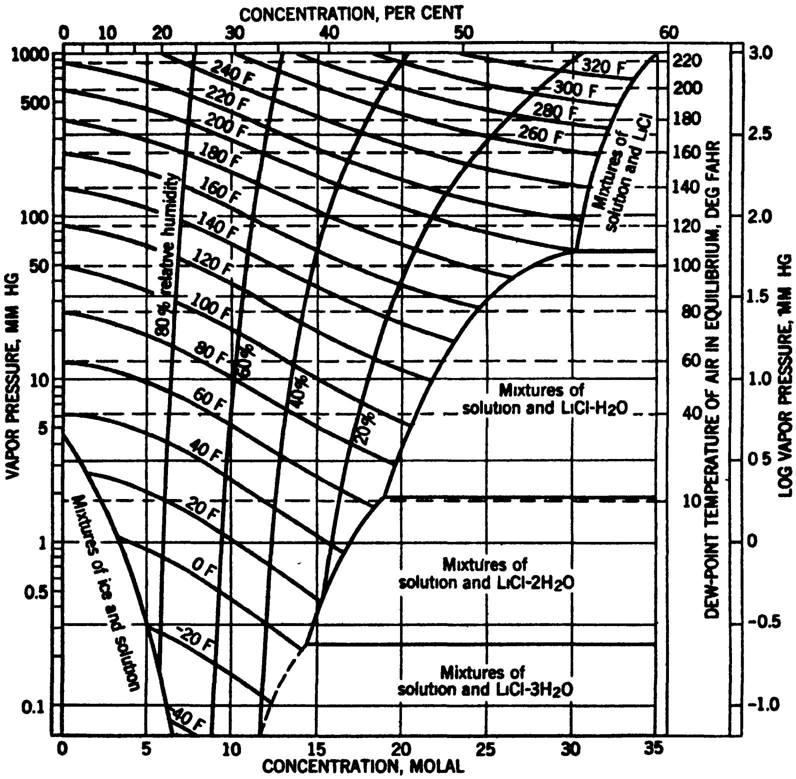

FIG. 2. TEMPERATURE—PRESSURE—CONCENTRATIONS FOR LITHIUM CHLORIDE

stance, at a dew-point temperature of 60 F the amount of this heat is about 1057 Btu. In addition to this heat there is involved also the so-called heat of mixing which is frequently considerable. The heat thus produced in the contacting packs of the average air conditioning installation is to a large extent transferred to the chemical solution itself and the balance to the air being dried.

A more complete cycle involves heat removal from the contacting medium, either within or external to the interchanger. Thus the temperature of the medium may be higher than, equal to, or lower than that of the air, depending on the agent used and the function to be performed. In such a cycle, the process may be accompanied by cooling or heating, or either, and such effect, if present, may be either a necessary by-product of the process, or for the specific purpose of obtaining both latent and

sensible heat removal simultaneously. In many industrial applications it is not necessary to aftercool the air. In applications requiring low dry-bulb temperatures excess sensible heat must be removed by an aftercooler.

Since the absorption process can continue only as long as there is a difference in vapor pressure between the absorbent and the air-vapor mixture and since at a given temperature of the absorbent the vapor pressure depends on the concentration of the solution, evidently there must be a relation between these quantities which if known would state

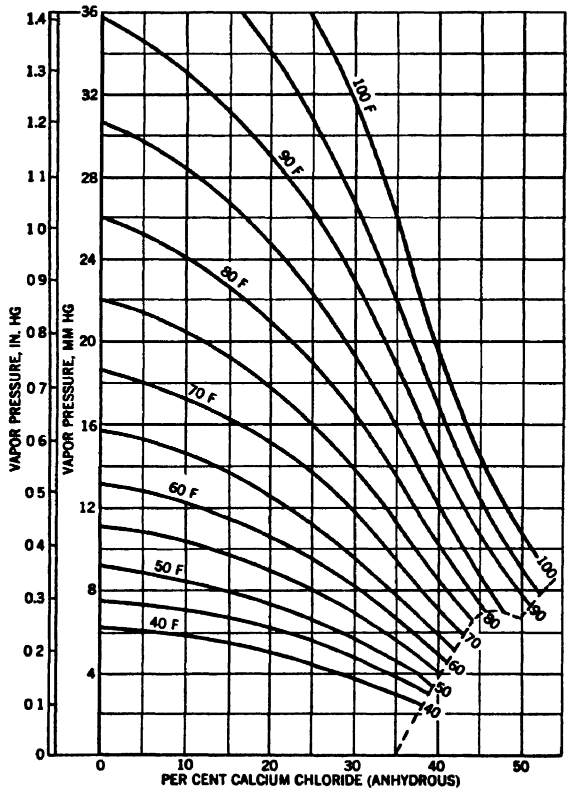

FIG. 3. TEMPERATURE—PRESSURE—CONCENTRATIONS FOR CALCIUM CHLORIDE

the limits of the process. The relationship would also depend on the absorbent being used, and would have to be determined for each substance used as an absorbent. This relationship is shown graphically in Fig. 2 for lithium chloride, and Fig. 3 presents similar data for calcium chloride. These charts are essentially similar to Fig. 1, and their direct usefulness is limited by much the same considerations. Other physical properties of lithium chloride are shown in Tables 1, 2 and 3.

In Fig. 2 and Table 1 the unit of concentration is the *mol*. An M molal solution is defined as a solution containing $M \times 42.37$ grains of anhydrous lithium chloride per 1000 grains of water. The formula connecting concentration in mols with weight in per cent is equivalent to: $(100 \times M \times 42.37) \div [1000 + (M \times 42.37)]$.

Table 1. Properties of Lithium Chloride Solutions

Concentration Pound Mols (42.4 Lb) LiCl per 1000 Lb Water	Concentration Per Cent by Weight	Specific Gravity at 100 F	Viscosity (Millipoise)		Partial Heat of Mixing at 0 F, Btu per Lb	Temp. Coeff. of Partial Heat of Mixing Btu per Lb per F	Specific Heat at 70 F	Boiling Point F at (760 mm Hg)	Freezing Point, F
			80 F	180 F					
0	0.0	1.000	8.61	3.48	0.00	0.000	0 998	212 0	32.0
2	7.8	1.037	11 19	4.56	2.04	−0.014	0.901	215.8	16.3
4	14.5	1.076	14.42	6.01	7.24	−0.036	0.831	221.5	− 5.8
6	20.2	1.111	18.62	7.78	16.70	−0.069	0.778	228.9	−34.2
8	25.3	1.143	24.32	10.00	31.90	−0.109	0.739	238.1	−69.0
10	29.7	1.172	32.28	12.91	51.10	−0 143	0 710	248.4	−90 0
12	33 7	1.199	43.45	16.56	75.70	−0 160	0.687	258.8	−40.0
14	37.4	1.225	60.26	21.28	90.80	−0.167	0.666	268.9	1.0
16	40.4	1.248	82.04	27.10	124.80	−0.176	0.647	277.9	36.5
18	43.3	1.270	113.80	35.48	145.00	−0 186	0.631	285.8	58.1
20	46.0	1.291		46.45	162.00	−0 194		293 2	86 4
22	48.4			60.67	171.00	−0 200		300.2	133.0
24	50.3			84.33	177.00	−0.200		307.0	156.0
26	52.4				182.00	−0 210		313.0	180.0
28	54.3				191.00	−0 210		318 0	190.0
30	56.1				194.00	−0 210		323.0	195.0
32	57.5				198 00	−0 220		328 0	280.0

Table 2. Dew-Point of Air in Equilibrium with Lithium Chloride Solutions Concentration in Pound Mols (42.4 lb) Lithium Chloride per 1000 lb Water

Dew-Point at Zero Conc.	Concentration of Lithium Chloride														
	2.0	4.0	6.0	8.0	10.0	12.0	14.0	16.0	18.0	20.0	22.0	24.0	26.0	28.0	30.0
320	315 2	308.7	299.9	290.2	279.7	269.4	259.6	251.5	244.1	236.5	230.0	223.8	218.6	214 5	210.3
300	295 4	289.1	280.5	270.9	260.6	250.5	240.8	232.6	225.4	218 0	211.8	205.8	200.8	196.9	192.8
280	275.6	269.5	261.1	251.7	241.5	231.6	222.2	214.0	206.7	199.7	193.5	187.8	183.2	179.3	175.2
260	255.8	250.0	241.9	232.6	222.7	212.8	203.5	195.5	188.4	181.7	175.4	170.0	165.6	162.0	158.4
240	236.0	230.4	222.5	213.5	203.8	194.2	185.0	177.1	170.0	163.6	157.5	152.2	148.3	144.6	140.5
220	216.2	210.8	203.2	194.4	184.9	175.5	166.4	158.6	151.6	145.3	139 6	134.6	130.7	127.3	124.2
200	196.4	191.2	183.9	175.4	166.1	156.7	148.0	140.3	133.5	127.3	121.9	117.0	113.3	110.1	
180	176.6	171.6	164.7	156.4	147.3	138.1	129.6	122.1	115.5	109.4	104.2	99.6	96.0		
160	156.8	152.1	145.4	137.4	128.6	119.7	111.3	103.9	97.4	91.6	86.6	82.2			
140	137.0	132.6	126 1	118.4	109.9	101.3	93.1	85.9	79.5	73 8	69 0				
120	117.2	113.0	106.8	99.4	91.1	82.7	74.7	67.8	61.5	56.0					
110	107.3	103.2	97 2	89.9	81.9	73.5	65.6	58.8	52 6	47.1					
100	97.4	93.4	87 5	80.5	72.7	64.4	56.6	49.8	43.7	38.2					
90	87.5	83 6	77.9	71.0	63.3	55.2	47.6	40.8	34.8	29.3					
80	77.6	73 8	68.4	61.6	54.0	46.1	38.5	31.8	25.9	20 6					
70	67.7	64.0	58.7	52.2	44.8	37.0	29.5	22.9	17.2	12.0					
60	57.8	54.3	49.1	42.7	35.5	27.9	20.5	14.0	8.3						
40	38.0	34.7	29.9	23.9	16.9	9.6	2.4	−3.9							
20		15.1	10.7	5.0	−1.7	−8.7	−15.4								
0		−4.5	−8.6	−13.9	−20.2	−27.0	−33.3								

TABLE 3. DENSITY OF LITHIUM CHLORIDE SOLUTIONS

Concentration Pound Mols (42 4 lb) *LiCl* per 1000 lb Water	Temperature Deg F						
	0	50	100	150	200	250	300
0							
2		1.045	1.037	1 026	1 012		
4	1.090	1.085	1.076	1.064	1.052		
6	1.124	1.119	1.111	1.100	1.087		
8	1.156	1 150	1.143	1.132	1.122		
10	1.188	1.181	1.172	1 162	1.152	1.142	
12	1.217	1.209	1.199	1.188	1.178	1.168	
14	1 242	1 235	1.225	1.214	1.203	1.192	
16		1 257	1.248	1 236	1.226	1.215	
18		1 279	1.270	1.259	1.248	1.237	
20			1 291	1.280	1 279	1.568	
22				1.310	1.289	1.278	1 267
24				1 317	1.307	1.296	1.286
26					1 313	1.312	1.302
28					1 338	1.327	1.318
30						1 34	1.33
32							1.35

Example 1. Determine the dew-point, wet-bulb, relative humidity and absolute humidity of air in equilibrium at 100 F with pure lithium chloride solution of density 1.270.

Solution. From Table 1 the concentration of a solution of density 1.270 at 100 F is 18.0 *M* From Fig. 2 the dew-point of 18 *M* lithium chloride at 100 F is 43.7 F. From Table 6, Chapter 1, the partial pressure of water over the solution is 0.2858 in. of Hg, and the absolute humidity is 42.00 grains per pound dry air. From the psychrometric chart, the wet-bulb is 66.3 F, and the relative humidity is 15 per cent.

Example 2. Determine the boiling point, and freezing point of 18 *M* lithium chloride solutions.

Solution From Table 1, boiling point (standard) is 285.8 F, freezing point is 58.1 F.

Example 3. Calculate the heat of vaporization of 1 lb of water from a large amount of 18 *M* lithium chloride solution at the boiling point.

Solution. The heat of boiling is equal to the heat of mixing plus the heat of boiling pure water at the same temperature. The heat of mixing from Table 1 at 18 *M* and 285.8 F is $145 - (0.186 \times 285.8) = 92$ Btu per pound. The heat of vaporization of water from steam tables at 285.8 F is 920 Btu per pound. Therefore the heat of vaporization of water from the solution is $920 + 92 = 1012$ Btu per pound.

Example 4. One thousand pounds of air per minute at 100 F dry-bulb with a dew-point of 70 F and a relative humidity of 39 per cent are passed over 18 *M* lithium chloride solution. The rate of flow of the solution is 200 gpm and the entering temperature is 80 F. The air leaves the absorber at 85 F dry-bulb and dew-point of 35 F. Calculate (*a*) the heat to be removed from the lithium chloride solution to maintain these conditions, and (*b*) the temperature rise of the solution in passing through the absorber.

Solution. (*a*) The enthalpy of the entering air at 100 F dry-bulb and 39 per cent relative humidity $= h_a + \mu h_{as} = 24.00 + (0.39 \times 47.40) = 42.49$ Btu per pound (Table 6, Chapter 1).

The relative humidity of the air leaving at 85 F dry-bulb and 35 F dew-point is 16.7 per cent (psychrometric chart). Enthalpy of leaving air $= 20.39 + (0.167 \times 28.85) = 25.21$ Btu per pound (Table 6, Chapter 1).

Heat to be extracted from air $= 1000\ (42.49 - 25.21) = 17{,}280$ Btu per minute. Heat of mixing $= 145 - (0.186 \times 80) = 130$ Btu per pound of moisture removed. From Table 6, Chapter 1, the moisture removal per pound of air $= 0.01574 - 0.00426 = 0.01148$ lb. Heat of mixing for 1000 lb of air $= 1000 \times 0.01148 \times 130 = 1492$ Btu. Total heat extraction $= 17{,}280 + 1492 = 18{,}772$ Btu per minute.

(*b*) The weight of solution circulated is 200×1.27 (Table 1) $\times 8.33 = 2116$ lb per minute. Its heat capacity $= 2116 \times 0.631$ (Table 1) $= 1335$ Btu per minute per degree Fahrenheit. The temperature rise $= 18{,}772 \div 1335 = 14.1$ F.

DEHUMIDIFICATION WITH SOLID SORBENTS

One type of equipment suitable for producing dehumidification with solid drying agents utilizes an apparatus with continuously rotating beds or dampers as illustrated in Fig. 4. The apparatus consists essentially of a cylinder or drum filled with a dehumidifying or drying agent. Air flow through the drum is directed by baffles which permit three independent air streams to flow through the adsorbing material. One air stream consists of the wet air which is to be dehumidified. The second is heated activation air used for drying that part of the dehumidifying material which has become saturated. The third air stream cools the bed to a temperature below 150 F to permit pickup of moisture when that part of the bed returns to the dehydration cycle.

In the rotating bed apparatus, the baffle sheets are stationary and the screened bed rotates at a definite speed to permit the proper time of contact in the activation, cooling and dehumidifying cycles. In the

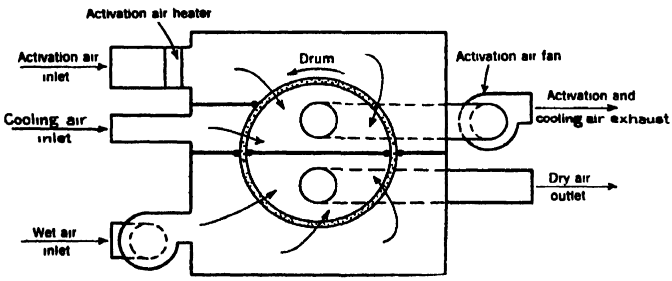

Fig. 4. Solid Adsorbent Dehumidifier—Rotating Bed Type

rotating damper apparatus, the bed remains stationary and a sectionalized damper rotates. This rotating damper produces the same general effect as if the stationary baffles previously mentioned rotated.

Clean air for activation is supplied at temperatures normally ranging from 300 to 350 F. Any source of clean heated air can be used such as air heated by electric heaters or steam coils, or air indirectly heated by coal or oil fired interchangers. Direct fired heaters are usually designed for gas since there must be no condensable, tarry, combustion products to contaminate the adsorbent. To avoid excessive contamination or clogging of the absorbent bed by dust it is frequently desirable to provide filters for the activation air.

Where activation air heating coils are supplied with steam at 80 lb per square inch gage pressure or higher, activation efficiency will remain normal and moisture removal capacity will not be reduced. When lower steam pressures are used, incomplete reactivation will result in reduced moisture removal.

Another type of solid adsorption equipment uses two complete sets of stationary adsorbing beds, arranged so that one set is dehumidifying the air while the other set is being activated. With the dampers in the position shown in Fig. 5, air to be dried flows through one set of beds and is dehumidified, while activation air is heated and circulated through the other set. After activation is complete, the beds are cooled by shutting off the activation air heaters and allowing unheated air to circulate through them.

After the beds have adsorbed moisture to a degree which begins to

impair performance, a timer-controller causes the dampers to rotate to the opposite side. Thus the beds which on the previous cycle were adsorbing have activation air circulated through them, and vice versa. Activation air is heated in the same manner as with continuous equipment.

DEHUMIDIFICATION WITH LIQUID SORBENTS

One type of system utilizing liquid sorbents includes an external interchanger having essential parts consisting of a liquid contactor, a solution concentrator, a solution heater and a solution cooler, all as shown in

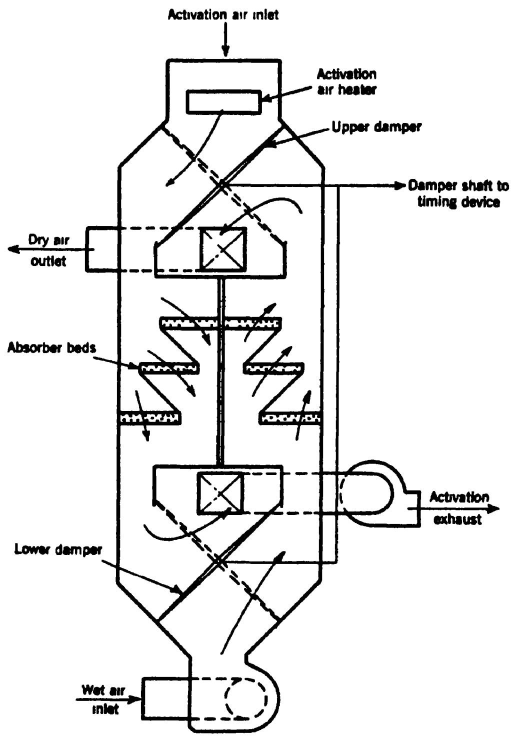

FIG. 5. SOLID ADSORBENT DEHUMIDIFIER—STATIONARY BED TYPE

Fig. 6. The contactor is located in the wet air stream. The air to be conditioned is brought into contact with an aqueous brine solution having a vapor pressure below that of the entering air, resulting in a transfer of moisture (latent heat). As previously described, this results in a conversion of latent heat to sensible heat which raises the solution temperature and consequently the air temperature. The temperature change of the air being processed is determined by the cooling water temperature and the amount of moisture removed in the equipment. Control of leaving air temperature is obtained by precooling the absorbent solution in a suitable surface cooler, by tap, well, or artificially chilled water.

The excess water of condensation, which dilutes the brine, is removed in the solution concentrator. This is a low pressure steam heat exchanger which over-concentrates a portion of the weak liquor and returns it to

the main brine reservoir for re-pumping. The concentrator operates in the manner of an evaporative condenser, whereby moisture is evaporated from the brine, by the heating coils, into a stream of regeneration air taken from and rejected to the outside atmosphere. Low pressure steam is normally used for heating the brine. When it is desirable or necessary to use gas or electricity, an auxiliary low pressure steam boiler is usually added to the equipment. Concentrators operating on a simple boiler principle have not as yet been commercially practical.

It should be noted that the solution concentration phase is the reverse of the absorption process. During concentration the aqueous vapor pressure of the solution is greater than that of the surrounding air, while during dehumidification, the reverse is the case. Utilization of this principle permits winter humidification, by heating (instead of cooling) the solution pumped to the contactor. Water is thereby evaporated into, instead of being condensed out of, the conditioned air stream. This requires dilution of the brine externally to the contactor, rather than concentration.

Another type of liquid dehumidifier utilizes an integral interchanger

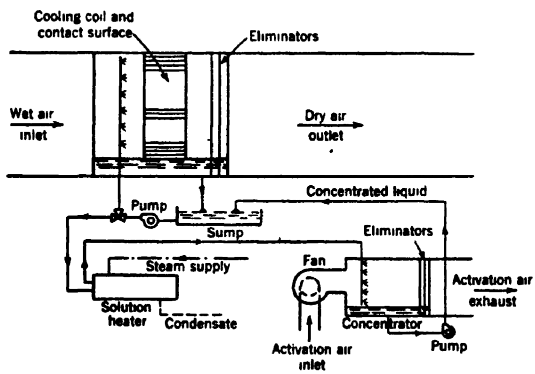

Fig. 6. Liquid Absorbent Equipment in Which Solution Cooler and Contactor are Combined

which employs the same type of solution concentrator as described for the system with the external interchanger. However, the dehumidifying liquid contactor and solution cooler are combined by placing a cooling coil directly in the wet air stream. This coil provides the contacting surface between air and the warm concentrated solution which is sprayed over the cooling surface. By circulating a cooling medium through this coil, control of solution temperature (hence its vapor pressure) is accomplished directly in the air stream.

ESTIMATING LOADS

Where equipment is used which removes sensible and latent heat simultaneously such as a chilled water or direct expansion dehumidifier, the basis of selection is usually the *maximum total heat load*. The operating characteristics of such equipment normally produce satisfactory dew-points with adequate capacity at other loads, including the maximum latent load which occurs at a less-than-maximum total load. With dehumidification equipment in which moisture removal is achieved independently of sensible cooling, it is necessary that equipment be chosen for

the maximum load of each functional element. The sensible cooler should be selected for the maximum sensible load; the dehumidifier for the maximum latent load. These loads need not occur simultaneously, and in fact, rarely do.

In estimating the maximum latent heat load for comfort applications, it is considered good practice to select an outside design dew-point for the locality which is exceeded on not more than 5 per cent of the days during the season. A smaller percentage, or even the maximum dew-point recorded, may be advisable for rigorous industrial applications.

Special consideration should also be given to moisture seepage through building materials and vapor infiltration through openings which appears to be a more serious factor than would be anticipated from consideration of infiltration in the normal manner. These items become important at dew-points below 50 F and have an extreme effect upon load and equipment selection at dew-points below 35 F.

LOCATION AND USE OF EQUIPMENT

Direct or chemical methods of dehumidifying air are frequently employed in conjunction with other equipment installed for the primary purpose of caring for sensible heat removal requirements. They may thus be superimposed upon extensive air conditioning and circulation systems designed to care for independent cooling.

When used in this manner, such equipment may be located to treat outside air only. It is well adapted to this service as it is capable of delivering outside air for ventilation purposes at dry-bulb temperatures close to the normally desired inside temperature, while maintaining a satisfactory humidity condition during those periods when internal sensible heat loads are reduced to a point where no cooling is required.

Because the delivered moisture content of air from all direct dehumidifying devices is materially reduced when the entering air temperature and humidity are lowered, these devices are also frequently applied to the independent dehumidification of mixtures of outside and recirculated air. This practice is advantageous where the moisture removal requirement cannot economically be met by the dehumidification of only the ventilation air component of the total air circulation.

In cases where extremely low dew-point temperatures are required, direct dehumidifying equipment may be located so as to handle air precooled and dehumidified by cooling devices and may also be used in cascade. Solid adsorption equipment is generally favored for such applications because there is no fixed lower limit of operation such as inherently set by the freezing or crystalization temperature of the solution in the absorption system.

In summing up considerations in the location of direct dehumidifying equipment it should be borne in mind that these devices will remove the greatest quantity of moisture from air supplied at the highest working moisture level, but will produce the lowest leaving moisture condition when supplied with air from the lowest working moisture level.

EQUIPMENT AUXILIARIES

Precoolers. When cold water is available, it is generally economical to use this water in a precooling coil in the outside air stream. The dehumidifying accomplished by this coil reduces the load; and moreover,

lowering the temperature of the inlet air results in a higher moisture removal efficiency.

Dry Air Coolers. Particularly with the solid adsorbent process, and to a lesser extent with liquid absorbents, a dry air cooler is employed to remove sensible heat from the dehumidified air whenever it leaves the apparatus at an elevated temperature. A cooling coil using city water is usual practice, and is considered economical whenever the difference between effluent air and entering water temperatures is greater than 15 F.

Sensible Heat Coolers. Since the normal conditioning system requires sensible heat removal, auxiliary equipment may be needed for this function. This is almost always in the form of cooling surface using water, brine or direct expansion refrigerant. It is located on the leaving side of the dehumidifier, but frequently treats in addition a large volume of room air which is not circulated through the machine for moisture reduction.

CONTROLS

The use of direct or chemical dehumidifying equipment makes possible the use of a relatively simple control system with a humidistat or, alternatively, a wet-bulb controller, to regulate the operation of the machine, and a thermostat to control the sensible cooling apparatus Functionally, the relative humidity control may consist of one of the following:

1. *Stop—Start*—Where the humidistat starts the dehumidifier on rising humidity and stops it on falling humidity.

2. *By-pass*—Where the humidistat modulates face and by-pass dampers located at the wet air inlet of the dehumidifier. Thus the quantity of air passing through the dehumidifier is proportioned in accordance with the change in latent heat load.

3. *Vapor Pressure Control* (used with liquid absorbents)—Where the humidistat directly controls the temperature or concentration of the contacting solution, thereby matching the latent heat removal to the load requirement.

EQUIPMENT PERFORMANCE

It is recognized that, whereas the curves relating temperature and vapor pressure of the several dehumidifying agents (Figs. 1, 2 and 3) accurately define the equilibrium limits for these materials, these curves cannot be used for predicting performance of available equipment. This occurs because (*a*) the materials themselves can only be utilized efficiently within certain ranges of moisture concentration, and (*b*) the degree to which the vapor pressure of the air being treated approaches that at the surface of the material depends upon the completeness of the contact. It is obvious, for example, that the thickness of a bed of solid adsorbent for a given air flow will have a profound effect upon the degree of moisture removal attained. However, it will have an inverse effect upon the power required to drive air through the bed. It must also be borne in mind that the moisture removal efficiency of all solid adsorbents and absorbent solutions is related to the temperature of the medium as well as to its state of unsaturation. On the other hand, it will be noted that the lower limits of dehumidification, which can be obtained with absorbent solutions, are likely to be controlled by the equilibrium dew-point at the lowest temperature at which the solution will resist freezing or crystali-

zation. For this reason, actual moisture removing capacity is determined from performance curves of the several materials under practical conditions of temperature, concentration and contacting efficiency as shown in Figs. 7, 8, 9 and 10. While these curves by no means explore all the performance possibilities, they may be considered to be representative of sound design and application practice.

Performance data for standard production designs of silica gel *rotating bed* dehumidifiers are shown in Fig. 7 for a *direct gas fired type*, and in

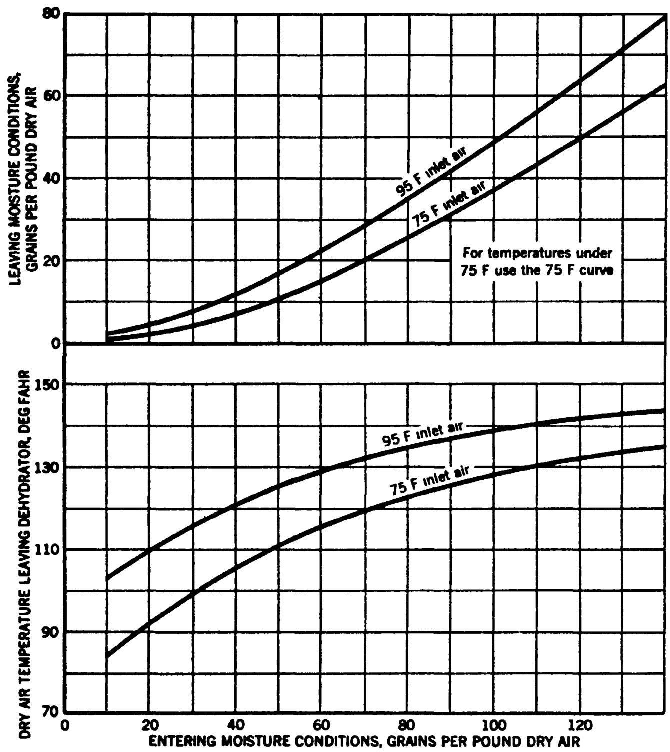

FIG. 7. PERFORMANCE DATA FOR ROTATING BED DIRECT GAS FIRED SILICA GEL DEHUMIDIFIER

Fig. 8 for *steam activation* types for steam pressures of 5 to 80 lb per square inch. They are designed for normal comfort cooling and moderate humidity industrial air conditioning applications. Performance data for a *heavy duty type activated alumina* dehumidifier are given in Fig. 9. This unit has extended surface coils, imbedded in the adsorber, which are supplied with high pressure steam for heating during activation and cool water for direct bed cooling during adsorption. Capacities shown in Fig. 9 are based upon 75 F cooling water. Performance data for *standardized package lithium chloride* equipment are shown in Fig. 10. This is a single unit including both absorption contactor and regeneration section together with necessary heat exchangers.

COMPARISON OF METHODS OF DEHUMIDIFICATION

Almost all summer comfort air conditioning, as well as much industrial and commercial air conditioning, requires both of the functions of sensible and latent heat removal from air. Each of the methods of cooling and dehumidification has as its objective either the removal of sensible or latent heat, or both simultaneously. Choice of method and medium therefore depends solely on whether (*a*) method and medium are physically able to accomplish the desired result with practical equipment, and (*b*) method and medium are justifiable economically.

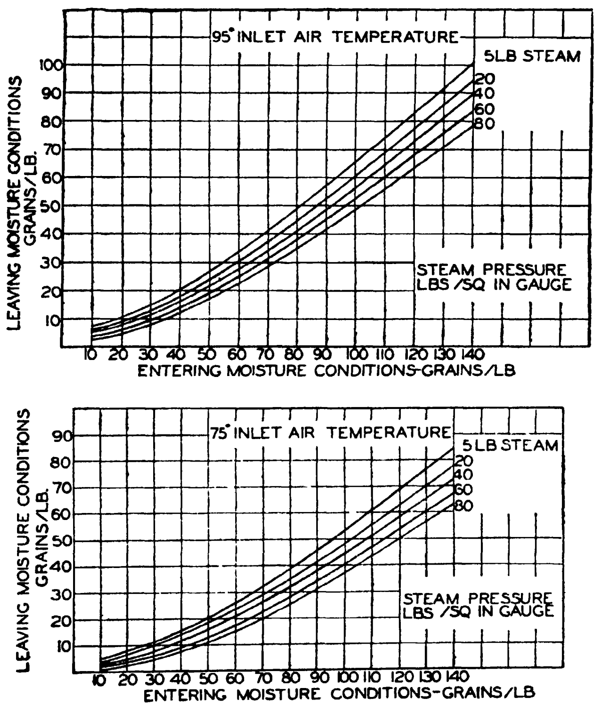

FIG. 8. SILICA GEL DEHUMIDIFIER PERFORMANCE DATA STEAM ACTIVATION TYPE

Referring particularly to the problem of moisture removal, it may be stated that dehumidification either by using chilled water, brine or direct expansion refrigerant, or by using solid or liquid drying agents, is *equally practical* from the viewpoint of engineering performance for the vast majority of comfort and for a great many industrial applications requiring dew-point temperatures above 45 F.

For this reason, in the normal operating range, consideration is usually based upon practical factors which affect the over-all economics of the situation or upon the desire or necessity of attaining independent control of either temperature or humidity. In this connection, it should be remembered that though a satisfactory balance may exist between cooling

and dehumidification requirements at maximum design conditions, there are many periods when practical operating conditions will result in an unsatisfactory balance of sensible and latent loads unless there is a simultaneous use of cooling and reheating.

In the normal range the choice between direct or chemical dehumidification, mechanical refrigeration, natural cold water, or ice, must be justified by the initial investment of available equipment; the availability and cost of prime energy sources; the charges to be allocated to space occupied, labor of operation and maintenance; and the degree of control required.

When required dew-point temperatures are below 45 F, direct or chemical methods deserve special consideration, as they do in applications where the task is primarily one of moisture removal with no necessity for sensible temperature reduction.

It is evident that it is not possible to set forth definite rules governing

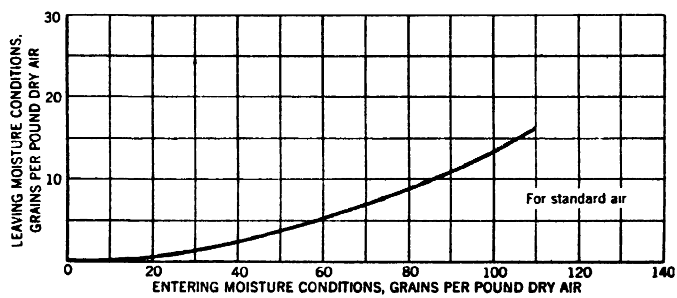

FIG. 9 ACTIVATED ALUMINA DEHUMIDIFIER PERFORMANCE DATA HEAVY DUTY TYPE

the choice of the method of dehumidification It is only possible to state certain general conditions.

Direct or chemical methods tend to be favorable where:

1. Low temperature water is available (below 65 F), to meet sensible cooling requirements.

2 Gas or steam is available at costs competitive with electricity.

3. Abnormally high internal latent heat load is encountered.

4. Abnormally low dew-points are required (40 F or lower).

5. Required dry-bulb temperature is high or control of temperature is unimportant in comparison to maintenance of proper humidity.

6. In low temperature dryers where complementary heat exchange can be utilized. (In such cases the sensible heat of the dry air is reduced by the evaporation of moisture from the product being dried)

Of the factors enumerated, Item 1, coupled with Item 2, tends to have the greatest influence since these factors affect the applicability to normal range requirements. However, each of the items listed has a direct influence upon economic considerations.

Direct or chemical methods tend to be unfavorable where:

1. Normal comfort dew-point temperatures are required with a predominantly sensible heat load and where mechanical refrigeration is required for sensible heat removal.

2. Water temperature is too high for practical sensible heat removal (above 65 F).
3. Cold water (below 55 F) is available in adequate quantities so that it can be directly used for both sensible and latent heat removal, or can be further chilled cheaply by mechanical refrigeration.
4. Electricity is low in cost compared with gas or steam.

No single unfavorable item listed will necessarily disqualify a method, but generally there will be several offsetting favorable factors required to make it the choice on a purely economic basis.

When analyzed with respect to the broadening scope of applications for air conditioning, it is now evident that direct or chemical dehumidification

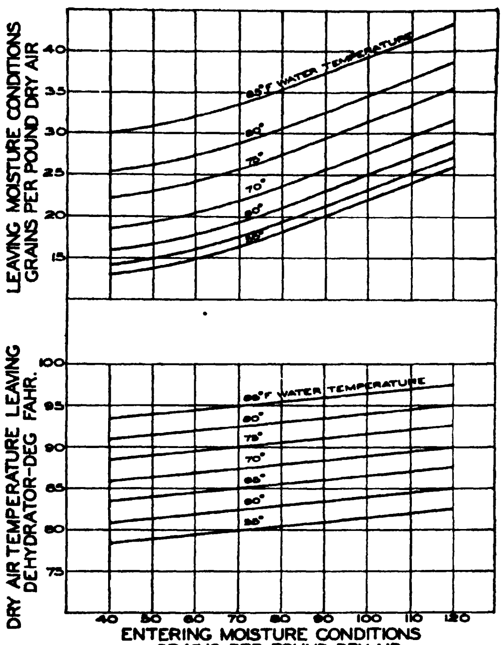

Fig. 10. Lithium Chloride Equipment Performance Data

can be used, within its legitimate economic limits, for human comfort, commercial drying or storage of food products requiring low humidities, and industrial processing and drying.

It provides an additional choice as to the type of equipment best suited to meet the requirements of special applications and conditions. Particular attention is called to those industrial and drying applications in which the dried air can be used at effluent temperature without further treatment.

REFERENCES

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CHAPTER 24

Refrigeration

Mechanical Refrigeration, Characteristics of Compression System, Absorption Systems, Expansion Valves, Condensers, Evaporators and Coolers, Refrigerant Pipe Sizes, Ice Systems, Storage Systems, Equipment Selection, Reverse Cycle

COOLING and dehumidification in air conditioning work usually require refrigeration equipment. The localities where cold water from a natural source is at a sufficiently low temperature for comfort air conditioning are rare, and evaporative cooling is generally restricted to sections of the country where humidities are naturally low.

The important difference between the refrigeration equipment used for comfort air conditioning and that used for commercial refrigeration is the use of a relatively higher evaporator temperature. This temperature is usually above freezing in air conditioning refrigeration equipment. The higher evaporator temperature (that is high suction pressure) affects the design of the system used, and makes possible the use of systems that are not always practical for commercial refrigeration.

MECHANICAL REFRIGERATION

The fundamentals of mechanical refrigeration systems are similar, although they differ in the methods used for compression of the refrigerant vapor.

Refrigerant vapor, usually saturated or slightly superheated, is drawn into the compressor as diagrammed in Fig. 1. It is then compressed and discharged at a higher pressure to a condenser. The vapor is condensed as it contacts a heat transfer surface over which is flowing a cooling medium such as water, air or a combination of the two. The liquid refrigerant flows to the evaporator through an expansion valve which reduces its pressure and regulates its flow. In the evaporator, the refrigerant absorbs heat from the medium which is to be cooled. When this medium is water or brine, the evaporator is known as a water or brine cooler and the refrigeration system, if used for air cooling, is known as an indirect system. When the medium cooled is air, the evaporator is known as a direct expansion cooler and the system is known as a direct expansion system.

Fundamentally, the function of the system is to absorb heat at one temperature and pump it to a higher temperature, where it may be removed by an available cooling medium. In order to conserve refrigerant, virtually all refrigeration systems are completely closed and the same refrigerant is recirculated.

The fundamental heat equations (disregarding losses) which should be kept in mind are: (1) the heat absorbed in the evaporator plus the heat added to the refrigerant during compression equals the heat rejected by the condenser; (2) the heat added to the refrigerant during compression is equal to the input to the compressor shaft less the heat dissipated from the compressor to the surroundings.

In the case where the compressor is driven by an electric motor, the heat due to compression is equal to the motor input less the electrical

motor losses, less the power transmission losses and less the heat dissipated from the compressor to the surroundings.

Refrigerants

There are many substances which might be used as refrigerants in mechanical refrigeration systems, but in practice the choice is limited by a wide variety of considerations including availability, cost, safety, chemical stability and adaptability to the type of refrigerating system to be used.

In this chapter detailed consideration is limited to six substances, viz: ammonia, dichlorodifluoromethane (F_{12}), methyl chloride, carbon dioxide, monofluorotrichloromethane (F_{11}), and water, properties for each of which are given in Tables 1, 2, 3, 4, 5 and 6. Each table gives the principal physical properties of the saturated substance, and all are arranged in uniform fashion. In all it will be noted that columns are included which give

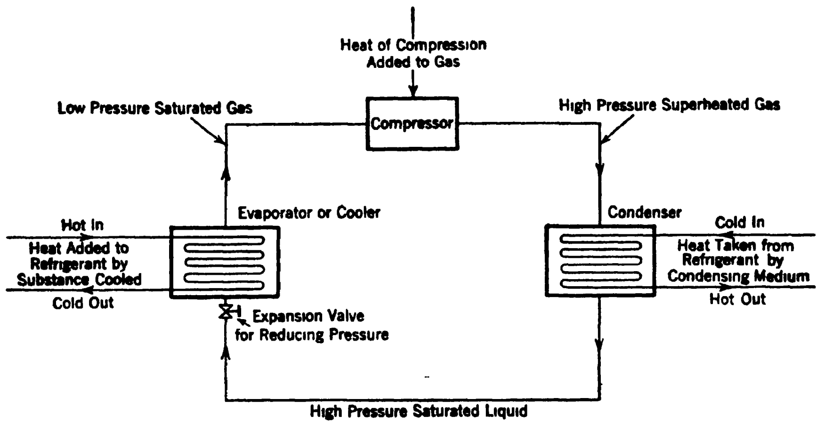

FIG. 1. MECHANICAL REFRIGERATION SYSTEM

the heat content and entropy of the superheated vapor at two selected points. The first four refrigerants named are used in reciprocating and rotary compressors. The last two are used in centrifugal compressors. Water is also used in steam jet equipment.

Types of Compressors

There are many different types of compressors, using various refrigerants. Each type has its advantages for its particular application, and those generally used for air conditioning are of the following types:

1. Reciprocating compressors (commonly referred to as piston type).
2. Centrifugal compressors.
3. Steam jet.

Reciprocating compressors are available in a wide range of sizes and types. Any of a number of refrigerants, including dichlorodifluoromethane (F_{12}), methyl chloride, ammonia, carbon dioxide, and sulphur dioxide may be used in reciprocating machines. The first of these is used extensively in direct expansion systems of comfort air conditioning.

Compressors may be classified into two general types, (*a*) open type,

Table 1. Properties of Ammonia

Sat. Temp F	Abs. Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	30.42	0.02419	9.116	42.9	611.8	0.0975	1.3352	666.8	1.4439	720.3	1.5317
2	31.92	0.02424	8.714	45.1	612.4	0.1022	1.3312	667.6	1.4400	721.2	1.5277
4	33.47	0.02430	8.333	47.2	613.0	0.1069	1.3273	668.4	1.4360	722.2	1.5236
5	34.27	0.02432	8.150	48.3	613.3	0.1092	1.3253	668.8	1.4340	722.6	1.5216
6	35.09	0.02435	7.971	49.4	613.6	0.1115	1.3234	669.3	1.4321	723.1	1.5196
8	36.77	0.02440	7.629	51.6	614.3	0.1162	1.3195	670.1	1.4281	724.1	1.5155
10	38.51	0.02446	7.304	53.8	614.9	0.1208	1.3157	670.9	1.4242	725.0	1.5115
12	40.31	0.02451	6.996	56.0	615.5	0.1254	1.3118	671.7	1.4205	725.9	1.5077
14	42.18	0.02457	6.703	58.2	616.1	0.1300	1.3081	672.5	1.4168	726.8	1.5039
16	44.12	0.02462	6.425	60.3	616.6	0.1346	1.3043	673.4	1.4130	727.8	1.5001
18	46.13	0.02468	6.161	62.5	617.2	0.1392	1.3006	674.2	1.4093	728.7	1.4963
20	48.21	0.02474	5.910	64.7	617.8	0.1437	1.2969	675.0	1.4056	729.6	1.4925
22	50.36	0.02479	5.671	66.9	618.3	0.1483	1.2933	675.8	1.4021	730.5	1.4889
24	52.59	0.02485	5.443	69.1	618.9	0.1528	1.2897	676.6	1.3985	731.4	1.4853
26	54.90	0.02491	5.227	71.3	619.4	0.1573	1.2861	677.3	1.3950	732.4	1.4816
28	57.28	0.02497	5.021	73.5	619.9	0.1618	1.2825	678.1	1.3914	733.3	1.4780
30	59.74	0.02503	4.825	75.7	620.5	0.1663	1.2790	678.9	1.3879	734.2	1.4744
32	62.29	0.02508	4.637	77.9	621.0	0.1708	1.2755	679.7	1.3846	735.1	1.4710
34	64.91	0.02514	4.459	80.1	621.5	0.1753	1.2721	680.4	1.3812	736.0	1.4676
36	67.63	0.02521	4.289	82.3	622.0	0.1797	1.2686	681.2	1.3779	736.8	1.4643
38	70.43	0.02527	4.126	84.6	622.5	0.1841	1.2652	681.9	1.3745	737.7	1.4609
39	71.87	0.02530	4.048	85.7	622.7	0.1863	1.2635	682.3	1.3729	738.2	1.4592
40	73.32	0.02533	3.971	86.8	623.0	0.1885	1.2618	682.7	1.3712	738.6	1.4575
41	74.80	0.02536	3.897	87.9	623.2	0.1908	1.2602	683.1	1.3696	739.0	1.4559
42	76.31	0.02539	3.823	89.0	623.4	0.1930	1.2585	683.4	1.3680	739.5	1.4542
44	79.38	0.02545	3.682	91.2	623.9	0.1974	1.2552	684.2	1.3648	740.4	1.4510
46	82.55	0.02551	3.547	93.5	624.4	0.2018	1.2519	684.9	1.3616	741.3	1.4477
48	85.82	0.02558	3.418	95.7	624.8	0.2062	1.2486	685.6	1.3584	742.2	1.4445
50	89.19	0.02564	3.294	97.9	625.2	0.2105	1.2453	686.4	1.3552	743.1	1.4412
52	92.66	0.02571	3.176	100.2	625.7	0.2149	1.2421	687.1	1.3521	744.0	1.4382
54	96.23	0.02577	3.063	102.4	626.1	0.2192	1.2389	687.8	1.3491	744.8	1.4351
56	99.91	0.02584	2.954	104.7	626.5	0.2236	1.2357	688.5	1.3460	745.7	1.4321
58	103.7	0.02590	2.851	106.9	626.9	0.2279	1.2325	689.2	1.3430	746.5	1.4290
60	107.6	0.02597	2.751	109.2	627.3	0.2322	1.2294	689.9	1.3399	747.4	1.4260
62	111.6	0.02604	2.656	111.5	627.7	0.2365	1.2262	690.6	1.3370	748.2	1.4231
64	115.7	0.02611	2.565	113.7	628.0	0.2408	1.2231	691.3	1.3341	749.1	1.4202
66	120.0	0.02618	2.477	116.0	628.4	0.2451	1.2201	691.9	1.3312	749.9	1.4172
68	124.3	0.02625	2.393	118.3	628.8	0.2494	1.2170	692.6	1.3283	750.8	1.4143
70	128.8	0.02632	2.312	120.5	629.1	0.2537	1.2140	693.3	1.3254	751.6	1.4114
72	133.4	0.02639	2.235	122.8	629.4	0.2579	1.2110	694.0	1.3226	752.4	1.4086
74	138.1	0.02646	2.161	125.1	629.8	0.2622	1.2080	694.6	1.3199	753.3	1.4059
76	143.0	0.02653	2.089	127.4	630.1	0.2664	1.2050	695.3	1.3171	754.1	1.4031
78	147.9	0.02661	2.021	129.7	630.4	0.2706	1.2020	695.9	1.3144	755.0	1.4004
80	153.0	0.02668	1.955	132.0	630.7	0.2749	1.1991	696.6	1.3116	755.8	1.3976
82	158.3	0.02675	1.892	134.3	631.0	0.2791	1.1962	697.2	1.3089	756.6	1.3949
84	163.7	0.02684	1.831	136.6	631.3	0.2833	1.1933	697.8	1.3063	757.4	1.3923
86	169.2	0.02691	1.772	138.9	631.5	0.2875	1.1904	698.5	1.3040	758.3	1.3896
88	174.8	0.02699	1.716	141.2	631.8	0.2917	1.1875	699.1	1.3010	759.1	1.3870
90	180.6	0.02707	1.661	143.5	632.0	0.2958	1.1846	699.7	1.2983	759.9	1.3843
92	186.6	0.02715	1.609	145.8	632.2	0.3000	1.1818	700.3	1.2957	760.7	1.3818
94	192.7	0.02723	1.559	148.2	632.5	0.3041	1.1789	700.9	1.2932	761.5	1.3793
96	198.9	0.02731	1.510	150.5	632.6	0.3083	1.1761	701.5	1.2906	762.2	1.3768
98	205.3	0.02739	1.464	152.9	632.9	0.3125	1.1733	702.1	1.2881	763.0	1.3743
100	211.9	0.02747	1.419	155.2	633.0	0.3166	1.1705	702.7	1.2855	763.8	1.3718
102	218.6	0.02756	1.375	157.6	633.2	0.3207	1.1677	703.3	1.2830	764.6	1.3693
104	225.4	0.02764	1.334	159.9	633.4	0.3248	1.1649	703.8	1.2805	765.3	1.3668
106	232.5	0.02773	1.293	162.3	633.5	0.3289	1.1621	704.3	1.2780	766.1	1.3643
108	239.7	0.02782	1.254	164.6	633.6	0.3330	1.1593	705.0	1.2755	766.9	1.3619
110	247.0	0.02790	1.217	167.0	633.7	0.3372	1.1566	705.5	1.2731	767.6	1.3596
112	254.5	0.02799	1.180	169.4	633.8	0.3413	1.1538	706.1	1.2708	768.3	1.3573
114	262.2	0.02808	1.145	171.8	633.9	0.3453	1.1510	706.6	1.2684	769.1	1.3550
116	270.1	0.02817	1.112	174.2	634.0	0.3495	1.1483	707.2	1.2661	769.8	1.3527
118	278.2	0.02827	1.079	176.6	634.0	0.3535	1.1455	707.7	1.2636	770.5	1.3503
120	286.4	0.02836	1.047	179.0	634.0	0.3576	1.1427	708.2	1.2612	771.3	1.3479
122	294.8	0.02846	1.017	181.4	634.0	0.3618	1.1400	708.6	1.2587	772.0	1.3455
124	303.4	0.02855	0.987	183.9	634.0	0.3659	1.1372	709.1	1.2563	772.8	1.3431
126	312.2	0.02865	0.958	186.3	633.9	0.3700	1.1344	709.6	1.2538	773.5	1.3407
128	321.2	0.02875	0.931	188.8	633.9	0.3741	1.1316	710.0	1.2513	774.2	1.3383

TABLE 2. PROPERTIES OF DICHLORODIFLUOROMETHANE(F_{12})

Sat Temp F	Abs Press Lb per Sq In.	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht Ct.	Entropy	Ht. Ct.	Entropy
0	23.87	0 0110	1.637	8.25	78.21	0 01869	0 17091	81.71	0.17829	85.26	0.18547
2	24 89	0.0110	1.574	8 67	78.44	0.01961	0 17075	81.94	0 17812	85.51	0.18529
4	25 96	0.0111	1.514	9 10	78.67	0 02052	0 17060	82.17	0 17795	85 76	0 18511
5	26 51	0.0111	1.485	9 32	78.79	0 02097	0 17052	82.29	0.17786	85 89	0.18502
6	27.05	0.0111	1.457	9.53	78.90	0 02143	0.17045	82.41	0.17778	86.01	0.18494
8	28.18	0.0111	1.403	9 96	79.13	0 02235	0.17030	82 66	0.17763	86 26	0.18477
10	29.35	0.0112	1.351	10 39	79 36	0.02328	0.17015	82.90	0.17747	86.51	0 18460
12	30 56	0.0112	1.301	10 82	79.59	0.02419	0 17001	83 14	0.17733	86 76	0 18444
14	31 80	0.0112	1 253	11.26	79 82	0 02510	0 16987	83 38	0.17720	87.01	0 18429
16	33 08	0.0112	1.207	11.70	80.05	0.02601	0 16974	83 61	0.17706	87.26	0 18413
18	34 40	0 0113	1.163	12 12	80.27	0 02692	0 16961	83 85	0.17693	87 51	0 18397
20	35 75	0.0113	1.121	12 55	80 49	0.02783	0 16949	84 09	0.17679	87.76	0.18382
22	37.15	0.0113	1.081	13.00	80 72	0 02873	0 16938	84 32	0.17666	88 00	0 18369
24	38 58	0 0113	1.043	13.44	80 95	0 02963	0.16926	84 55	0.17652	88 24	0 18355
26	40 07	0 0114	1 007	13 88	81.17	0.03053	0 16913	84 79	0.17639	88 49	0.18342
28	41.59	0.0114	0 973	14.32	81.39	0 03143	0 16900	85 02	0 17625	88 73	0 18328
30	43.16	0 0115	0 939	14.76	81.61	0 03233	0 16887	85.25	0 17612	88 97	0 18315
32	44.77	0.0115	0.908	15 21	81.83	0 03323	0.16876	85 48	0 17600	89 21	0 18303
34	46.42	0 0115	0.877	15.65	82.05	0.03413	0 16865	85.71	0 17589	89.45	0.18291
36	48 13	0.0116	0.848	16 10	82.27	0 03502	0 16854	85.95	0.17577	89.68	0.18280
38	49 88	0 0116	0 819	16 55	82 49	0 03591	0 16843	86 18	0.17566	89 92	0 18268
39	50 78	0 0116	0.806	16 77	82 60	0 03635	0 16838	86 29	0 17560	90 04	0 18262
40	51 68	0.0116	0 792	17 00	82 71	0 03680	0 16833	86 41	0 17554	90 16	0.18256
41	52 70	0 0116	0.779	17.23	82 82	0 03725	0 16828	86 52	0.17549	90 28	0 18251
42	53 51	0.0116	0 767	17 46	82.93	0 03770	0.16823	86 64	0.17544	90 40	0 18245
44	55 40	0 0117	0 742	17.91	83 15	0 03859	0.16813	86 86	0.17534	90.65	0 18235
46	57.35	0.0117	0 718	18.36	83.36	0 03948	0 16803	87.09	0 17525	90.89	0 18224
48	59 35	0 0117	0 695	18 82	83.57	0 04037	0 16794	87.31	0.17515	91 14	0 18214
50	61.39	0 0118	0.673	19.27	83 78	0 04126	0.16785	87 54	0.17505	91 38	0 18203
52	63 49	0 0118	0.652	19.72	83.99	0 04215	0 16776	87.76	0.17496	91.61	0.18193
54	65 63	0 0118	0 632	20.18	84 20	0 04304	0 16767	87 98	0 17486	91 83	0 18184
56	67 84	0 0119	0.612	20 64	84 41	0 04392	0 16758	88.20	0 17477	92 06	0 18174
58	70 10	0 0119	0.593	21 11	84 62	0 04480	0 16749	88 42	0.17467	92 28	0.18165
60	72.41	0 0119	0 575	21.57	84 82	0 04568	0.16741	88 64	0.17458	92 51	0.18155
62	74 77	0.0120	0.557	22.03	85 02	0 04657	0 16733	88.86	0.17450	92 74	0 18147
64	77 20	0.0120	0.540	22.49	85 22	0.04745	0 16725	89 07	0 17442	92 97	0.18139
66	79.67	0.0120	0.524	22.95	85 42	0 04833	0.16717	89 29	0 17433	93 20	0 18130
68	82 24	0.0121	0.508	23 42	85 62	0 04921	0 16709	89 50	0 17425	93.43	0 18122
70	84.82	0 0121	0.493	23 90	85.82	0 05009	0 16701	89 72	0 17417	93 66	0 18114
72	87 50	0.0121	0.479	24 37	86 02	0 05097	0 16693	89 93	0 17409	93 99	0 18106
74	90 20	0 0122	0.464	24 84	86.22	0 05185	0 16685	90 14	0 17402	94 12	0 18098
76	93 00	0.0122	0 451	25.32	86 42	0 05272	0 16677	90 36	0 17394	94 34	0 18091
78	95 85	0 0123	0.438	25 80	86.61	0.05359	0 16669	90 57	0.17387	94 57	0 18083
80	98 76	0 0123	0 425	26 28	86 80	0 05446	0 16662	90 78	0 17379	94 80	0 18075
82	101.70	0.0123	0 413	26 76	86.99	0.05534	0.16655	90 98	0.17372	95.01	0 18068
84	104 8	0.0124	0 401	27 24	87.18	0 05621	0 16648	91 18	0.17365	95 22	0 18061
86	107.9	0 0124	0.389	27 72	87.37	0.05708	0 16640	91 37	0.17358	95.44	0 18054
88	111.1	0 0124	0 378	28 21	87.56	0.05795	0.16632	91.57	0.17351	95 65	0 18047
90	114 3	0 0125	0.368	28.70	87.74	0 05882	0 16624	91.77	0 17344	95 86	0 18040
92	117.7	0.0125	0.357	29.19	87.92	0 05969	0 16616	91 97	0 17337	96 07	0 18033
94	121 0	0.0126	0.347	29.68	88.10	0.06056	0 16608	92 16	0 17330	96.28	0.18026
96	124.5	0 0126	0.338	30.18	88.28	0.06143	0 16600	92.36	0 17322	96 50	0 18018
98	128.0	0.0126	0.328	30.67	88 45	0 06230	0.16592	92.55	0 17315	96.71	0 18011
100	131.6	0 0127	0.319	31.16	88 62	0 06316	0 16584	92 75	0.17308	96 92	0 18004
102	135.3	0 0127	0 310	31.65	88.79	0 06403	0.16576	92 93	0.17301	97.12	0 17998
104	139.0	0.0128	0 302	32 15	88 95	0.06490	0.16568	93 11	0 17294	97.32	0 17993
106	142.8	0.0128	0 293	32 65	89 11	0 06577	0 16560	93.30	0.17288	97.53	0 17987
108	146.8	0.0129	0 285	33.15	89 27	0 06663	0 16551	93.48	0 17281	97.73	0 17982
110	150.7	0.0129	0 277	33 65	89.43	0 06749	0 16542	93.66	0.17274	97 93	0.17976
112	154 8	0 0130	0.269	34 15	89 58	0 06836	0 16533	93 82	0 17266	98 11	0.17969
114	158.9	0 0130	0.262	34 65	89.73	0 06922	0.16524	93 98	0 17258	98.29	0 17961
116	163 1	0 0131	0 254	35 15	89.87	0 07008	0 16515	94 15	0.17249	98 48	0 17954
118	167 4	0.0131	0 247	35 65	90 01	0 07094	0 16505	94 31	0 17241	98 66	0 17946
120	171.8	0.0132	0.240	36 16	90 15	0 07180	0 16495	94 47	0 17233	98 84	0 17939
122	176.2	0 0132	0.233	36 66	90 28	0 07266	0 16484	94 63	0 17224	99.01	0 17931
124	180 8	0.0133	0 227	37 16	90 40	0 07352	0 16473	94 78	0.17215	99 18	0 17922
126	185 4	0 0133	0.220	37 67	90 52	0 07437	0.16462	94 94	0 17206	99 35	0 17914
128	190.1	0 0134	0 214	38 18	90 64	0 07522	0 16450	95.09	0 17196	99 53	0 17906
130	194.9	0 0134	0 208	38 69	90 76	0 07607	0 16438	95 25	0.17186	99 70	0 17897
132	199.8	0.0135	0 202	39 19	90 86	0.07691	0.16425	95.41	0.17176	99 87	0 17889
134	204 8	0 0135	0 196	39 70	90.96	0 07775	0.16411	95.56	0 17166	100 04	0 17881
136	209 9	0.0136	0 191	40.21	91 06	0.07858	0 16396	95.72	0.17156	100 22	0 17873
138	215 0	0 0137	0 185	40 72	91 15	0 07941	0 16380	95.87	0 17145	100.39	0 17864
140	220.2	0.0138	0.180	41.24	91.24	0.08024	0.16363	96 03	0.17134	100.56	0 17856

Table 3. Properties of Methyl Chloride

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	18.73	0.0162	5.052	14.4	192.4	0.0328	0.4197	215.6	0.467	237.2	0.507
2	19.60	0.0162	4.856	15.1	193.1	0.0344	0.4196	216.2	0.466	237.7	0.505
4	20.47	0.0163	4.661	15.8	193.8	0.0360	0.4195	216.7	0.465	238.2	0.504
5	20.91	0.0163	4.563	16.2	194.1	0.0368	0.4195	217.0	0.464	238.5	0.503
6	21.39	0.0163	4.476	16.6	194.4	0.0376	0.4194	217.3	0.464	238.8	0.502
8	22.34	0.0164	4.303	17.3	195.1	0.0391	0.4193	217.9	0.463	239.4	0.501
10	23.30	0.0164	4.129	18.1	195.8	0.0407	0.4192	218.5	0.463	240.0	0.500
12	24.38	0.0164	3.984	18.8	196.3	0.0423	0.4184	219.0	0.462	240.5	0.499
14	25.46	0.0164	3.839	19.6	196.7	0.0439	0.4176	219.5	0.462	241.0	0.498
16	26.55	0.0165	3.693	20.3	197.2	0.0454	0.4168	220.0	0.461	241.5	0.498
18	27.63	0.0165	3.548	21.1	197.6	0.0472	0.4160	220.5	0.461	242.0	0.497
20	28.71	0.0166	3.403	21.8	198.1	0.0486	0.4152	221.0	0.460	242.5	0.496
22	29.98	0.0166	3.288	22.5	198.5	0.0501	0.4148	221.5	0.459	243.0	0.495
24	31.25	0.0166	3.172	23.3	198.9	0.0516	0.4143	222.0	0.459	243.6	0.495
26	32.53	0.0167	3.057	24.0	199.3	0.0532	0.4139	222.4	0.458	244.1	0.494
28	33.80	0.0167	2.941	24.8	199.7	0.0547	0.4134	222.9	0.458	244.7	0.494
30	35.07	0.0168	2.826	25.5	200.1	0.0562	0.4130	223.4	0.457	245.2	0.493
32	36.55	0.0168	2.734	26.2	200.5	0.0577	0.4124	223.9	0.456	245.7	0.492
34	38.03	0.0169	2.642	27.0	200.9	0.0592	0.4118	224.3	0.455	246.2	0.492
36	39.51	0.0169	2.549	27.7	201.4	0.0607	0.4111	224.8	0.455	246.7	0.491
38	40.99	0.0169	2.457	28.5	201.8	0.0622	0.4105	225.2	0.454	247.2	0.491
39	41.73	0.0170	2.411	28.8	202.0	0.0629	0.4102	225.5	0.453	247.4	0.490
40	42.47	0.0170	2.365	29.2	202.2	0.0637	0.4099	225.7	0.453	247.7	0.490
41	43.33	0.0170	2.328	29.6	202.4	0.0644	0.4096	225.9	0.453	248.0	0.490
42	44.18	0.0171	2.290	29.9	202.6	0.0651	0.4093	226.1	0.452	248.3	0.489
44	45.89	0.0171	2.216	30.7	203.0	0.0666	0.4087	226.6	0.451	248.8	0.489
46	47.61	0.0171	2.141	31.4	203.3	0.0680	0.4081	227.0	0.451	249.4	0.488
48	49.32	0.0172	2.067	32.2	203.7	0.0695	0.4075	227.5	0.450	249.9	0.488
50	51.03	0.0172	1.992	32.9	204.1	0.0709	0.4069	227.9	0.449	250.5	0.487
52	53.00	0.0172	1.931	33.7	204.4	0.0724	0.4063	228.2	0.448	251.0	0.486
54	54.97	0.0173	1.870	34.4	204.7	0.0739	0.4056	228.6	0.448	251.5	0.486
56	56.94	0.0173	1.810	35.2	205.1	0.0754	0.4050	228.9	0.447	252.0	0.485
58	58.91	0.0173	1.749	35.9	205.4	0.0769	0.4043	229.3	0.447	252.5	0.485
60	60.88	0.0174	1.688	36.7	205.7	0.0784	0.4037	229.6	0.446	253.0	0.484
62	63.13	0.0174	1.638	37.4	206.0	0.0798	0.4030	229.9	0.445	253.5	0.483
64	65.37	0.0174	1.588	38.2	206.3	0.0812	0.4024	230.3	0.444	254.0	0.483
66	67.62	0.0175	1.539	38.9	206.6	0.0827	0.4017	230.6	0.443	254.5	0.482
68	69.86	0.0175	1.489	39.7	206.9	0.0841	0.4011	231.0	0.442	255.0	0.482
70	72.11	0.0176	1.439	40.4	207.2	0.0855	0.4004	231.3	0.441	255.5	0.481
72	74.66	0.0176	1.398	41.1	207.5	0.0869	0.3998	231.6	0.440	256.0	0.480
74	77.21	0.0177	1.357	41.9	207.7	0.0883	0.3992	232.0	0.439	256.5	0.480
76	79.76	0.0177	1.315	42.6	208.0	0.0898	0.3985	232.3	0.439	256.9	0.479
78	82.31	0.0178	1.274	43.4	208.2	0.0912	0.3979	232.7	0.438	257.4	0.479
80	84.86	0.0178	1.233	44.1	208.5	0.0926	0.3973	233.0	0.437	257.9	0.478
82	87.74	0.0178	1.199	44.8	208.7	0.0940	0.3967	233.3	0.436	258.4	0.478
84	90.62	0.0179	1.165	45.6	209.0	0.0953	0.3960	233.6	0.435	258.9	0.477
86	93.50	0.0179	1.130	46.3	209.2	0.0967	0.3954	233.9	0.435	259.4	0.477
88	96.38	0.0180	1.096	47.1	209.5	0.0980	0.3947	234.2	0.434	259.9	0.476
90	99.26	0.0180	1.062	47.8	209.7	0.0994	0.3941	234.5	0.433	260.4	0.476
92	102.49	0.0180	1.033	48.6	209.9	0.1008	0.3935	234.8	0.433	260.8	0.476
94	105.72	0.0181	1.005	49.3	210.2	0.1022	0.3929	235.1	0.432	261.2	0.475
96	108.94	0.0181	0.9764	50.1	210.4	0.1035	0.3922	235.4	0.432	261.6	0.475
98	112.17	0.0182	0.9478	50.8	210.7	0.1049	0.3916	235.7	0.431	262.0	0.474
100	115.40	0.0182	0.9193	51.6	210.9	0.1063	0.3910	236.0	0.431	262.4	0.474
102	119.00	0.0183	0.8952	52.3	211.1	0.1076	0.3903	236.4	0.430	262.8	0.474
104	122.60	0.0183	0.8712	53.1	211.3	0.1090	0.3897	236.8	0.430	263.2	0.473
106	126.20	0.0184	0.8471	53.8	211.4	0.1103	0.3890	237.1	0.429	263.5	0.473
108	129.80	0.0184	0.8231	54.6	211.6	0.1117	0.3884	237.5	0.429	263.9	0.472
110	133.40	0.0185	0.7990	55.3	211.8	0.1130	0.3877	237.9	0.428	264.3	0.472
112	137.42	0.0185	0.7786	56.1	212.0	0.1144	0.3871	238.1	0.427	264.6	0.471
114	141.44	0.0185	0.7583	56.8	212.2	0.1157	0.3864	238.3	0.427	264.8	0.470
116	145.46	0.0186	0.7379	57.6	212.4	0.1171	0.3858	238.6	0.426	265.1	0.470
118	149.48	0.0186	0.7176	58.3	212.6	0.1184	0.3851	238.8	0.426	265.3	0.469
120	153.50	0.0187	0.6972	59.1	212.8	0.1198	0.3845	239.0	0.425	265.6	0.468

TABLE 4. PROPERTIES OF CARBON DIOXIDE

Sat. Temp. F	Abs. Press. Lb per Sq In	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		50 F Superheat		100 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht Ct.	Entropy	Ht. Ct.	Entropy
0	305.5	0.01570	0 29040	18 8	138 9	0 0418	0.3024	153.7	0.3342	167.5	0.3612
2	315.9	0.01579	0 28030	19 8	138 8	0 0440	0.3014	153.7	0 3330	167.6	0.3600
4	326.5	0.01588	0.27070	20 8	138 8	0.0461	0.3005	153.7	0.3318	167.7	0.3588
5	332.0	0 01592	0 26610	21.3	138 8	0.0472	0.3000	153.7	0.3312	167 7	0 3582
6	337.4	0.01596	0 26140	21.8	138 7	0 0483	0.2994	153 7	0.3306	167 8	0.3576
8	348.7	0.01605	0 25260	22 9	138 7	0 0504	0.2982	153.7	0.3293	167 9	0.3563
10	360 2	0.01614	0 24370	24.0	138.7	0 0526	0.2970	153 7	0.3281	168 0	0.3550
12	371 9	0.01623	0.23540	25 0	138.6	0 0548	0.2958	153.7	0.3270	168 1	0.3538
14	383.9	0 01632	0.22740	26.1	138 6	0 0571	0.2946	153 7	0.3259	168.2	0.3526
16	396 2	0 01642	0 21970	27.2	138 5	0 0593	0 2933	153.7	0 3249	168.3	0.3513
18	408.9	0.01652	0.21210	28 3	138.4	0 0616	0 2921	153 7	0.3238	168 5	0.3501
20	421 8	0.01663	0.20490	29 4	138 3	0 0638	0.2909	153.7	0.3227	168.6	0.3489
22	434.0	0.01673	0 19790	30.5	138 2	0 0662	0.2897	153 7	0.3214	168 7	0.3479
24	448 4	0 01684	0 19120	31.7	138 1	0 0686	0 2885	153.7	0.3202	168 8	0.3470
26	462.2	0.01695	0.18460	32 9	138 0	0 0710	0.2873	153.7	0.3189	168 9	0.3460
28	476 3	0 01707	0.17830	34 1	137 9	0 0734	0 2861	153 7	0 3177	169 0	0 3451
30	490 8	0 01719	0.17220	35 4	137 8	0 0758	0 2849	153.7	0 3164	169 1	0 3441
32	505 5	0 01731	0 16630	36 7	137 7	0 0781	0 2834	153 7	0 3158	169.2	0 3431
34	522 6	0 01744	0.16030	37 9	137 4	0 0804	0 2820	153.7	0 3151	169 3	0 3421
36	536.0	0 01759	0 15500	39 1	137 2	0 0828	0 2805	153 7	0 3145	169 4	0 3411
38	551 7	0.01773	0 14960	40 4	136 9	0 0851	0 2791	153 7	0 3138	169 5	0 3401
39	559 7	0.01780	0.14700	41 0	136 8	0 0862	0 2783	153 7	0 3135	169 5	0 3396
40	567 8	0 01787	0.14440	41.7	136 7	0 0874	0 2776	153 7	0 3132	169 6	0 3391
41	576 0	0 01794	0.14185	42 3	136 5	0 0887	0 2768	153 7	0 3127	169 6	0 3386
42	584.3	0 01801	0 13930	42 9	136 3	0 0899	0 2761	153 7	0 3122	169 7	0 3381
44	601.1	0.01817	0.13440	44 3	136 1	0 0924	0 2745	153 7	0 3112	169 8	0 3371
46	618 2	0 01834	0.12970	45 6	135.7	0 0950	0.2730	153 7	0 3101	169 9	0 3362
48	635 7	0 01851	0 12500	47 0	135 4	0 0975	0 2714	153 7	0 3091	170 0	0 3352
50	653 6	0 01868	0.12050	48 4	135 0	0 1000	0 2699	153 7	0 3081	170 1	0 3342
52	671 9	0 01887	0 11610	49 8	134 5	0 1027	0 2681	153 7	0 3069	170 2	0 3333
54	690 6	0 01906	0 11170	51 2	133 9	0 1054	0 2663	153 7	0 3057	170 3	0 3324
56	709.5	0 01927	0 10750	52 6	133 4	0 1081	0 2644	153 7	0 3046	170 5	0 3315
58	728 8	0 01948	0 10340	54 0	132 7	0 1108	0 2626	153 7	0 3034	170 6	0 3306
60	748.6	0 01970	0 09940	55 5	132 1	0 1135	0 2608	153 7	0 3022	170 7	0 3297
62	769 0	0 01995	0 09545	57 0	131 3	0 1164	0 2584	153 7	0 3012	170 8	0 3289
64	789 4	0 02020	0 09180	58 6	130 6	0 1194	0 2560	153 7	0 3002	170 9	0 3281
66	810.3	0 02048	0 08800	60 2	129 7	0 1223	0 2535	153 7	0.2991	171 0	0 3273
68	831.6	0 02079	0 08422	61 9	128 7	0 1253	0 2511	153 7	0 2981	171 1	0 3265
70	853.4	0 02112	0.08040	63 7	127 5	0 1282	0 2487	153 7	0 2971	171 2	0 3257
72	875 8	0 02152	0 07654	65 5	126 0	0.1321	0 2450	153 7	0.2962	171 3	0 3250
74	898.2	0 02192	0 07269	67 3	124 5	0 1360	0 2414	153 7	0 2953	171 4	0 3242
76	921.3	0 02242	0 06875	69 4	122 8	0 1398	0 2377	153 7	0 2945	171 5	0 3235
78	944 8	0 02300	0 06473	71 6	120 9	0 1437	0.2341	153 7	0 2936	171 6	0 3227
80	968 7	0 02370	0 06064	73 9	118 7	0 1476	0 2304	153 7	0 2927	171 7	0 3220
82	993 0	0 02456	0.05648	76 4	116 6	0 1578	0 2195	153 7	0 2920	173 8	0 3215
84	1017.7	0 02553	0 05223	79 4	113 9	0 1679	0 2087	153 7	0 2914	176 0	0 3209
86	1043 0	0 02686	0 04789	83 3	110 4	0 1781	0 1978	153 7	0 2907	178 2	0 3204
87 8	1069 9	0 03454	0 03454	97 0	97 0	0 1880	0 1880	153 7	0 2901	180 1	0 3199

(*b*) enclosed type. If the driving mechanism is external to the compressor, then the shaft must be brought out through the crankcase and a shaft seal or stuffing box must be used to prevent escape of the refrigerant. This type of compressor is known as an open-type compressor. When the driving mechanism is located within the crankcase of the compressor in such a way as to avoid the necessity of a shaft seal, the compressor is known as the completely enclosed or hermetically sealed type.

Open type compressors may be further classified as belt driven and directly connected. A great number of direct-driven units are now being used which generally operate at higher rotational speeds than the belt-driven type.

The present tendency is toward forced lubrication of the bearings of compressors by means of an oil pump driven from the crankshaft, although there are many splash lubricated compressors on the market. The chief

TABLE 5. PROPERTIES OF MONOFLUOROTRICHLOROMETHANE (F_{11})

SAT. TEMP. F	ABS. PRESS. LB PER SQ IN.	VOLUME		HEAT CONTENT AND ENTROPY TAKEN FROM −40 F							
				Heat Content		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht Ct.	Entropy	Ht. Ct.	Entropy
0	2.59	0.01020	13.700	7.81	90.4	0.0178	0.1975	93.9	0.2049	97.4	0.2120
5	2.96	0.01024	12.100	8.81	91.2	0.0200	0.1974	94.7	0.2047	98.2	0.2117
10	3.38	0.01028	10.700	9.82	92.0	0.0222	0.1973	95.5	0.2045	99.0	0.2114
15	3.85	0.01032	9.530	10.80	92.8	0.0243	0.1971	96.3	0.2043	99.8	0.2111
20	4.36	0.01036	8.490	11.90	93.7	0.0264	0.1970	97.2	0.2041	100.7	0.2109
25	4.94	0.01040	7.580	12.90	94.5	0.0286	0.1969	98.0	0.2039	101.5	0.2107
30	5.57	0.01045	6.770	13.90	95.3	0.0307	0.1969	98.8	0.2038	102.3	0.2105
35	6.27	0.01049	6.080	14.90	96.1	0.0328	0.1968	99.6	0.2037	103.1	0.2103
40	7.03	0.01053	5.460	16.00	96.8	0.0349	0.1968	100.3	0.2036	103.8	0.2101
45	7.88	0.01057	4.920	17.00	97.6	0.0370	0.1967	101.1	0.2035	104.6	0.2099
50	8.79	0.01062	4.440	18.10	98.4	0.0391	0.1967	101.9	0.2034	105.4	0.2098
55	9.80	0.01066	4 020	19.10	99.2	0.0412	0.1967	102.7	0.2033	106.2	0.2097
60	10.90	0.01071	3.640	20.20	100.0	0.0432	0.1967	103.5	0.2033	107.0	0.2096
65	12.10	0.01076	3.300	21.30	100.8	0.0453	0.1967	104.3	0.2032	107.8	0.2094
70	13.40	0.01081	3.000	22.40	101.5	0.0473	0.1967	105.0	0.2032	108.5	0.2093
75	14.80	0.01086	2.740	23.50	102.2	0.0493	0.1967	105.7	0.2031	109.2	0.2092
80	16.30	0.01091	2.500	24.50	102.9	0.0513	0.1966	106.4	0.2030	109.9	0.2090
85	17.90	0.01096	2 280	25.60	103.6	0.0533	0.1966	107.1	0.2029	110.6	0.2089
90	19.70	0.01101	2.090	26 70	104.4	0.0553	0.1966	107.9	0.2028	111.4	0.2088
95	21.60	0.01106	1.918	27.80	105.1	0.0573	0.1966	108.6	0.2028	112.1	0.2087
100	23.60	0.01111	1.761	28.90	105.7	0.0593	0.1965	109.2	0 2027	112.7	0.2085
105	25.90	0.01116	1 620	30.10	106.4	0.0613	0.1965	109.9	0.2026	113.4	0 2084

TABLE 6. PROPERTIES OF WATER

SAT TEMP F	ABS. PRESS LB PER SQ IN	VOLUME		HEAT CONTENT AND ENTROPY TAKEN FROM +32 F							
				Heat Content		Entropy		50 F Superheat		100 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct	Entropy	Ht. Ct.	Entropy
32	0 0887	0 01602	3296.0	0 00	1073.0	0.0000	2.1826	1096.9	2.2277	1120.8	2.2688
35	0.1000	0 01602	2941 0	3.02	1074.4	0.0062	2.1724	1098.3	2.2172	1122.2	2 2581
40	0.1217	0.01602	2441.0	8.05	1076.8	0.0163	2.1555	1100.6	2.2000	1124.5	2.2406
45	0.1475	0.01602	2034 0	13.07	1079.2	0.0262	2.1390	1102.9	2.1832	1126.7	2.2234
50	0.1780	0.01602	1702.0	18.08	1081.5	0.0361	2.1230	1105.2	2.1667	1129.0	2 2066
55	0.2140	0.01603	1430.0	23.08	1083.9	0.0459	2.1073	1107.5	2.1506	1131.3	2.1902
60	0.2561	0.01603	1206 0	28.08	1086.2	0 0556	2.0920	1109.8	2.1349	1133.5	2.1742
65	0.3054	0.01604	1021.0	33.08	1088.6	0.0652	2.0771	1112 2	2.1196	1135.8	2.1585
70	0.3628	0.01605	868.0	38.07	1090.9	0 0746	2.0625	1114.5	2.1046	1138.1	2.1432
75	0.4295	0.01606	740.0	43.06	1093.2	0.0840	2.0483	1116.7	2.0900	1140.3	2.1283
80	0.507	0.01607	632.9	48 05	1095.5	0.0933	2.0344	1119.0	2.0758	1142.5	2.1138
85	0.596	0.01609	543.3	53 04	1097.8	0.1025	2.0208	1121.2	2.0619	1144.7	2.0996
90	0.698	0.01610	467.9	58 03	1100.0	0.1116	2.0075	1123.4	2.0483	1146.8	2.0857
95	0.815	0.01612	404.2	63.01	1102.3	0.1206	1.9946	1125.6	2.0350	1148.9	2.0721
100	0.949	0.01613	350.3	68.00	1104.6	0.1296	1.9819	1127.9	2.0220	1151.1	2 0588
105	1.101	0.01615	304.4	72 98	1106.8	0.1384	1.9695	1130 2	2.0093	1153.2	2.0458

For properties of steam at high temperatures, see Table 8, Chapter 1.

advantages of the forced lubricated compressor are that the lubrication system requires less energy for its operation than the splash type, the oil can be easily filtered before it enters the bearings, and less oil is usually required.

The compressor capacity must be selected for and matched to the maximum load for the installation on which it is to be used. Air-conditioning loads, however, vary over a wide range, and a wide fluctuation in air conditions may result during periods of light load if on-and-off control of full compressor capacity is used. To prevent such undesirable fluctuation, several methods are employed to vary the capacity of reciprocating compressors, such as:

1. By-passing one or more cylinders, of a multi-cylinder compressor, from discharge to suction.
2. Rendering the suction valves of one or more cylinders of a multi-cylinder compressor inoperative. This is usually accomplished by depressing the suction valves.
3. Varying the speed of the compressor, usually by using variable speed or two-speed electric motors.
4. Using clearance pockets to control the quantity of refrigerant pumped.
5. Restricting the suction inlet to one or more of the cylinders of a multi-cylinder compressor either by an automatic modulating valve or by an *on* and *off* valve.

All of these methods, with the exception of variable speed, result in a slightly lowered over-all compressor efficiency when in use, since the mechanical losses remain constant whereas the quantity of refrigerant pumped is lowered.

Centrifugal compressors are used with very low pressure refrigerants; usually both evaporator and condenser work below atmospheric pressure. Water and monofluorotrichloromethane (F_{11}) are the refrigerants commonly used in centrifugal machines.

Compression of the refrigerant is accomplished by means of centrifugal force; therefore, this type of compressor is inherently suitable for large volumes of refrigerant at low pressure differentials. Two or more stages are usually required and high speeds are necessary to obtain good efficiency.

The evaporator is usually constructed as an integral part of the centrifugal type condensing unit, to chill water which is then circulated to the air conditioning system. This is done because it would not be economical to pipe these large volumes of refrigerant any distance.

Centrifugal compressors like reciprocating compressors can be divided into two general types, open and enclosed. In general, the open type compressor is geared to the driving mechanism, and operates at higher speed than the driving motor or turbine. A modern completely enclosed direct-driven, centrifugal compressor is illustrated in Fig. 2.

The compressor capacity can be varied by controlling the condensing pressure. This is accomplished by regulating the quantity and temperature of the condenser cooling water. The capacity falls off with increasing condensing pressure. Centrifugal compressors are seldom built for less than 50 tons capacity, since it is not practical to make impellers which pump much less than the volume of refrigerant required for this tonnage.

The steam jet type of compressor, under certain circumstances, is desirable for use in air conditioning[1]. Steam supplies directly the power

[1]Application and Economy of Steam Jet Refrigeration to Air Conditioning, by A. R. Mumford and A. A. Markson (A.S H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 33).

used for compressing the refrigerant, thus eliminating the losses connected with other methods of supplying energy. As the compression ratio between the evaporator and condenser under normal circumstances is large, the mechanical efficiency of the equipment is somewhat lower than that of the positive mechanical type compressor. The condensing water requirements are considerably greater, as both the refrigerant and the impelling steam must be condensed.

The steam jet system functions on the principle that water under high vacuum will vaporize at low temperatures. Steam jet boosters or compressors of the type commonly used in power plants for various processes will produce the necessary low absolute pressure to cause evaporation of the water.

A diagrammatic representation of a typical steam ejector water cooling system is shown in Fig. 3. The figures correspond to an average representative system. The water to be cooled enters the evaporator and is cooled to a temperature corresponding to the vacuum maintained.

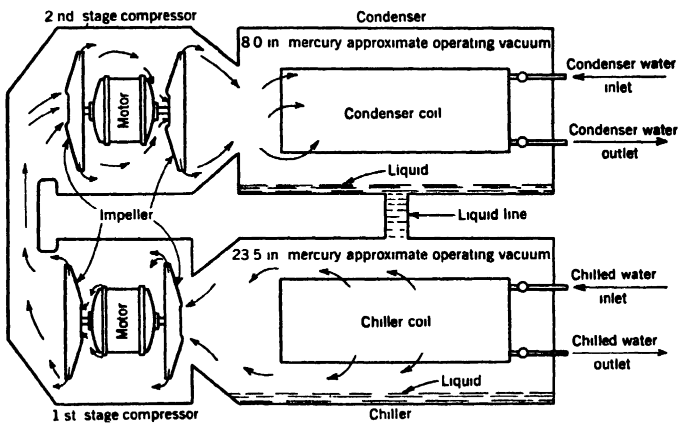

FIG. 2. ENCLOSED TYPE CENTRIFUGAL CONDENSING UNIT

Because of the high vacuum, a small amount of the water introduced in the evaporator is flashed into steam. As this requires heat, and the only source of heat is the rest of the water in the evaporator tank, this other water is almost instantly cooled to a temperature corresponding to the boiling point determined by the vacuum maintained. The amount of water flashed into steam is a small percentage of the total water circulated through the evaporator, amounting to approximately 11 lb per hour per ton of refrigeration developed. The remainder of the water at the desired low temperature is pumped out of the evaporator and used at the point where it is required.

The ejector compresses the vapor which has been flashed in the evaporator, plus any entrained air taken from the circulated water, to a somewhat higher absolute pressure and the vapor and air mix with the impelling steam on the discharge side of the jet. The total mixture then passes from the ejector into the condenser.

The slight amount of air which may be entrained in the cooled water is removed by a small secondary ejector which raises the pressure sufficiently so that the air can be discharged to the atmosphere. A secondary

condenser is then necessary to condense the steam in the secondary jet.

While a single booster of smaller than 15 tons capacity is difficult to build, steam jet vacuum cooling units have been built for as small as 5 to 6 tons capacity. They can readily be built for steam pressures of from 5 to 200 lb per square inch and condenser water temperatures as high as 90 F. The steam consumption in pounds per hour per ton of refrigeration increases rapidly as the booster steam pressure is lowered. For example, the lowering of the booster steam pressure from 200 to 90 lb per square inch results in an increase in steam consumption of approximately 5 per cent, whereas a further decrease in booster steam pressure to 10 lb per square inch increases the steam consumption by approximately 72 per cent over that required at 200 lb per square inch.

The capacity of a steam jet system is usually controlled by controlling

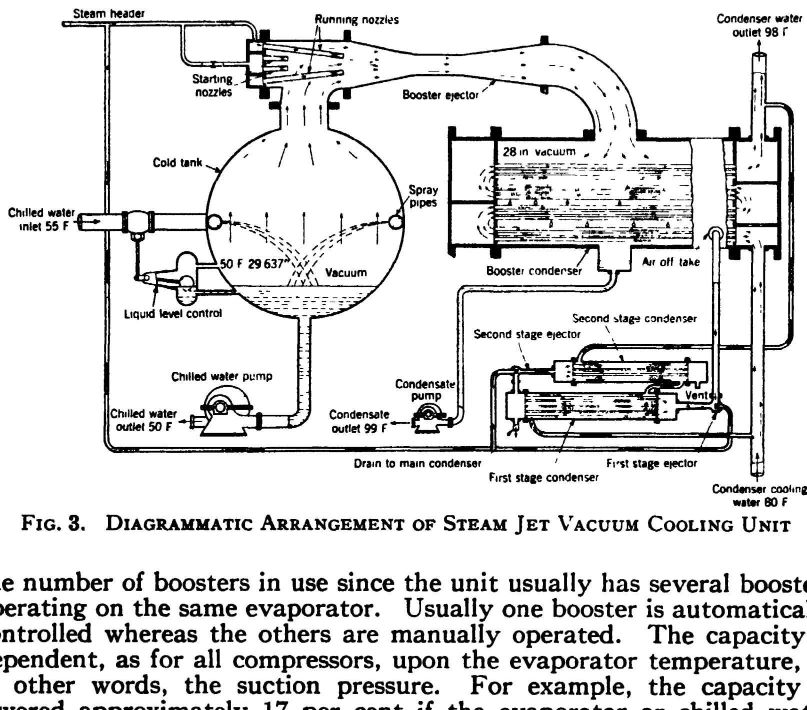

FIG. 3. DIAGRAMMATIC ARRANGEMENT OF STEAM JET VACUUM COOLING UNIT

the number of boosters in use since the unit usually has several boosters operating on the same evaporator. Usually one booster is automatically controlled whereas the others are manually operated. The capacity is dependent, as for all compressors, upon the evaporator temperature, or in other words, the suction pressure. For example, the capacity is lowered approximately 17 per cent if the evaporator or chilled water temperature is lowered from 50 to 45 F. The capacity therefore can be controlled to some extent by regulating the evaporator temperature.

CHARACTERISTICS OF COMPRESSION SYSTEMS

The various types of compression systems have quite different characteristics of capacity and power with varying evaporator and condenser temperatures, as may be noted from curves in Figs. 4 and 5.

From Fig. 5 it may be observed that power requirements for the centrifugal compressor increase much more rapidly than for the reciprocating compressor with increase in evaporator temperature. Similarly, the capacities of the steam ejector and centrifugal compressors increase more rapidly than those of the reciprocating compressor with increase in evapor-

ator temperature. Thus, both the steam jet and centrifugal machines tend to be more self-regulating than the reciprocating. It is also evident from Fig. 5 that the steam jet equipment is best suited for operation at high evaporator temperatures.

The effect of condenser temperature upon the power and capacity of the different types of compressors is shown in Fig. 6. It may be noted that the power required by the reciprocating compressor increases rapidly with increase in condenser temperature, while the power curve for the centrifugal compressor is relatively flat. It is also evident that the

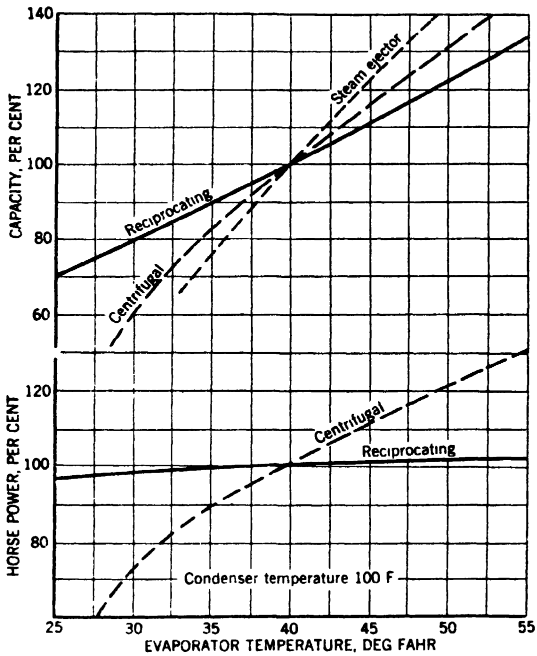

FIG. 4. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

capacity of the steam jet compressor is independent of condenser temperature until a certain point is reached where it drops to zero. As previously stated, steam jet equipment requires more condensing water than other types of compression systems. Consequently, steam jet systems are well suited to those applications where condensing water is cheap, or where condensing water is rather high in temperature.

ABSORPTION SYSTEMS

The fundamental rule governing the absorption (in a closed system) of a gas by a liquid is Raoult's Law, which states that at any given temperature the ratio of the partial pressure of a volatile component in a solution to the vapor pressure of the pure component at the same temperature is equal to its mol fraction in the solution. The mol fraction in turn is equal to the number of mols of substance divided by the total number of mols present. The number of mols in a given weight of a compound is equal to the weight divided by the molecular weight.

This law applies strictly, only to what is known as an ideal solution, that is, one in which the inter-molecular forces between the substances present in the solution are equal. Actually, no such solutions exist, so that deviations from Raoult's Law are always found in practice. The deviation is called positive when the observed pressure is greater than that calculated from Raoult's Law, while the term negative deviation refers to the opposite case. Negative deviations are found wherever chemical attraction exists between the solvent and the solute. Positive deviation occurs when there is a difference in the internal pressure of the components, chemical attraction between them being absent.

In order to make an effective absorption machine, large negative

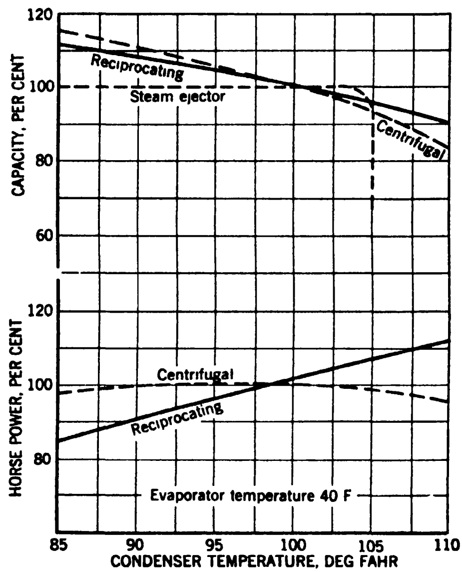

FIG. 5. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

deviations from Raoult's Law must be shown by solutions of the refrigerant in the liquid absorbent, because the larger the negative deviation, the greater is the amount of refrigerant that can be cycled, using a given weight of absorbent. Cycling a large amount of refrigerant for a given weight of absorbent is important because of the heat required to raise the temperature of the mixture and disassociate the refrigerant and the absorbent. Only the latent heat of the refrigerant can be recovered for useful work.

Many refrigerant-absorbent combinations have been proposed and quite a number have been tested. A diagrammatic representation of a typical closed absorption system is outlined in Fig. 6. In this system a mixture of refrigerant and absorbent is evaporated in the generator, passes to an analyzer and rectifier where it is purified, and then to a condenser where the refrigerant and remaining absorbent are condensed. It then passes through an expansion valve to an evaporator, where heat is absorbed from a cooling load. From the evaporator the vapor and residual absorbent passes to an absorber where it meets absorbent which is

initially low (weak) in refrigerant concentration. The absorbent absorbs the vapor and the strong absorbent liquor is transferred to the generator through an interchanger with the weak liquor returning from the generator.

A cooling medium, ordinarily water, is used in the absorber to remove the heat of absorption and maintain the absorptive power of the absorber at a maximum.

Like the steam jet system, the absorption system compares most favorably when a cheap source of cooling water and steam or other heat is available. Unlike the steam jet system, the comparative performance is usually best with a wide range of temperature between the evaporator and absorber, since with a good refrigerant-absorbent combination, the amount of heat and water required for a given refrigerating effect in-

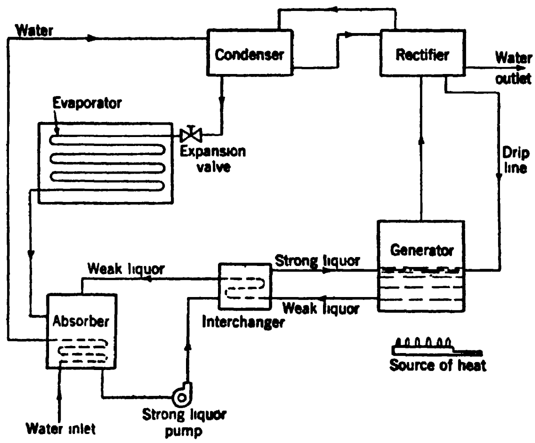

FIG. 6. CLOSED ABSORPTION SYSTEM

creases slowly with an increase of evaporator-condenser temperature range.

At the present time the most used refrigerant-absorbent combinations are: (1) water and ammonia, and (2) dichloromonofluoromethane and dimethyl ether of tetraethylene glycol. With the latter combination the boiling points of the refrigerant and absorbent are sufficiently wide apart that almost pure refrigerant is obtained without the use of a rectifier.

EXPANSION VALVES

The thermostatic expansion valve is a device to regulate the flow of liquid refrigerant so that the evaporator will always be used to best advantage. The evaporator coil must be kept as full as possible without any chance of liquid refrigerant entering the suction line. The expansion valve accomplishes this by regulating the supply of refrigerant, so that the temperature of the gas leaving the evaporator is always slightly higher than the temperature of the boiling refrigerant inside of it. This difference in temperature between the outgoing (suction) gas and the liquid refrigerant in the evaporator is called the superheat of the gas.

The operation of the thermostatic expansion valve can best be explained by means of a diagram, Fig. 7. A small refrigerant charge in the control bulb exerts a pressure through the tube to the upper side of the diaphragm, which tends to open the valve.

The magnitude of this pressure is determined by the temperature of the suction gas leaving the evaporator, as the control bulb is attached to the suction line at this point and is at approximately the same temperature. The suction pressure in the evaporator is transmitted through the equalizer tap and exerts an opposing force on the other side of the diaphragm in the direction to close the valve. This pressure corresponds to the temperature of the boiling refrigerant. The resulting force on the diaphragm is determined by the differential between the temperature of the suction gas and the boiling point of the refrigerant, which is the amount of superheat in the gas. If this temperature differential becomes greater (superheat increases), the resultant force on the diaphragm opens the valve and admits more refrigerant. The reverse is true if the superheat decreases, and the valve partly closes, thus admitting less refrigerant. The spring keeps the valve closed until the resultant force on the diaphragm corresponds to the desired superheat. The adjustment of the spring will change the amount of superheat to be maintained in the suction gas.

The selection of the expansion valve is, of course, determined by the capacity of the valve. The capacity of a valve with a given orifice is

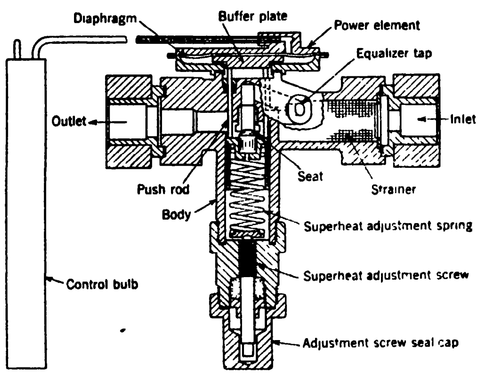

FIG. 7. TYPICAL THERMOSTATIC EXPANSION VALVE

determined by the refrigerant used, the differential of pressure across the valve and the amount the liquid is sub-cooled as it enters the valve. The expansion valves are usually rated at zero sub-cooling of the liquid, or 100 per cent liquid. Frequently special devices are used to properly distribute the refrigerant among the parallel paths of the evaporator. These distributing devices usually have considerable pressure drop. Where they are used, the pressure drop across the expansion valve is not the difference between suction and discharge pressures, as allowance must be made for the pressure drop across the distributing device. An equalizer connection from the evaporator suction line must be made to the underside of the diaphragm (see Fig. 7) whenever the valve outlet is not at the evaporator pressure so as to insure suction pressure at this point. When distributing devices are used, this equalizer connection is essential for proper operation of the valve. Another pressure drop allowance must be made for the liquid line, particularly when the liquid line has an appreciable vertical rise.

CONDENSERS

Condensers used for liquifying the refrigerant are of three general designs: (1) air, (2) water, and (3) evaporative (combination air and water).

Air Cooled

Air cooled condensers are seldom used for capacities above 3 tons of refrigeration, unless an adequate water supply is extremely difficult to obtain, as, for instance, in railway air conditioning. Even on fractional tonnage installations, air is used as the condensing medium only where water is expensive or where simplicity of installation warrants the higher condensing pressure, and consequent higher power costs than would be obtained using water as the condensing medium.

The conventional air cooled condenser consists of an extended surface coil across which air is blown by a fan. The hot discharge gas enters the coil at the top and, as it is condensed, flows to a receiver located below the condenser. Air cooled condensers should always be located in a well ventilated space so that the heated air may escape and be replaced by cooled air.

The principal disadvantages of air cooled condensers are the power required to move the air and the reduction of capacity on hot days. This loss of capacity due to high condensing pressures on hot days requires that equipment of increased capacity be selected to meet the peak load. Thus at normal loads the equipment is oversized.

Water Cooled

Water cooled condensers are of the double pipe type, the shell and tube type, or the shell and coil type. Double pipe condensers are arranged so that water passes through the inner of two concentric pipes and refrigerant circulates through the annular space between the pipes. Where possible, there should be counter-flow of the refrigerant and the condensing water to obtain maximum temperature differences. This type is usually used only with small condensing units.

The amount and temperature of the condensing water determine the condensing temperature and pressure, and indirectly the power required for compression. It is therefore necessary to determine a balance so that the quantity of water insures economical compressor operation.

Because there is a decided tendency to conserve the water in city mains and because most large cities are restricting the use of water for air conditioning and refrigeration equipment, it is often necessary to install cooling towers or evaporative condensers. Cooling towers, unfortunately, produce the warmest condensing water at the time when the load on the system is greatest, so that the refrigeration equipment must be designed to meet the maximum load at abnormal condensing water temperatures. If properly designed, this makes little difference in the efficiency of operation throughout the year except at those times when the condensing water temperature is highest. As this occurs only for 5 per cent of the entire cooling period it can be disregarded as a factor in establishing yearly operating costs.

The cooling tower has a certain advantage over the use of water from the city mains. Economies are possible when a cooling tower is used, which cannot be achieved by the use of condensing water from city mains. In certain localities, the lowest city water temperature met during the summer months is from 65 to 70 F. This temperature range takes place for the entire cooling period, regardless of the outdoor temperature. With a cooling tower, the temperature of the condensing water may rise to 80 or 85 F under maximum conditions, but under less than maximum conditions the temperature of the water leaving the cooling tower drops

considerably. It has been established that in these localities during 50 per cent of the time, the outdoor wet-bulb temperature varies from 60 to 70 F and the cooling tower water, for the same periods, varies from 65 to 75 F. When the outdoor wet-bulb temperature drops below 60 F, which occurs approximately 30 per cent of the time, the condensing water temperature is still lower. The cost of water used for condensing is small as the only water required is that used to make up the loss by evaporation in the cooling tower itself. Refer to the section on Cooling Towers in Chapter 26.

Shell and coil condensers are in general use for medium sized condensing units, and consist of a coil of tubing mounted inside a shell. The cooling water passes through the coil.

Evaporative Condensers

Due to the high cost of city water for condenser purposes, and due to ordinances in some localities prohibiting the discharge of large quantities of such water into the sewage systems, there has been developed a condenser which uses a minimum amount of water on a finned surface, cooling it to approximately the wet-bulb temperature of the surrounding atmosphere.

The end view of a typical evaporative condenser is shown in Fig. 8. The fan draws the air over a finned tube condenser which is kept wet by a water spray. The discharge refrigerant gas from the compressor enters the top of the condenser coil and the liquid refrigerant is drained from the bottom of the coil into a liquid receiver and then circulates through the remaining portion of the system in the usual way.

The water is circulated through the spray nozzles and the level is maintained in the sump by means of a float valve. The eliminator plates are placed in the path of the water-air mixture so as to remove the entrained water. The air leaving the unit is almost completely saturated, so that care must be taken in locating discharge ducts to prevent condensation.

Evaporative condensers are available in sizes up to 100 tons or more. These units use only a small portion of the water required for a water cooled condenser. The water is vaporized by the heat of the refrigerant so that each pound of water used extracts approximately 1000 Btu from the refrigerant, whereas under standard rating conditions where the water temperature rise is 20 F, each pound of water extracts only 20 Btu from the refrigerant. Including the water lost by entrainment in the discharge air, by overflow and stand-by evaporation, the water used is about 3 to 5 per cent of the amount that would be required for a water cooled condenser.

The evaporative condenser requires more maintenance, occupies greater space (must be located where air is available), and has a higher first cost than the water cooled condenser, but where the use of water is restricted or expensive, the evaporative condenser has become widely accepted. Compared with a water cooled condenser and cooling tower, which combination uses about the same quantity of water, the evaporative condenser has the advantage of lower cost and smaller space requirements.

EVAPORATORS AND COOLERS

The types of coolers used in connection with air conditioning work fall into three general groups. The *first*, is the direct cooling of water; the

second, direct cooling of air; and the *third*, cooling of brine for circulation in a closed system, which can cool either water or air. One method of the direct cooling of water is to install direct expansion coils in the spray chamber so that the water sprayed into the air comes in direct contact with the cooling coils. Another common and efficient method of cooling spray water is to use a Baudelot type of heat absorber where the water flows over direct expansion coils at a rate sufficiently high to give efficient heat transfer from water to refrigerant.

Another type of spray water cooler is the shell and tube heat exchanger in which the refrigerant is expanded into a shell enclosing the tubes

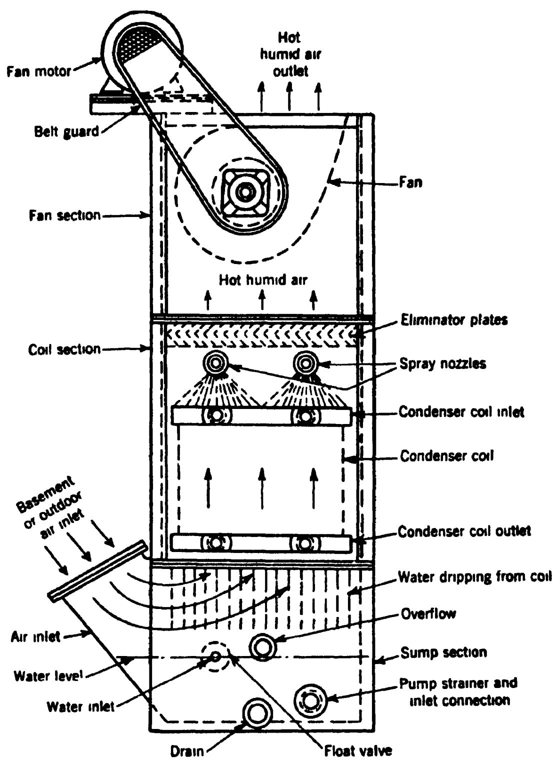

FIG. 8. SCHEMATIC VIEW OF AN EVAPORATIVE CONDENSER

through which the water flows. The velocity of the water in the tubes affects the rate of heat transfer, and as the refrigerant is in the shell completely surrounding the tubes at all times, good contact and a high rate of heat transfer are insured. The disadvantage of such a system is that with the falling off of load on the compressor the suction temperature or the temperature in the evaporator drops and there is a possibility of freezing the water in the tubes, which, of course, might split the tubes and allow the refrigerant to escape into the water passage. This danger can be eliminated by automatic safety devices.

Another system of cooling spray water is to submerge coils in the spray collecting tank, or in a separate tank used for storage. The heat transmission through the walls of the coils, however, is low and a great deal

more surface is required than for any other type of cooler. However, with large storage tanks this type of cooling can be utilized to advantage.

When direct cooling of air is employed, the refrigerant is inside the coil and the air passes over it. Cooling depends upon convection and conduction for removing the heat from the air. The type of coil used can be either smooth or finned, the finned coil being more economical in space

TABLE 7. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE DISCHARGE OR HOT GAS LINES[a]

Capacity Btu per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[b]									
	Line Sizes, Inches									
	5/8	3/4	7/8	1 1/8	1 3/8	1 5/8	2 1/8	2 5/8	3 1/8	3 5/8
10,000	2.3	1.0	0.6							
15,000	4.9	2.0	1.0							
20,000	8.5	3.4	1.7	0 6						
25,000		5.3	2.6	0.9						
30,000		7.5	3.6	1.2	0.5					
40,000			6.4	2.1	0.7					
50,000			9.8	3.1	1.0	0 5				
60,000				4.4	1 3	0 7				
70,000				6 0	1.9	0 9				
80,000				8.0	2.5	1.1				
90,000				10 2	3 1	1.4				
100,000					3.8	1.7	0 5			
125,000					6.0	2 6	0 7			
150,000					8.5	3 8	1.0			
175,000					11 6	5.1	1.3			
200,000						6 7	1.7	0.6		
250,000						10 4	2.6	0 9		
300,000							3.7	1.2	0.5	
400,000							6.7	2.2	0.9	
500,000							10 5	3.5	1 5	0 7
600,000								5 0	2.1	1.0
800,000								9 0	3.8	1 8
1,000,000									5 8	2.9
1,250,000									9 5	4.4
1,500,000										6.4
2,000,000										11 3

[a]Soft annealed copper tubing up to and including 3/4 in. outside diameter. Hard copper pipe 7/8 in. outside diameter and larger.

[b]Length of tubing includes the average number of fittings.

requirement than the smooth coil. The fins, however, must be far enough apart so as not to retain the moisture which condenses out of the air.

The indirect cooler, where brine is cooled by the refrigerant and the resulting cold brine is used to cool either air or water, introduces several other considerations. It is not the most economical from a power consumption standpoint, as it is necessary to cool the brine to a temperature sufficiently low so that there is an appreciable difference between the average brine temperature and that of the substance being cooled. This requires that the temperature of the refrigerant must be still lower, and consequently the amount of power required to produce a given amount of refrigeration increases due to the higher compression ratio. There are other considerations which make such a system desirable. In the first place, where a toxic refrigerant is undesirable or cannot be used because of fire or other risks, especially in densely populated areas, the brine can be cooled in an isolated room or building and can then be circulated through the air conditioning equipment. This arrangement eliminates any possibility of direct contact between the air and refrigerant.

REFRIGERANT PIPE SIZES

The selection of proper pipe sizes and frictional pressure losses varies with the installation and the capacity of the system. Generally the suction piping should be selected so that the pressure loss is between 2 and 3 lb per square inch. The pressure drop in liquid lines should be maintained so as to permit no vaporization in the pipes with limiting

TABLE 8. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE LIQUID REFRIGERANT LINES

Capacity Btu per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a]			
	Pipe Sizes, Inches			
	7/8	1 1/8	1 3/8	1 5/8
100,000	0.6			
125,000	0 9			
150,000	1 3			
175,000	1 8			
200,000	2 3	0 6		
225,000	2 9	0 8		
250,000	3 6	1 0		
275,000	4 3	1 2		
300,000	5 1	1 4		
325,000	5 9	1 6		
350,000	6 9	1 8		
375,000	7 9	2 1		
400,000	9 0	2 3	0.8	
450,000		2.9	1 0	
500,000		3 5	1 3	
550,000		4 3	1 5	0 7
600,000		5 0	1 8	0 8
700,000		6 7	2 4	1 1
800,000		8.7	3 1	1 4
900,000			3.9	1 7
1,000,000			4 7	2 1
1,200,000			6 7	3 0
1,400,000			9 0	4 0
1,600,000				5 1
1,800,000				6 3
2,000,000				7 9
2,200,000				9 2

[a]Length of tubing includes the average number of fittings.

pressure drops not to exceed 5 lb per square inch. Hot or discharge gas lines should be limited to approximately 4 lb per square inch pressure drop. All pressure drops mentioned are total system losses and include not only the piping losses, but also the pressure losses in the valves, fittings and coils.

Pressure drops for discharge or hot gas lines may be determined from Table 7. Pressure losses in liquid refrigerant lines of various sizes and capacities are given in Table 8. Pressure drops of suction refrigerant pipe lines at varying capacities and refrigerant temperatures are given in Table 9. Oil circulating with the refrigerant appreciably increases the pressure losses in both suction and discharge lines from that given in these tables. All tables are for 100 ft of pipe, including an average number of fittings, and for other lengths the losses are proportionate. Losses through control and regulating valves must be added to the other pipe losses to determine the total drop. All copper pipe referred to in these tables is of type L wall thickness and is designated by outside diameter.

The effect of the sizes of refrigerant lines on the system may be studied

Table 9. Pressure Losses in Dichlorodifluoromethane Suction Refrigerant Lines

Copper Pipe Actual O.D. Inches	Capacity Btu per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a]						
		Refrigerant Temperature Deg F						
		−10	0	10	20	30	40	50
¾	2,000	0.3	0.3	0.2	0.2	0.2	0.1	0.1
	4,000	1.3	1.0	0.8	0.7	0.6	0.5	0.4
	6,000	2.8	2.2	1.8	1.5	1.2	1.0	0.9
	8,000	4.8	3.8	3.1	2.6	2.1	1.8	1.5
	10,000	7.4	5.8	4.8	3.9	3.3	2.8	2.3
	12,000	10.5	8.4	6.8	5.6	4.7	4.0	3.3
	14,000	14.0	11.0	9.1	7.6	6.4	5.4	4.5
	16,000		14.5	12.0	9.8	8.3	7.0	5.8
	18,000			15.0	12.3	10.4	8.7	7.2
	20,000				15.0	12.7	10.7	8.9
1⅛	7,000	0.4	0.3	0.3	0.2	0.2	0.2	0.1
	10,000	1.0	0.7	0.5	0.5	0.4	0.3	0.3
	15,000	1.9	1.5	1.2	1.0	0.8	0.7	0.6
	20,000	3.3	2.6	2.1	1.7	1.4	1.2	1.0
	25,000	5.0	4.0	3.2	2.7	2.2	1.9	1.6
	35,000	9.7	7.7	6.2	5.1	4.3	3.6	3.0
	45,000	15.8	12.6	10.0	8.4	7.0	5.9	4.9
	60,000				14.8	12.2	10.2	8.6
	70,000						14.0	11.7
1⅜	10,000	0.3	0.2	0.2	0.2	0.1	0.1	0.1
	15,000	0.7	0.5	0.4	0.3	0.3	0.2	0.2
	20,000	1.2	0.9	0.7	0.6	0.5	0.4	0.4
	30,000	2.6	2.1	1.6	1.3	1.1	0.9	0.8
	40,000	4.6	3.6	2.8	2.3	1.9	1.6	1.4
	50,000	7.0	5.5	4.4	3.5	2.9	2.5	2.1
	60,000	10.0	7.8	6.2	5.0	4.2	3.5	3.0
	80,000		14.0	11.0	8.7	7.3	6.2	5.2
	100,000				13.5	11.3	9.5	8.2
1⅝	30,000	1.6	1.3	1.0	0.8	0.7	0.6	0.5
	40,000	2.7	2.1	1.7	1.4	1.1	0.9	0.8
	50,000	4.2	3.2	2.5	2.1	1.7	1.4	1.2
	60,000	6.1	4.5	3.6	2.9	2.4	2.0	1.7
	70,000	8.7	6.3	4.8	3.8	3.1	2.6	2.2
	80,000		8.4	6.3	4.9	4.0	3.3	2.8
	90,000			8.0	6.2	4.9	4.1	3.5
	100,000			10.0	7.6	6.1	5.0	4.2
	120,000					8.6	7.0	5.9
	140,000						9.5	7.9
2⅛	50,000	0.7	0.5	0.4	0.3	0.3	0.2	0.2
	100,000	2.6	1.8	1.4	1.1	0.9	0.8	0.7
	150,000	5.6	3.9	3.0	2.4	2.0	1.6	1.4
	200,000	9.8	6.7	5.2	4.1	3.4	2.8	2.4
	250,000	14.8	10.3	8.0	6.3	5.1	4.2	3.6
	300,000		14.5	11.3	9.0	7.2	6.0	5.0
	350,000		19.5	15.3	12.0	9.7	7.8	6.7
	400,000			19.6	15.3	12.5	10.0	8.5

[a]Length of tubing includes the average number of fittings.

by referring to the preceding discussion on Characteristics of Compression Systems. It will be noted that any lowering of the suction pressure at the compressor lowers the capacity. Therefore, excessive pressure drop through the suction piping should be avoided. On the other hand, the suction line must not be made too large when using refrigerants which are soluble in oil, because under such circumstances the velocity of the returning refrigerant may become too low to carry back the entrained oil. Pressure drop in the discharge line also lowers the capacity of the system but not to the same extent as does the pressure drop in the suction line. The velocities of the refrigerant in either suction or discharge lines must not be excessive or noise will result. Velocities of 1000 to 2000 fpm are common in suction lines, and from 2000 to 3500 fpm are used in discharge

Table 9. Pressure Losses in Dichlorodifluoromethane Suction Refrigerant Lines (Concluded)

Copper Pipe Actual O.D. Inches	Capacity Btu per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a] — Refrigerant Temperature Deg F						
		−10	0	10	20	30	40	50
2⅝	50,000	0 2	0 2	0 1	0 1	0 1	0 1	0 1
	100,000	0 7	0 6	0 5	0 4	0 3	0 2	0 2
	150,000	1 6	1 2	1.0	0 8	0 6	0 5	0 4
	200,000	2.8	2 1	1 7	1 4	1 1	0 9	0 7
	250,000	4.3	3 4	2 6	2 1	1 7	1 3	1 1
	300,000	6 1	4 5	3.7	3 0	2 4	1 9	1 5
	350,000	8 2	6 0	5 0	4 0	3 2	2 5	2 0
	400,000		7 8	6 5	5 1	4 2	3 3	2.7
	450,000			7 7	6 4	5 3	4 0	3 5
	500,000				7 8	6 4	5 0	4.2
	550,000					7 7	6 2	5 1
	600,000						7.4	6.2
3⅛	200,000	1.2	1 0	0 8	0 6	0 5	0 4	0 4
	300,000	2.6	2 0	1 6	1 3	1 0	0 8	0 7
	400,000	4 5	3 4	2 6	2 1	1 7	1 4	1 3
	500,000	7 3	5 4	4 1	3 3	2 7	2 2	1 9
	600,000		8 1	6 0	4 7	3 8	3 1	2 7
	700,000			8 4	6 5	5 2	4 2	3 5
	800,000				8.6	6 8	5 5	4 6
	900,000					8 7	7 0	5 9
	1,000,000						8 9	7.3
3⅝	300,000	1 2	0 9	0 7	0 6	0 5	0 4	0 3
	400,000	2 0	1 6	1 3	1 0	0 8	0 7	0 6
	500,000	3 2	2 5	1 9	1.6	1 3	1 0	0 9
	600,000	4 6	3 6	2.8	2 2	1.8	1 5	1 3
	700 000	6.4	4.9	3 8	3 0	2 5	2 0	1 7
	800,000	8 7	6 4	4 9	3 9	3 2	2 5	2 2
	900 000		8 2	6 2	4 9	3.9	3 2	2.7
	1,000,000			7.7	6 1	4 9	4 0	3.3
	1,100 000			9 4	7 3	5 8	4 8	4 0
	1,200 000				8 7	6 9	5 6	4 8
	1,300,000					8 0	6 6	5 6
	1,400,000					9 3	7 6	6 4
4⅛	400 000	1 0	0 8	0 6	0 4	0 4	0 3	0 3
	600 000	2 4	1 8	1 4	1 1	0 9	0 7	0 6
	800,000	4 1	3 1	2 4	2 0	1 6	1 3	1 1
	1,000,000	6 6	4 8	3 7	3 0	2 5	2 0	1 6
	1,200,000	10 0	7 1	5 4	4 4	3 5	2 9	2.4
	1,400,000		10 0	7 5	5 9	4 8	3 9	3 3
	1,600,000			10 0	7 7	6 2	5 1	4 2
	1,800 000				10 0	7 9	6 4	5 3
	2,000,000					9 7	7 9	6 6
	2,200,000						9 5	7 9

[a]Length of tubing includes the average number of fittings.

lines. Velocities in the discharge lines as high as 5000 fpm can only be used where the fittings and bends are all streamlined as noise will otherwise result.

The pressure drop in the liquid line affects the capacity of the expansion valve as the pressure drop across the valve is reduced by the amount of the pipe line drop. If the liquid line drop is sufficient to cause flashing (*i.e.* vaporizing) of some of the liquid refrigerant, a hissing noise in the lines and valves usually develops.

ICE SYSTEMS

Cold water systems using ice as the cooling agent have been installed

in many theaters, restaurants, funeral homes, churches and other places where short hours of operation and high peaks of cooling demand make this type of system desirable. A comparatively small quantity of ice in the water cooling tank of such a system can release refrigeration at a relatively rapid rate. For instance, neighborhood theaters having a peak demand of 1,200,000 Btu per hour (100 tons refrigeration) have found 8 ton capacity ice bunkers satisfactory.

In operation, the water in the air conditioning system is circulated over ice placed in an insulated box and is cooled to the 38 or 40 F range or higher if desired. This cold water is pumped from the ice bunker to air cooling coils or spray type air washers. The blowers, coils, air washer or air handling sections are the same as those parts in any system employing cold water as a refrigerant.

The ice water cooler or ice bunker is usually built at the installation in a location where it can easily be iced. It can be constructed of any desired material such as concrete, steel, or wood with an adequate amount of insulation to save the ice from one period of use to the next. The basic

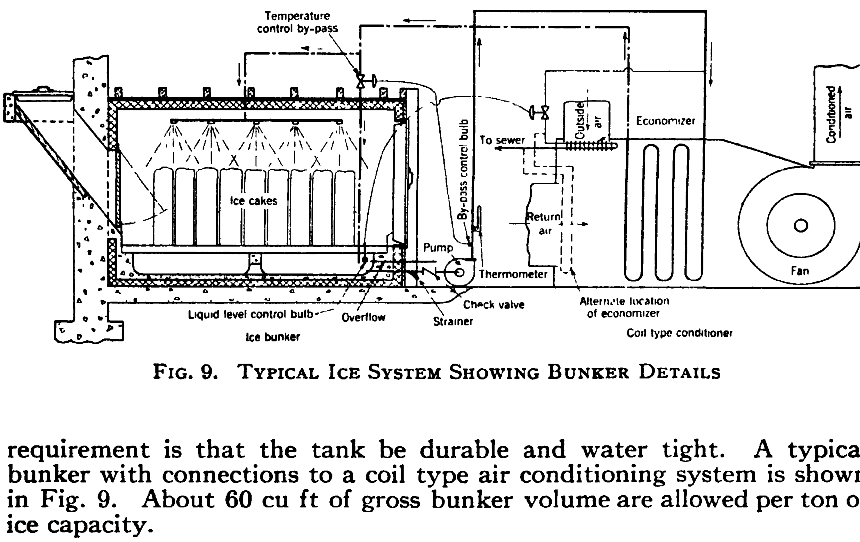

FIG. 9. TYPICAL ICE SYSTEM SHOWING BUNKER DETAILS

requirement is that the tank be durable and water tight. A typical bunker with connections to a coil type air conditioning system is shown in Fig. 9. About 60 cu ft of gross bunker volume are allowed per ton of ice capacity.

The shape of the bunker usually conforms to the available space. The one illustrated has overhead sprays, but if head-room is lacking the ice is placed on the floor of the bunker with the water returned around the lower part of the blocks from a perforated distribution pipe run along one side of the bunker. To secure good circulation the supply water is extracted from a similar perforated pipe on the opposite side of the bunker.

The temperature of the water is controlled at a predetermined point by a thermostat in the supply line. If the temperature drops too low, a part of the return water is by-passed directly to the sump and is not cooled over the ice. In the larger systems it is customary to install an overflow control which, as the ice melts, discards the excess water through an economizer coil. The surface of the economizer is large in relation to the flow so that the water is warmed to 60 F or more as it is discharged from the system.

STORAGE SYSTEMS

In an attempt to lower initial equipment cost and operating expense, or increase the refrigeration capacity of an existing air conditioning system, storage refrigeration has been utilized in a few applications. Some of the methods which have been adopted include the storage of refrigeration in the form of chilled water, chilled brine, ice on evaporator coils[2] and the accumulation of thin sheets of ice on copper plates in a steel tank[3]. If the peak load factor is low as compared with a long period of operation, such as in a restaurant, or if the hours of operation are short but the usage factor high as in a church, then it is possible to consider storage refrigeration. This method of accumulating refrigeration frequently makes it possible to use low cost off-peak electric power. Power costs may also be reduced by installing a smaller refrigeration plant, augmented by a storage system, and by operating it for longer periods.

TABLE 10. BASIS OF EQUIPMENT SELECTION

CAPACITY TONS	MAJORITY USED	SOME USED	FEW USED
0 to 5	Unit systems in conditioned space.	Unit central systems using duct distribution.	Built up central systems.
5 to 25	Built up central systems using reciprocating compressors.	Unit central systems using duct distribution.	Unit systems in conditioned space. Built up systems using absorption and adsorption systems.
25 to 50	Built up central systems using reciprocating compressors.	Built up central systems using centrifugal compressors.	Central systems using adsorption systems.
50 to 400	Built up central systems using reciprocating compressors	Built up central systems using steam jet and centrifugal compressors.	
400 and Over	Built up central systems using centrifugal compressors.	Built up central systems using steam jet.	

EQUIPMENT SELECTION

The selection of proper refrigeration equipment for any air conditioning job is of utmost importance for satisfactory results. The most important factors in the selection of the equipment are:

1. Loads (as determined by the conditions of the space to be cooled).
2. Economics (both initial and operating costs).
3. Codes (local safety codes must be adhered to and influence the type of system to be used).

A broad division of equipment to be used for a particular installation or

[2]The Application of Storage Refrigeration to Air Conditioning, by C. F. Boester (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 675).

[3]Use of Cold Accumulators in the Air Conditioning Field, by R. W. Evans and C. J. Otterholm (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 123).

application may be made on the basis of the magnitude of the load. Current general practice is outlined in Table 10.

Unit or *packaged* systems, consisting of a reciprocating compressor, condenser, evaporator and fans, are generally used in the smaller sized jobs where electric power is available, as they are manufactured complete, ready to install and are the most economical (see Chapter 22).

The reciprocating compressor in the built-up central system (see Chapter 20) covers the widest range of application since it is applicable to either the direct expansion or indirect systems and can be driven by steam or gas engines, or by electric motors. The quantity of condensing cooling medium required is also less than for any other system with the exception of the centrifugal compressor, which uses the same amount.

Centrifugal compressors are used for large installations, and usually where the indirect system is required. The driving mechanism can be

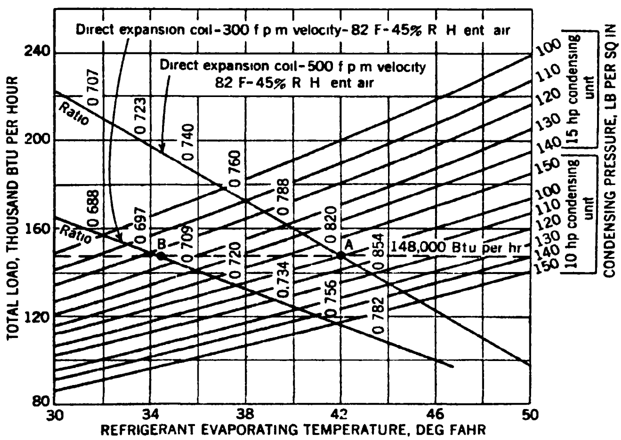

FIG. 10. COMPRESSOR AND COIL PERFORMANCE

steam turbine or electric motor. The steam jet system is used where steam is available and cooling water can be had in large quantities.

It will be noted by referring to Fig. 4 that all systems using compressors have a common characteristic and that is, that the capacity varies with the evaporating temperature. Not only can the equipment be selected to produce a given result but the performance can be predicted under varying load conditions by the simple expedient of using the variable of evaporating temperature as the abscissa and the load or capacity as the ordinate in a series of curves.

Manufacturers of compressors and cooling coils furnish performance data for apparatus that can be plotted in the form of curves similar to those shown in Fig. 10. The performance of a compressor is plotted as a series of curves, each curve being drawn for a given condensing pressure. The performance of a direct expansion coil at two different air velocities is plotted on the same graph. The operating point will be, of course, where the two curves cross.

Data given in Table 11 illustrate two types of conditioned enclosures having the same total load of 148,000 Btu per hour, but with two different

ratios of sensible to total heat. In the case of the office with a ratio of 82 per cent sensible to total heat, the operating point *A* in Fig. 10 is found to be 42.2 F evaporating temperature with a face velocity of 500 fpm. In the case of the restaurant, with a ratio of 69.5 per cent sensible to total heat, the air velocity is lowered to 300 fpm and the evaporating temperature is lowered to 34.4 F as shown in point *B* of Fig. 10. In order to obtain the same capacity, a larger condensing unit is used. This illustration assumes zero pressure drop through the suction line. The pressure drop can be taken into account by shifting the compressor performance curves by the amount of pressure drop expressed in degrees Fahrenheit.

THE REVERSE CYCLE

In heating by the reverse refrigeration cycle energy is absorbed in an evaporator from some available source of heat, pumped to a higher temperature and delivered to a condenser[4]. The heat from the condenser is used for heating purposes. The compressor acts as a heat pump whose fundamental function is to raise the potential of the heat. The theoretical

TABLE 11. TYPICAL OPERATING CONDITIONS FOR TWO TYPES OF LOAD

Type of Enclosure	Load, Btu per Hour			Ratio Sensible to Total	Air Entering Coil		Operating Balance Point		
	Sensible	Latent	Total		Deg F	Per Cent R H.	Evaporator Temp Deg F	Condenser Pressure Lb per Sq In.	Per Cent Sensible Heat
Restaurant	103,000	45,000	148,000	0 695	82	45	34.4	123	69.9
Office	121,000	27,000	148,000	0.820	82	45	42 2	100	82.1

ratio of the heat delivered to the work of compression is given in Equation 1.

$$\frac{T_2}{T_2 - T_1} \tag{1}$$

where

T_1 = absolute temperature of evaporator.
T_2 = absolute temperature of condenser.

Thus, with a small spread of temperature between the evaporator and the condenser, 6 or 8 times as much heat may be obtained theoretically, and 3 to 5 times practically, as the work introduced. There are a number of limitations, however, the most serious of which is the lack of ready availability of a practical source of heat.

1. Well water is the most desirable since its temperature is higher than other sources even in the winter, and thus a large amount of heat may be removed in relation to the weight of water handled.

[4]Cooling Homes, A Field for Refrigeration, by A. R. Stevenson, presented at the symposium of the Refrigeration with Gas Committee of the *American Gas Association*, April 20, 1926. The Heat Pump, An Economical Method of Producing Low-grade Heat from Electricity, by T G N. Haldane (*Electric Review*, Vol. 105, p 1161-1162, December 27, 1929, and *I. E. E. Journal*, Vol 68, p 666-675, June, 1930). Edison Building Heated and Cooled by Electricity, by H L Doolittle (*Power*, Vol. 74, p. 384, September 8, 1931). House Heating by Pump with 5 to 1 Pick-up Ratio, by Gilbert Wilkes and R E Marbury (*Electrical World*, Vol. 100, p 828, December 17, 1932) An All Electric Heating, Cooling and Air Conditioning System, by Philip Sporn and D. W. McLenegan (A S H.V.E. TRANSACTIONS, Vol 41, 1935, p. 307). Using the Reversed Cycle Refrigerating Principle for a Self-Contained Heating and Cooling Unit, by Henry L. Galson (A.S H V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1935, p. 497). Heating by Reversed Refrigeration, by A. J. Lawless (*Heating, Piping and Air Conditioning*, August, p. 473, September, p. 519, 1940).

2. Air may be used but its specific heat is low and its temperature uncertain. When the most heat is needed, the temperature of the air is lowest, thus resulting in the least favorable temperature combination.

3. It has been proposed to obtain heat by freezing water but this is still in the experimental stage.

Some of the other factors which act as limitations are: the large temperature spread when using air as a source of heat and when attempting to cool with even moderately low outside temperatures, the frequent disparity between the size of the cooling load and heating load requiring extra equipment for a complete heating load, and the relatively high initial cost of equipment as compared to that at present available for heating by conventional means.

Because of these limitations, the present application of the system is largely limited to temperate climates, such as Florida and Southern

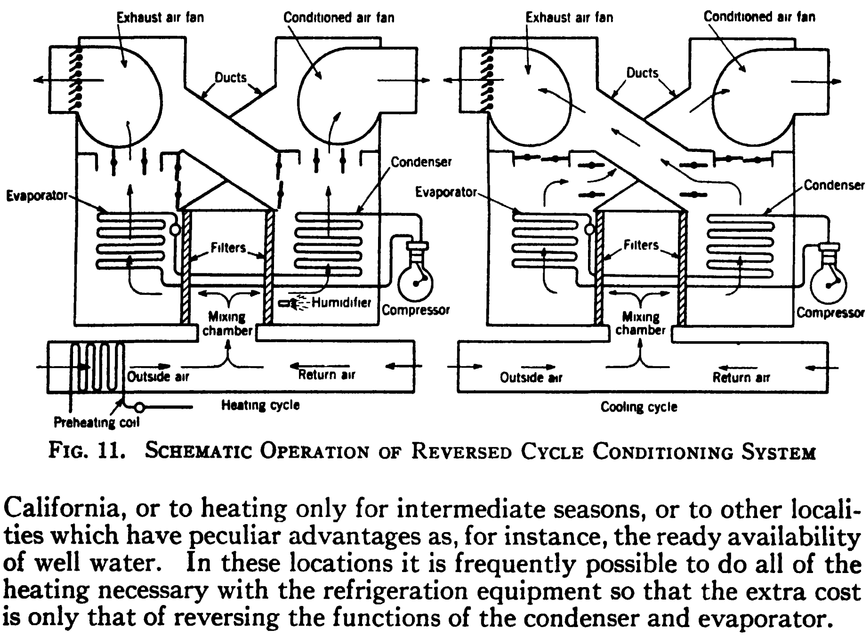

FIG. 11. SCHEMATIC OPERATION OF REVERSED CYCLE CONDITIONING SYSTEM

California, or to heating only for intermediate seasons, or to other localities which have peculiar advantages as, for instance, the ready availability of well water. In these locations it is frequently possible to do all of the heating necessary with the refrigeration equipment so that the extra cost is only that of reversing the functions of the condenser and evaporator.

There are a number of reversed systems now in operation, particularly among utility companies, using well water as the source of heat. These systems range in size up to 320 hp. In the case of the largest system in operation at present, the cost of the electrical energy would have to be approximately 0.7 cents per kilowatthour in order to compete with oil at 6 cents per gallon.

A typical arrangement of a reversed cycle conditioning system where air is used as a source of heat is shown in Fig. 11. If the air seldom drops below freezing, heat is often required in the morning and cooling during the afternoon in order to maintain comfortable conditions in such a system. The arrangement as shown lends itself to automatically changing over as required.

CHAPTER 25

Heat Transfer Surface Coils

Coil Applications, Construction and Arrangement, Steam Coils, Water Coils, Direct-Expansion Coils, Flow Arrangements, Applications, Calculation of Heat Transfer, Air Flow Resistance, Coil Performance, Selection

THE coils described in this chapter are used in air conditioning systems for heating or cooling an air stream under forced convection. The surface coil equipment may be made up of a number of banks assembled in the field, or the entire assembly may be factory constructed. The applications of each type of coil are limited to the field within which it is rated. Other limitations are imposed by code regulations, by proper choice of materials for the refrigerants used and the condition of the air handled, or by an economic analysis of the possible alternates on each installation.

For heating service, these coils are used as preheaters, reheaters or booster heaters, (see Chapters 20 and 21). The function of the coils is air heating only, but the apparatus assembly may include means for humidification and air cleaning. Steam or hot water are the usual heating media, although others are used in special cases, such as reheating by means of discharge gas from a refrigerating system.

Coils are used for air cooling with or without accompanying dehumidification. Examples of cooling applications without dehumidification are precooling coils using well water or other relatively high temperature water to reduce the load on the refrigerating machinery, or water cooled coils to remove sensible heat in connection with chemical moisture-absorption apparatus. By proper coil selection it is possible to handle both sensible cooling and dehumidification together as further explained later. The apparatus assembly usually includes an air cleaning means to protect the coil from accumulation of dirt and to keep dust and foreign matter out of the conditioned space. Although cooling and dehumidification are the usual functions, there are cases of cooling coils purposely wetted as an aid to air cleaning and odor absorption.

The usual cooling media used in surface coils are cold water or Group I (*ASA* Classification) refrigerants, but others are used in special cases. Brines are seldom required for the range of applications covered by this chapter, although there are cases where low entering air temperatures with large latent heat loads require a refrigerant temperature so low that water becomes impractical. Sometimes, also, brine from an industrial system already installed is the only convenient source of refrigeration.

For combined cooling and dehumidifying, surface coils present an alternate to spray dehumidifiers. For many applications it is possible, by proper selection of apparatus, choice of air velocities, refrigerant temperatures, etc., to perform the same duty with either. In a few cases both sprays and coils are used. The coils may then be installed within the spray chamber, either in series with the sprays or below them. In making the selection between spray and surface dehumidifiers, certain advantages of each should be considered. The fact that a spray dehumidifier is usually designed to deliver nearly saturated air tends to simplify the control problem. In this case the dry-bulb temperature is also the dew-point, and hence a dew-point control can be arranged by using a simple duct thermostat. Spray dehumidifiers have the advantage over

unwetted coils of a certain degree of air cleaning and odor absorption. On the other hand, coils make possible a closed and balanced cooling water circuit, obviating the unbalanced pumping head, the complication of water level control, and danger from possible floods incidental to multiple-spray dehumidifiers, especially if located on different levels. The use of coils often makes it possible for the same surface to serve for summer cooling and winter heating by circulating cold water in the one season and hot water in the other, with consequent saving in apparatus and piping. Another advantage is that where the surface coil system can be used with direct expansion of refrigerant, it is comparatively low in initial and operating costs. Of course the safety of the occupant must be kept in mind in comfort conditioning applications. Some localities have refrigeration codes which restrict the use of direct-expansion coils in the air stream, and hence local codes should be consulted by the engineer before a system employing direct expansion methods is designed. The choice between spray dehumidifiers and coils depends upon the necessities and the economic aspects of each case and no general rule can be given. There are many installations in which either can be used.

COIL CONSTRUCTION AND ARRANGEMENT

Coils are basically of two types, those consisting of bare tubes or pipe and those of *extended* surface construction. The former are little used for the applications covered by this chapter, but are often employed where conditions cause frost accumulation, and for cooling surface within spray dehumidifiers.

The heat transmission from air passing over a tube to a refrigerant flowing within it is impeded by three resistances. The same is true when the air is being heated by steam or hot water in the tube. The first resistance is from the air to the surface of the tube, usually called the outside surface resistance or air-film resistance. Second is the resistance to the flow of heat by conduction through the metal itself. Finally there is another surface or film resistance to the flow of heat between the inside surface of the metal and the fluid in the tube. For the applications under consideration both the resistance of the metal wall to heat conduction, and the inside surface or film resistance are usually low as compared with the air-side surface resistance. Economy in space, weight and cost makes it advantageous to decrease the external surface resistance, where it is proportionately large, to approach that of the tube wall, and that from tube to refrigerant. This is accomplished by increasing the external surface by means of fins. Sometimes water spray is applied to the same type surface as would have been used without it. The overall heat transfer is not necessarily increased much by such an arrangement, but the water spray may serve other purposes than to increase the flow of heat, such as air and coil cleaning.

In fin or *extended* surface coils the external surface of the tubes is known as *primary* and the fin surface is called *secondary*. The primary surface consists generally of round tubes or pipes. In some cases these are staggered and in others in line with respect to the air flow. The staggered arrangement gives a somewhat higher heat transfer value but also a higher resistance to air flow and in some cases makes the header and return bend arrangement more complicated. A number of types of fin arrangement are used, the most common of which are spiral, flat and flat-crinkled or corrugated, all as shown in Fig. 1. While the spiral fin surrounds each tube individually in all cases, the flat types may be con-

tinuous (including several rows of tubes), or they may be round or square, with individual fins for each tube. All of these, as well as other less common types, are in use, the selection for a particular installation being based on economic considerations, space requirements and resistances of individual designs of coils. A most important factor in the performance of extended surface coils is the bond between the fin and the tube. An intimate contact is assured in a number of ways. The assembled coil may be coated with tin, zinc, etc., after fabrication. The spiral type fin may be knurled into a shallow groove on the exterior of the tube. The tube may be expanded after the fins are assembled, or the tube hole flanges of a flat or corrugated fin may be made to override those in the preceding fin and so compress them upon the tube. There are also types of construction where the fin is formed out of the material of the tube itself. In any case the successful performance of a fin surface depends upon the bond between fin and tube being secure and remaining so in service.

For heating coils the materials most generally used are copper, steel and aluminum. Sometimes aluminum or brass fins are used on copper tubes.

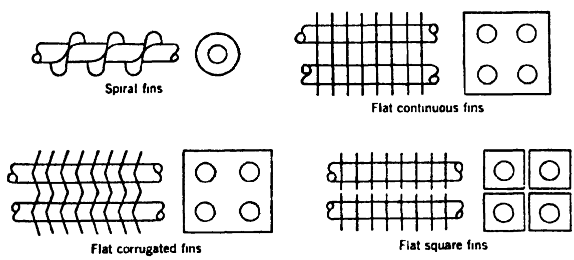

FIG. 1. TYPES OF FIN COIL ARRANGEMENT

Steel is uncommon except in special cases. Some types of heating coils are made of cast-iron. There are sufficient practical installations of each of these to demonstrate that they can all give good service. However for equal performances brass and aluminum fins must be of greater thickness than copper fins on account of their lower coefficients of conduction. The copper coils are frequently tin-dipped and steel coils galvanized to protect them from corrosion and to assure a bond between fin and tube.

Cooling coils for water or for volatile refrigerants are most frequently of copper, both fin and tube. Aluminum fins on copper tubes are also used. For brines such as sodium or calcium chloride and for ammonia, steel fins and tubes are common.

Although there are many variations for special cases, tube and fin sizes and spacings for air conditioning coils, both heating and cooling, fall within fairly narrow limits. The tubes are usually ⅜, ½, ⅝, or ¾ in. OD, and the fins spaced from 4 to 8 per inch, 7 per inch being a common design. The tube spacing generally varies from about 1⅛ to 2 in. on centers. Small tube size and close fin spacing give large capacity with small space demand, but the resistance, both over the surface and through the tubes, is higher than with larger tubes and more widely spaced fins. Moreover, too close a fin spacing may result in trouble from dirt accumulation, especially on dehumidifying coils, and may also cause trouble from water *hold-up* between the fins, particularly with air flow vertically upward. This condition increases the air resistance and de-

creases the capacity of the coil. Water hold-up sometimes causes flooding trouble in vertical air flow units by accumulating too much water for the drain to handle all at once when the fan is stopped.

Steam Coils

For proper performance of steam heating coils, condensate and air must be continually eliminated and the steam must be evenly distributed to the individual tubes. This distribution is usually accomplished by individual orifices in the tubes, by distributing plates and orifice in the steam header, or by perforated internal steam-distributing pipes extending into the individual tubes. The latter arrangement has the advantage of distributing the steam throughout the length of each tube, and is conducive to uniform delivered air temperatures. The tendency for freezing of condensate at the bottom of the coil with cold entering air and light

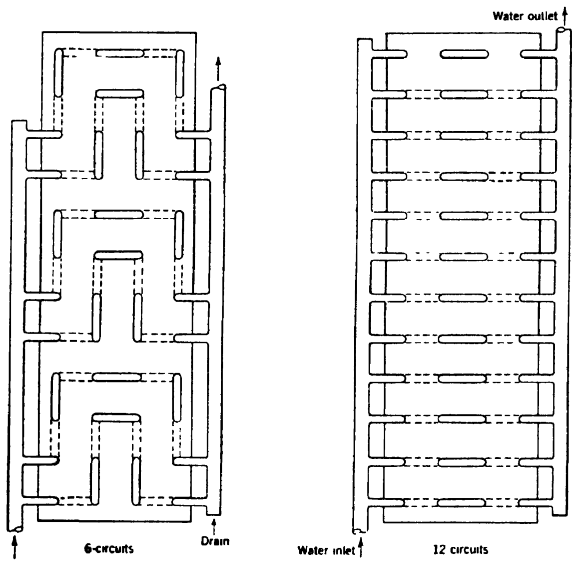

Fig. 2. Various Water Circuit Arrangements

heating loads is also minimized. This is especially valuable for outside air preheaters. Methods of air and condensate elimination are discussed in detail in Chapters 14 and 21.

Water Coils

The performance of water coils, for heating or cooling, depends on the elimination of air from the system and proper distribution of water. Air elimination is taken care of in the system piping as described in Chapter 15. To assure a pressure drop sufficient for adequate distribution but at the same time to provide against excessive pumping head where large water quantities are handled, water coils are provided with various water circuit arrangements. For instance, a typical coil 18 tubes high and 6 tubes deep in the direction of air flow can be arranged for 6, 9, 18 or 36 parallel water circuits as conditions may require. Orifices in individual tubes are occasionally employed but are usually unnecessary as the resistance of individual water circuits is generally sufficient to effect a

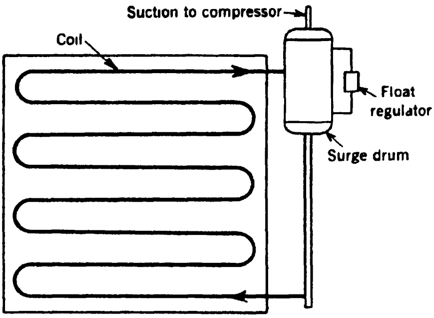

FIG. 3. DIRECT-EXPANSION COIL WITH FLOODED SYSTEM

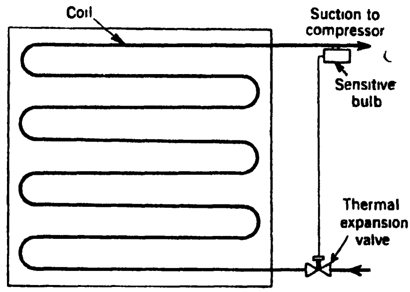

FIG. 4. DIRECT-EXPANSION COIL WITH THERMAL VALVE SYSTEM

satisfactory distribution. In cases such as well water precooling coils, where there may be considerable sand and other foreign matter in the water, provision for cleaning of individual tubes is of advantage. It is important to arrange water coils for drainage if located where they will be exposed to freezing. For this reason the circuits should be so laid out that there are no pockets to hold water. Fig. 2 shows such construction. The drains may be provided in the water piping although they are often arranged in the coil headers.

Direct-Expansion Coils

Coils for volatile refrigerants present more complex problems of fluid distribution than do water, brine or steam. It is desirable that the coil be effectively and uniformly cooled throughout, and necessary that the compressor be protected from entrained, unevaporated refrigerant. There are two types; namely, flooded systems, and thermal expansion valve systems, as shown in Figs. 3 and 4. In a flooded coil, the circulation is similar to that in a water tube boiler. The liquid is maintained at the proper level by the action of a float regulator as shown in Fig. 3. The thermal expansion valve system depends upon the thermal valve automatically feeding just as much liquid to the coils as is required to maintain the superheat at the coil suction outlet within predetermined limits

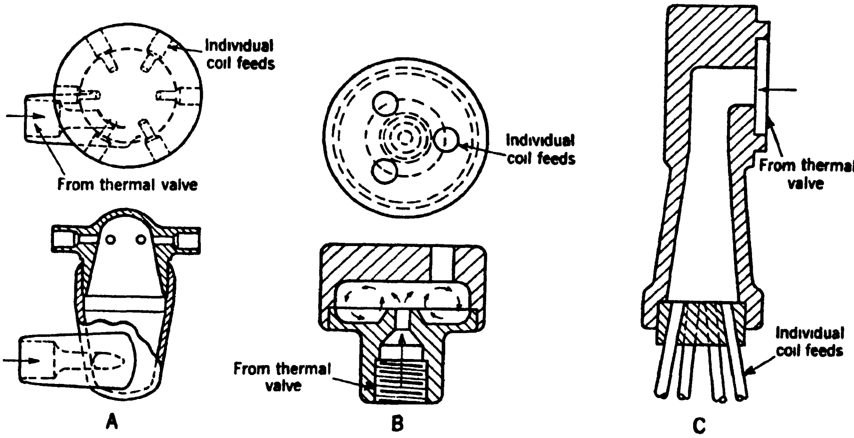

FIG. 5. TYPES OF REFRIGERANT FEED DISTRIBUTING HEADS

which vary from about 6 to 10 F. The thermal valve arrangement is in common use for the type of coils covered by this chapter, while the flooded system is comparatively rare.

With the flooded system the refrigerant distribution through the tubes depends on properly selecting the length of the feeds and the head of liquid imposed upon the liquid inlets. No auxiliary distributing devices are required. With the thermal valve system there are two factors to consider. There must be, generally, more than one refrigerant feed through the coil per thermal valve to keep the pressure drop through the refrigerant circuit within practical limits and to reduce the corresponding

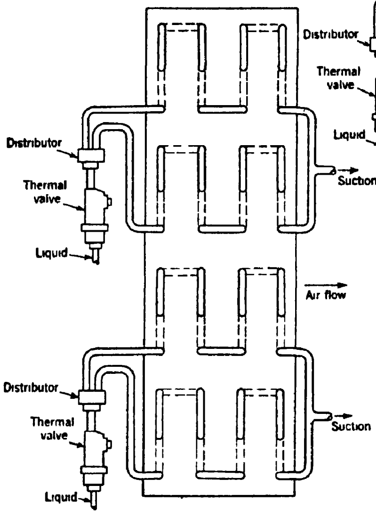

Fig. 6. Arrangement for Face Control

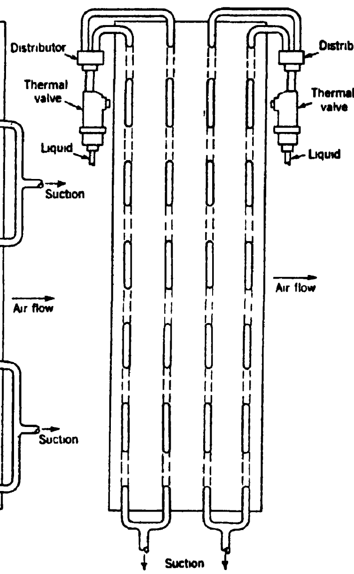

Fig. 7. Arrangement for Depth Control

penalty in increased evaporating temperature. At the same time the coil must be so arranged that the required suction superheat can be attained with a minimum sacrifice in the performance of the coil as a whole. It is general practice to attain this superheat within the coil itself and not by the use of external heat exchangers or other auxiliary devices.

With thermal expansion valves it is advantageous to keep the pressure drop through the refrigerant feeds as low as possible. The feeds are laid out to expose each to the same mean temperature difference so that it handles the same refrigerating load. A distributing means is imposed between valve and coil liquid inlets to divide the refrigerant equally among the feeds. Such a distributor must be effective for distributing both liquid and vapor, because the entering refrigerant is a mixture of the two. Fig. 5 shows three typical types of distributors. In distributor *A* the liquid and gas mixture from the thermal valve is led tangentially into a chamber. The coil feed connections extend outward radially at the top of this chamber. In distributor *B* the refrigerant is discharged at a high

velocity through a central jet against the end plate, forming a uniform mixture of gas and liquid within the distributor, from which individual connections are led as shown. In type *C* the refrigerant enters at high velocity from the thermal valve and is discharged against the end plug in which the individual liquid feeds are closely arranged. These distributors can be used in either vertical or horizontal position. Although there are other forms of distributors those mentioned are typical examples. The individual liquid connections from the distributor to the coil inlet are commonly made of small diameter tubing and are all of the same length and diameter in order to impose the same friction between the distributor and the coil. Since the thermal valves act in response to the superheat at the coil outlet, this superheat should be produced with the least possible sacrifice of active evaporating surface. Sometimes a single thermal

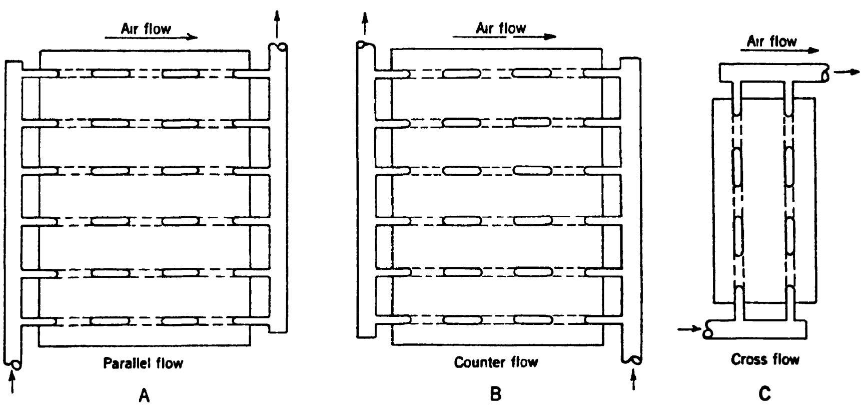

FIG. 8. FLOW OF MEDIA IN TUBES IN RELATION TO AIR FLOW

valve is used per coil. In other cases multiple valves are used, with the coil divided across the air flow or parallel to the air flow as shown in Fig. 6. The arrangement of Fig. 7 should be avoided since it offers the disadvantage of unequal load on the two parallel circuits.

Flow Arrangement

The relative direction of flow of the air outside the tubes and the medium within them influence the performance of the surface. There are three types of relative flow in common use. Fig. 8A shows parallel-flow in which the air and the medium in the tubes proceed through the coil in the same direction. Fig. 8B shows counter-flow in which the medium in the tubes proceeds in a direction opposite to the flow of air. Fig. 8C shows cross-flow in which the air and the medium in the tubes pass at right angle to each other. Parallel flow is seldom used with water-coils for the reason that a lesser mean temperature difference results than with counter-flow. The counter-flow arrangement is almost universally used in brine or water coils to take advantage of the highest possible mean temperature difference for given entering water and air temperatures. It is also invariably used in coils fed with volatile refrigerant to take advantage of the higher air temperature for superheating the leaving gas. This arrangement assists complete evaporation and superheating of the refrigerant which is essential to proper operation of the thermal expansion valve. Cross-flow is common in steam heating coils, the temperature

within the tubes being substantially uniform and the mean temperature difference the same whatever the direction of flow, relative to the air. Cross-flow is to be avoided in coils with volatile refrigerants on account of unequal loading of parallel circuits and danger of short circuiting of liquid refrigerant which will disturb proper functioning of the thermal expansion valve.

Applications

Heating coils in field assembled banks are used for a number of purposes as described in Chapter 20. They may be arranged with the air flow vertical or horizontal, although the latter is more common. For steam heating, the coils may be set with the tubes vertical or horizontal. In the latter case the coil should be sloped to provide for condensate drainage. Because of the multi-circuit feed arrangement, and the necessity for avoiding air and water pockets, water heating coils are generally arranged with the tubes horizontal. Certain precautions must be taken

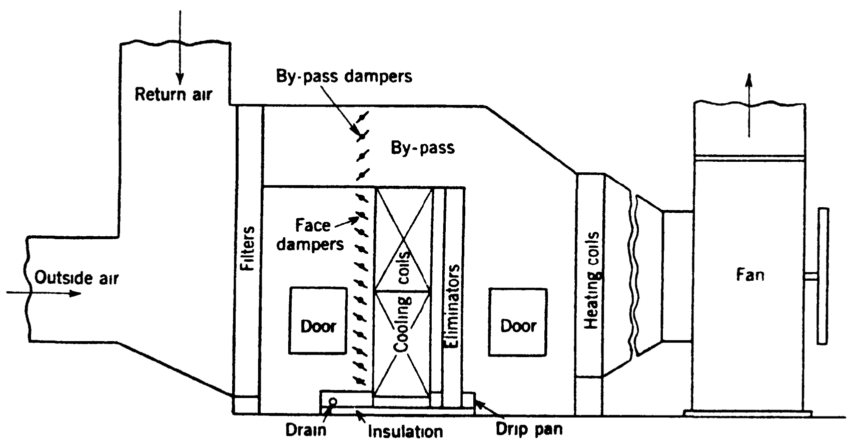

FIG. 9. TYPICAL ARRANGEMENT OF COOLING COILS IN A CENTRAL SYSTEM

against freezing. Where steam coils are used with entering air below freezing temperature, throttling the steam supply may result in freezing the condensate in the bottom of the coil if the tubes are of the variety not provided with internal distributing pipes, or an equivalent arrangement. If these are used, there is little danger of freezing the condensate as long as the leaving air temperature is not allowed to fall below about 40 F. As an added precaution with both steam and water coils the outside air inlet dampers are often closed automatically when the fan is stopped to avoid trouble caused by very cold outside air drifting in during *off* periods.

A typical arrangement of cooling coils is shown in Fig. 9. Some means should be provided to filter all the entering air to keep dirt and foreign matter from accumulating on the coils. The assembly is provided with a drip-pan to catch the condensate during summer dehumidifying duty and to collect the non-evaporated water from the humidifying sprays in winter. The drip connection should be made ample in size and liberally provided with plugged tees and crosses for cleaning. It should not be

exposed to freezing temperatures in winter if the apparatus is used on winter humidifying duty. Access doors should be provided for servicing filters, humidifying nozzles, and fan bearings and for cleaning the coils. With certain designs of coils when used for dehumidifying, eliminators must be used beyond the coil to catch any water which may be blown into the air stream. It is customary to include these eliminators when the air velocity exceeds about 450 fpm with the individual fins and about 600 fpm for the continuous flat fin type. Where a number of coil sections are stacked one upon another, and where the velocities are low, so that eliminators need not be used, occasional trouble results when water splashes down from one coil to the next and blows out into the air stream. In such cases drip troughs as shown in Fig. 10 are used to collect this water and conduct it to the condensate pan.

Sometimes finned surface coils on summer cooling and dehumidifying duty are provided with water sprays. These sprays are of two types.

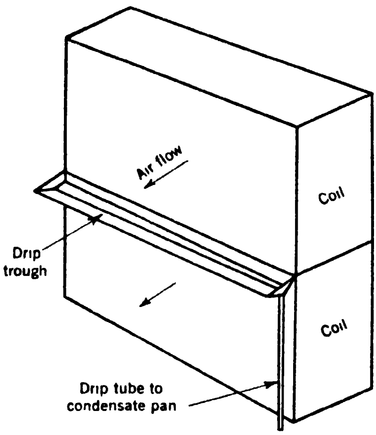

Fig. 10. Coil Arranged with Drip Trough

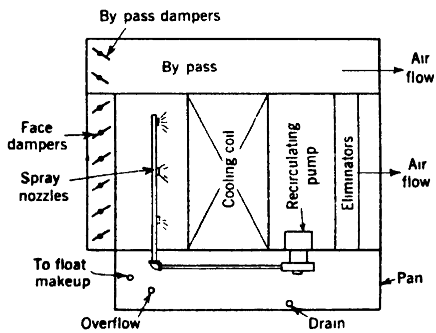

Fig. 11. Recirculating Spray System for Cleaning Coils

In the first type a set of spray nozzles is arranged for intermittent cleaning. The operator can wash the coils off as frequently as necessary. These sprays are not operative when the system is in use and no recirculating pump is provided. The second arrangement requires a collecting tank and a recirculating pump. The water is in circulation whenever the apparatus is in operation, and assists in keeping the coil clean and in absorbing odors. Fig. 11 illustrates such an arrangement. Wherever air by-passes are used around a coil on summer duty for control purposes, it is of advantage to direct only return air through the by-pass rather than a mixture of return and outside air. The casing should be arranged accordingly. To maintain the air quantity handled by the fan reasonably constant, and to assure the required design quantity of by-passed air when the by-pass damper is open, cooling coil banks are frequently furnished with both face and by-pass dampers as shown in Fig. 9.

Although both heating and cooling coils are made of sufficient strength to take up expansion and contraction arising within themselves, care should be taken to avoid imposing strains from the piping on to the coil connections. (See Chapter 14.)

HEAT TRANSFER AND AIR FLOW RESISTANCE

The transfer of heat between the heating or cooling medium and the air stream is influenced by several variables:

1. The temperature difference.
2. The design and surface arrangement of the coil.
3. The velocity and character of the air stream.
4. The velocity and character of the medium in the tubes.

The driving force is usually taken as the logarithmic mean temperature difference for heating or cooling without dehumidification. For combined cooling and dehumidification, a special measure of the propelling force is used as described later. Logarithmic differences are generally employed in practice although there are special flow relationships used, such as cross-flow, where they do not strictly apply. With volatile refrigerants there is often an appreciable pressure drop and corresponding change in evaporating temperature through the refrigerant circuit. The problem is further complicated by the fact that the refrigerant is evaporating in part of the circuit and superheating in the remainder. In spite of this, heat transfers and ratings for coils using volatile refrigerants are usually based in practice on a refrigerant temperature corresponding to the average pressure in the coil.

The design and surface arrangement of the coil includes such items as materials, type, thickness, height and spacing of the fins, and the ratio of this surface to that of the tube, the use of the staggered or in-line tube arrangement, and provisions to increase the air turbulence such as the use of corrugated as against flat fins. Staggered tubes increase the total heat transfer as against the in-line arrangement and corrugated fins are more effective than flat. Of especial importance is the bond between fin and tube.

The velocity of the air usually considered is the coil face velocity. This bears a varied relation to the actual velocity over the surface, depending upon the individual coil design. As long as a fixed design of coil is under consideration face velocities may be used, but they may be unsatisfactory in comparing different designs, as it is the actual surface velocity that is significant. The air volume is often based on standard air at 70 F and a barometric pressure of 29.92 in. Hg. The use of air volume in coil rating information may be misleading. The significant value is mass velocity in pounds per minute and not cubic feet per minute, because for a fixed volume the corresponding weight may vary widely, depending upon the temperature and barometric pressure under consideration.

At the same mass air velocity, varying performance can be obtained depending upon the turbulence of the air flow into the coil and upon the uniformity of distribution of air over the coil face. The latter is very important in obtaining reliable test ratings and in realizing rated performance in practical installations. The resistance through the coils will assist in properly distributing the air, but where the inlet duct connections are brought in at sharp angles to the coil face, the effect is frequently bad and there may even be reverse air currents through the coils. This reduces the capacity, but can be avoided by proper layout or by the use of directing baffles.

The heat transfer depends also upon the velocity of the medium in the tubes and upon its character, whether flowing water, condensing steam or evaporating volatile refrigerant. Heat transfer rates expressed as Btu

per square foot of internal surface per degree logarithmic mean effective temperature difference between the fluid and tube wall are, for example, about 150 to 300 for evaporating dichlorodifluoromethane, about 350 to 1200 for water at 2 and 6 fps and about 1200 for condensing steam. The influence of the medium in the tubes on the overall heat transfer rate is, therefore, apparent.

Because of these variables, reliable rating and performance information for any design of coil must be based on actual tests on that coil under the expected conditions of operation. A comparison between the performance of two designs, unless based on such tests on each, may lead to entirely erroneous conclusions.

PERFORMANCE OF HEATING AND COOLING COILS

Heating and cooling coils are heat exchangers and as such their performance depends in general upon:

1. The overall coefficient of heat transfer from the fluid within the coil to the air it heats or cools.
2. The mean temperature difference between the fluid within the coil and the air flowing over the coil.
3. The physical dimensions of the coil.

Thus, for any one definite operating condition, the heating or cooling capacity of a given coil is expressed by the following basic formula:

$$q = U \times MTD \times A \quad (1)$$

where

q = total heat transferred by the coil, Btu per (hour) (square foot of coil face area).

U = overall coefficient of heat transfer, Btu per (hour) (square foot of external coil surface) (degree Fahrenheit temperature difference between the fluid within the coil and the air flowing over the coil)

MTD = mean temperature difference, degrees Fahrenheit between the fluid within the coil and the air passing over it. (This is commonly taken as the logarithmic mean temperature difference.)

A = external surface area of the given coil, square feet per square foot of coil face area.

The performances of heating and cooling coils are influenced by the same factors in all but one very important exception, that is, when cooling coils operate wet or act as dehumidifying coils. For this reason, in the later discussion, heating and dry cooling coils are treated as one group and dehumidifying coils as another.

OVER-ALL COEFFICIENT OF HEAT TRANSFER

Of all factors affecting the performance of heating or cooling coils, the overall coefficient of heat transfer is the most difficult to determine as it is influenced by several factors which depend upon coil design and conditions of operation.

Considering any coil, whether of bare pipe or of finned type, the overall heat transfer coefficient for a given size and design of coil can always be considered as a combined effect of three individual heat transfer coefficients, namely:

1. The film coefficient of heat transfer between air and the external surface of the coil, usually given in Btu per (hour) (square foot external surface) (degree Fahrenheit mean temperature difference).
2. The coefficient of heat transfer through the coil material—tube wall, fins, ribs, etc.

3. The film coefficient of heat transfer between the internal surface of the coil and the fluid flowing within the coil, usually given in Btu per (hour) (square foot internal surface) (degree Fahrenheit mean temperature difference).

These three individual coefficients acting in series result in an overall coefficient of heat transfer in accordance with the basic laws. For a *bare pipe* coil the overall coefficient of heat transfer, whether for heating or for cooling (dry), can be expressed by a simplified basic formula as follows:

$$U \doteq \frac{1}{\frac{R}{h_r} + \frac{L}{k} + \frac{1}{h_a}} \tag{2}$$

where

U = overall coefficient of heat transfer, Btu per (hour) (square foot external surface) (degree Fahrenheit mean temperature difference between air and fluid within the coil).

h_r = film coefficient of heat transfer between the internal surface of the coil and the fluid flowing within the coil, Btu per (hour) (square foot internal surface) (degree Fahrenheit mean temperature difference between that surface and the average fluid temperature).

h_a = film coefficient of heat transfer between air and the external surface of the coil, Btu per (hour) (square foot external surface) (degree Fahrenheit mean temperature difference between the mass of air and the external surface).

k = conductivity of material from which the bare pipe is constructed, Btu per (hour) (square foot) (degree Fahrenheit per inch thickness).

L = thickness of tube wall, inches.

R = ratio between external and internal surface of the bare tube, usually varying from 1.03 to 1.15 for the tube used in typical heating or cooling coils. This ratio R is inserted in the formula in order to place internal fluid coefficient of heat transfer on the basis of external surface.

Frequently, when pipe or tube walls are thin and of material having high conductivity (as is the case in construction of typical heating and cooling coils) the term L/k in Equation 2 becomes negligible and is generally disregarded. (The effect of the term L/k in typical bare pipe heating or cooling coils seldom exceeds 1 to 2 per cent of the overall coefficient). Thus, in its simplest form, for bare pipe:

$$U = \frac{1}{\frac{R}{h_r} + \frac{1}{h_a}} \tag{3}$$

For finned coils the formula[1] for the overall coefficient of heat transfer can be conveniently written:

$$U = \frac{1}{\frac{R}{h_r} + \frac{1}{\eta_f h_a}} \tag{4}$$

in which the term η_f, called the *fin efficiency*, is introduced to allow for the resistance to heat flow encountered in the fins.

The term R, in this case, is the ratio of *total* external surface to internal surface. For typical designs of finned coils for heating or cooling, this ratio varies from 10 to 30. Term R is again introduced to place the internal surface coefficient of heat transfer on a basis of external surface. In the discussions which follow, coefficients h_r and $\eta_f h_a$ will be considered separately, and also various ways of combining them will be outlined.

External Film Coefficient

While formulae have been developed expressing the film coefficient h_a for air passing parallel to a plane surface, they cannot be used directly

[1]Rational Development and Rating of Extended Air Cooling Surface, by H. B. Pownall (*Refrigerating Engineering*, October, 1935, p. 211).

for fins on tubes because of air turbulence and because of the temperature gradient prevalent from the edge of a fin to its center. It is therefore necessary to make tests to evaluate the combined term $\eta_t h_a$. The term, $\eta_t h_a$, will be written merely h_a in this discussion as there is no necessity for separately evaluating η and because values of h_a are usually applied only to the particular coils for which tests are made.

Transfer of heat from a fluid to a solid is accomplished by the contacting of the molecules of the fluid with the solid. When a molecule strikes a solid, its energy level equalizes with the energy level of the solid. The total amount of heat exchanged between the molecules of a fluid and a solid is determined by the number of contacts per unit of surface per unit of time, and by the energy change of the fluid.[2] The energy change, in the case of air, is measured by the temperature change times the specific heat of the air. The number of contacts is measured by a percentage of the weight of air flowing per unit of time.

In the case where water vapor is mixed with air, and the water vapor is cooled but not condensed, the amount of heat transferred is increased by the energy change of the vapor particles. The additional energy is measured by the temperature change, by the specific heat of the water vapor, and by the weight of vapor contacting the surface per unit of time. In a mixture of air and vapor there is a definite ratio between the weight of the vapor and of the air per cubic foot of the mixture. Therefore, as the temperature of the mixture is lowered, the amount of heat lost by the vapor always bears a definite ratio to the amount of heat lost by the air. The amount of energy involved in the temperature change of the vapor is small, however, and it is usually included with that of the air by using a value of 0.245 for the specific heat of humid air.

Dehumidification of air by a cooling coil occurs whenever the surface temperature of any part of the coil is below the dew-point temperature of the air. Enough molecules of water vapor are condensed on the coil to create a state of equilibrium between the vapor pressure of the moisture on the coil surface and the vapor pressure of the moisture in that part of the air stream which is in immediate contact with the coil surface. Because of the good contact between the condensed film of water and the coil surface, the water film attains a temperature approaching that of the coil surface. Therefore, those particles of air which actually contact the water film leave the film with a dew-point temperature equal to the outer surface film temperature. However, many air particles, with their attendant water vapor particles, never contact the coil surface, but are by-passed between the fins. These air particles have the same dew-point temperature when they leave the coil as they had when they entered, but after leaving the coil they mix with the air particles which did contact the surface, producing a mixture of air which has a dew-point temperature that lies between the original dew-point temperature and the film surface temperature. This process explains why air seldom leaves a coil in a saturated condition.

The foregoing contact-mixture concept of heat transfer has been found by several independent investigators to be consistent with experimental data. The concept has been used successfully in analyzing the performance of evaporative condensers, cooling towers, condensers and evaporators. A relation[3] has been found between heat transfer and

[2]Graphical Method of Determining Finned Coil Capacities Described, by E. P. Wells (*Heating, Piping and Air Conditioning*, December, 1936, p. 665).

[3]The Contact-Mixture Analogy Applied to Heat Transfer with Mixtures of Air and Water Vapor, by W. H. Carrier (*A.S.M.E Transactions*, January, 1937, Vol. 59, No. 1, p. 49).

pressure drop of flowing fluids, by assuming that molecules of a fluid lose their momentum upon contact with a solid.

The fact that a coil starts to condense moisture when the surface temperature drops below the dew-point temperature of the entering air makes it possible to measure the surface temperature of a coil, an otherwise practically impossible task. After the surface temperature has been determined, it is possible to analyze completely the surface film coefficient of both the air side and refrigerant side of a coil.

The air side coefficient, h_a, of a dry coil of particular dimensions is an exponential function of the mass velocity of the air:

$$h_a = Z\,G^n \tag{5}$$

where

h_a = film coefficient of heat transfer, Btu per (hour) (square foot external surface) (degree Fahrenheit mean temperature difference between air and average surface temperature).

G = air mass velocity, pounds per (hour) (square foot of coil face area).

Z and n = constants which depend upon both air turbulence and surface arrangement.

The difficulty of obtaining sufficient tests to evaluate the constants Z and n for all conditions of coil design and operation makes it desirable to use Equation 6 for determining the air side coefficient:

$$h_a = 0.245 \times \frac{G}{A} \times 2.3 \times \log_{10}\left(\frac{1}{1-\eta_c}\right) \tag{6}$$

where

0.245 = specific heat of humid air, Btu per (pound) (degree Fahrenheit).

2.3 = the constant which converts logarithms from base e to base 10.

A = external surface area, square feet per square foot of coil face area.

η_c = coil efficiency, a decimal less than 1.0.

This formula gives values of h_a after tests have been made to evaluate the coil efficiency. Equation 6 can be derived[4] by combining the basic equations of heat transfer, mean temperature difference and coil efficiency:

$$q_s = h_a \times A \times MTD_a \tag{7}$$

$$MTD_a = \frac{t_1 - t_2}{2\,3 \log_{10}\left(\frac{t_1 - t_s}{t_2 - t_s}\right)} \tag{8}$$

$$\eta_c = \frac{t_1 - t_2}{t_1 - t_s} \text{ (by definition)} \tag{9}$$

$$q_s = 0.245 \times w \times (t_1 - t_2) \tag{10}$$

where

q_s = sensible heat transferred, Btu per hour per square foot of coil face area.

t_1 = temperature of air entering coil, degrees Fahrenheit.

t_2 = temperature of air leaving coil, degrees Fahrenheit.

t_s = average temperature of coil external surface, degrees Fahrenheit.

MTD_a = logarithmic mean temperature difference between air and coil surface.

Coil Efficiency

One method of expressing air-coil contact efficiency is the ratio between the weight of air that actually contacts the coil surface and the total weight of air passing through the coil. Due to the fact that the specific heat of air is fairly constant over a wide range of temperature, coil

[4]Loc. Cit. Note 2.

efficiency[5] can be expressed as equal to the number of degrees that the entire amount of air is cooled, divided by the degrees difference between the entering air temperature and the coil surface temperature.

For a particular heat transfer surface, coil efficiency is only a function of the mass velocity of the air, which may be observed by equating Formulae 5 and 6 and combining all constants into K and u:

$$\log_{10}\left(\frac{1}{1-\eta_c}\right) = \frac{K}{G^u} \qquad (11)$$

This equation can be used in graphical form by plotting coil efficiency

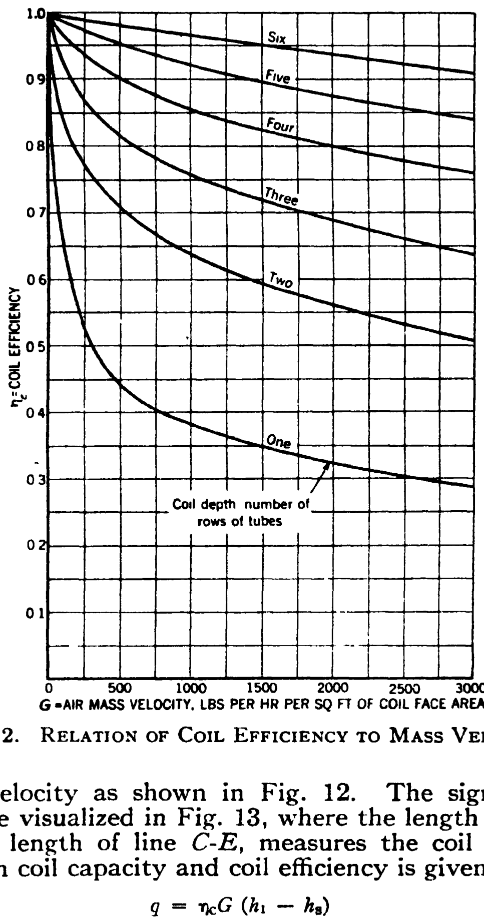

FIG. 12. RELATION OF COIL EFFICIENCY TO MASS VELOCITY

against mass velocity as shown in Fig. 12. The significance of coil efficiency can be visualized in Fig. 13, where the length of the line C-D, divided by the length of line C-E, measures the coil efficiency. The relation between coil capacity and coil efficiency is given by:

$$q = \eta_c G\,(h_1 - h_s) \qquad (12)$$

where

h_1 = specific enthalpy of air entering coil, Btu per pound.
h_s = specific enthalpy of saturated air at surface temperature, Btu per pound.

When no latent heat is being removed from air, the change in enthalpy

[5]When coil efficiency is used herein it is intended to express air-coil contact efficiency and does not express total performance efficiency.

is equal to the change in temperature times the specific heat, so Equation 12 can be changed to:

$$q = \eta_c G\,(t_1 - t_s)\,0.245 \qquad (13)$$

Dehumidification of Air

When moisture is being condensed on the coil surface Equation 12 can be used. If a coil has an efficiency of 0.8 (80 per cent) for the removal of sensible heat, it will at the same time remove 80 per cent of the difference in moisture content between the entering air and saturated air at the surface temperature. This is due to the fact that 80 per cent of the air particles contact the surface and attain a dew-point temperature equal to the surface temperature. This condition is expressed graphically in Fig. 13.

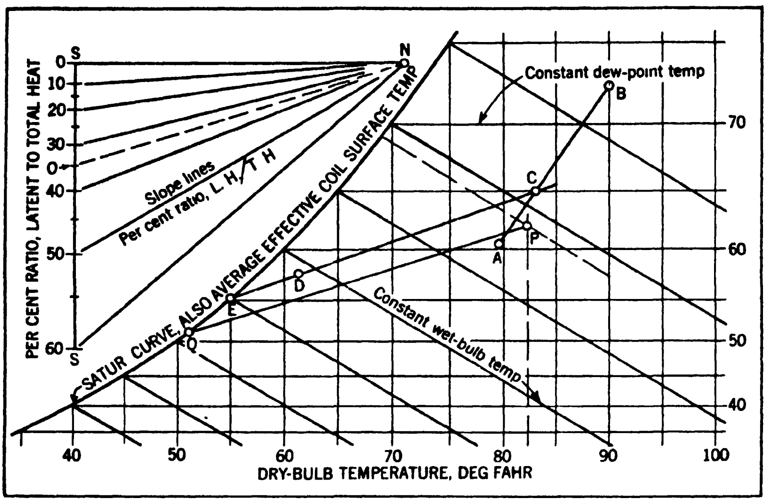

Fig. 13. Psychrometric Chart Showing Straight-Line Method for Representing Coil Performance

This psychrometric chart is constructed so that equal increments along the horizontal axis represent equal changes in sensible heat content, and equal increments along the vertical axis represent equal changes in latent heat content of air. Point A represents the condition of return or recirculated air, point B that of outside air, point C the mixture of two-thirds recirculated air and one-third outside air, and point E the average surface temperature. Point D, which represents the air leaving the coil, lies on a line which connects points C and E, and its distance from point C is equal to the length of the line C-E times the coil efficiency. The ratio between the vertical distance from C to D and the horizontal distance from C to D, expressed in heat units, is the ratio between latent heat and sensible heat removed. It can be shown by trigonometric relations that the slope of the line C-D is a measure of the ratio of latent to total heat removed, and that any line parallel to C-D gives the same heat ratio.

To enhance the practical usefulness of the psychrometric chart illustrated in Fig. 13, a set of marked master slope lines is included. The value of this arrangement is easily illustrated by the graphical example shown.

Example 1. To determine the required average effective external coil surface temperature. *Given:* (1) Air entering cooling coil at temperature of 83 F dry-bulb and 69 F wet-bulb. (2) Ratio of latent to total heat that must be removed from air is 35 per cent. *Required:* To find the average external coil surface temperature.

Solution. (1) Draw through point *N*, at the origin of the heat load ratio lines, a line *N-O* with a slope of 35 per cent in accordance with scale *S*. (2) Mark in the body of the chart, point *P* representing the condition of air entering the cooling coil at 83 F dry-bulb and 69 F wet-bulb. (3) Through point *P* draw a line *P-Q* parallel to line *N-O*. (4) The line *P-Q* intersects the saturation curve at 51 F, which means that the effective external coil surface temperature must be maintained at 51 F in order to obtain the desired 35 per cent latent to total ratio of heat removal from the air passing over the given cooling coil.

Inspection of Equation 12 reveals that the total capacity of a coil is dependent on the entering and leaving wet-bulb temperatures. The entering dry-bulb temperature is unimportant.

The amount of latent heat of condensation of a coil can be calculated from:

$$q_L = 1060\,\eta_c G\,(W_1 - W_s) \tag{14}$$

where

q_L = latent heat removed, Btu per hour per square foot of coil face area.
W_1 = pounds of moisture per pound of dry air entering the coil.
W_s = pounds of moisture per pound of dry air saturated at the average surface temperature.
1060 = average value of latent heat of water vapor, Btu per pound of vapor

The amount of sensible heat removed can be obtained by subtracting the value of q_L from the value of q in Equation 12.

Equation 12 gives accurate results when it is used for coils having a small change of temperature of the fluid in the tubes, as for example with evaporating refrigerants and with water having a small temperature rise. In cases where water in the tubes has a large temperature rise, the effective surface temperature changes throughout the depth of the coil, and in extreme cases moisture may be condensed on only a portion of the coil. In such cases it is possible to estimate the wet and dry portions of the coil separately, using cut-and-try methods.[6]

Internal Film Coefficient

The internal film coefficient, h_r, which appears in Equation 3, is evaluated in various ways, depending upon the nature of the fluid, and whether the fluid is changing state.

When evaporating refrigerants are being used in tubes, the temperature of the fluid is fairly constant, being affected principally by pressure drop through the tubes, by superheat of the evaporated refrigerant, and by the presence of oil in solution. To obtain maximum coil capacity it is necessary to keep the pressure drop through the tubes at a minimum (¼ lb per square inch), to keep the superheat as low as possible without carrying liquid back to the compressor, and to arrange for good separation and return of oil to the compressor. An additional important factor is the removal of gas so that the tube surface may be flooded with liquid as much as possible. The internal film coefficient is markedly increased by heavy heat loads, because the increased turbulence and gas velocity cause good contact of the liquid with the tubes. Values of h_r usually lie between 150 and 450. For rating of dehumidifying coils, satisfactory results are obtainable by first determining the average external surface

[6]Calculation of Coil Surface Areas for Air Cooling and Dehumidification, by John McElgin and D. C. Wiley (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 139).

temperature as previously described, and then using the difference between the external film temperature and the refrigerant for evaluating h_r in Equation 15.

$$h_r = \frac{q}{\frac{A}{R}(t_s - t_r)} \qquad (15)$$

The term $(t_s - t_r)$ is commonly written Δt. The usefulness of the foregoing equation is impaired by the fact that both h_r and Δt must be evaluated experimentally.

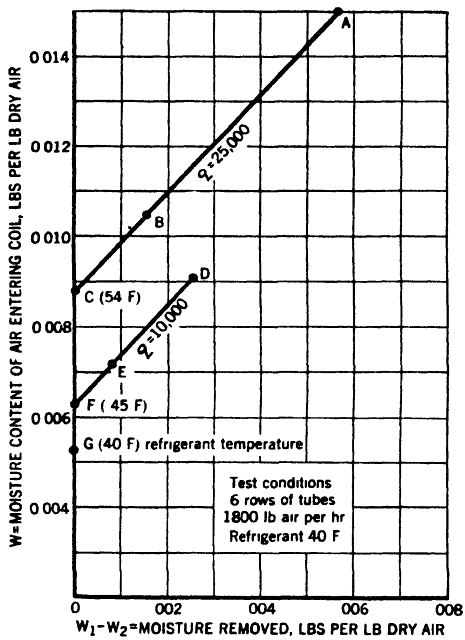

FIG. 14. DETERMINATION OF SURFACE TEMPERATURE

More direct results can be obtained by ignoring h_r and determining the relation between Δt and total coil capacity:

$$\Delta t = t_s - t_r = mq^n \qquad (16)$$

where

m and n = constants determined by tests.

When water is used as a cooling medium in tubes, the rate of heat transfer is a function of water velocity, because this results in an increase in the number of contacts of the water molecules with the tube surface, per unit of time. Thus increased water velocity and reduced tube diameter cause increased heat transfer. Heat transfer is also greater at higher temperatures of the water. The basic formula for the film coefficient of heat transfer for flow of water is as follows:

$$h_w = 1.5\,(t + 100)\,\frac{V^{0.8}}{D^{0.2}} \qquad (17)$$

where

h_w = internal film coefficient of heat transfer, Btu per (hour) (square foot of internal tube surface) (degree Fahrenheit).
V = water velocity, feet per second.
D = internal diameter of tube, inches.
t = average water temperature, degrees Fahrenheit.

In the case of finned tubes, values of h_w may be lower than those obtained by use of Equation 17. Accurate results can be obtained by using Equation 15, if the logarithmic mean temperature difference between surface and water is used in place of Δt.

When saturated steam is condensed in the tubes of coils, the film

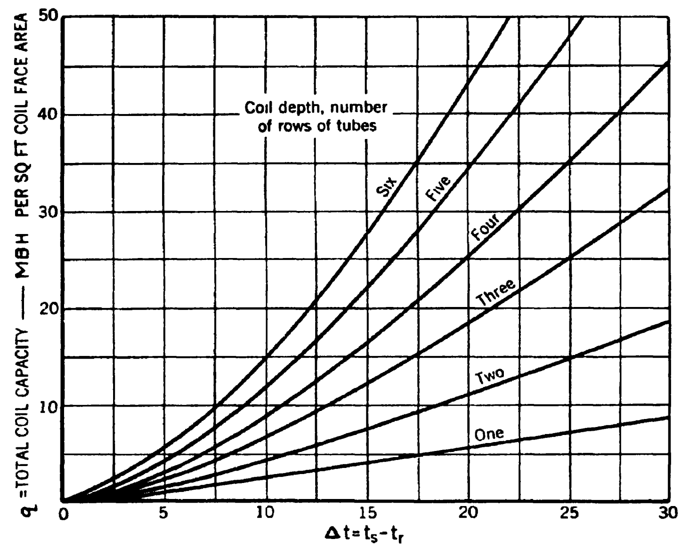

FIG. 15. TYPICAL CURVES SHOWING RELATION BETWEEN TOTAL CAPACITY AND TEMPERATURE DIFFERENCE FOR REFRIGERANTS

coefficient h_r varies from 1000 to 2000, depending on freedom from air in the steam, and upon good drainage of the tubes. The coefficient is fairly constant for a particular coil, giving values of Δt that are directly proportional to q.

GRAPHICAL ANALYSIS OF COIL PERFORMANCE

In testing coils, determination of surface temperatures is most important. A convenient way of determining surface temperatures is illustrated in Fig. 14. Test points A and B are made without varying the wet-bulb temperature of entering air, the air velocity, the refrigerant temperature, and the total capacity of the coil. Only the dry-bulb and dew-point temperatures of the entering air are varied. A straight line is drawn between points A and B, and is extended to the ordinate of zero moisture removal, giving point C which represents the moisture content of saturated air that corresponds to the surface temperature. Points D and E are similarly plotted, the only difference being that another

total coil capacity and entering air wet-bulb temperature are chosen. The saturation temperatures of points *C* and *F* are then used in Equation 16, in conjunction with the test values of t_r and q, so as to evaluate the constants m and n by solving two simultaneous equations. The resulting equation is plotted as shown in Fig. 15, or can be plotted as a straight line on logarithmic paper.

Having determined the surface temperature, the test data can be used to evaluate coil efficiency, from the ratio $(t_1 - t_2) \div (t_1 - t_s)$. Then constants of Equation 11 can be evaluated and a group of curves constructed as in Fig. 12.

Use of Graphs for Predicting Performance

Coil performance under any dehumidifying condition can be predicted

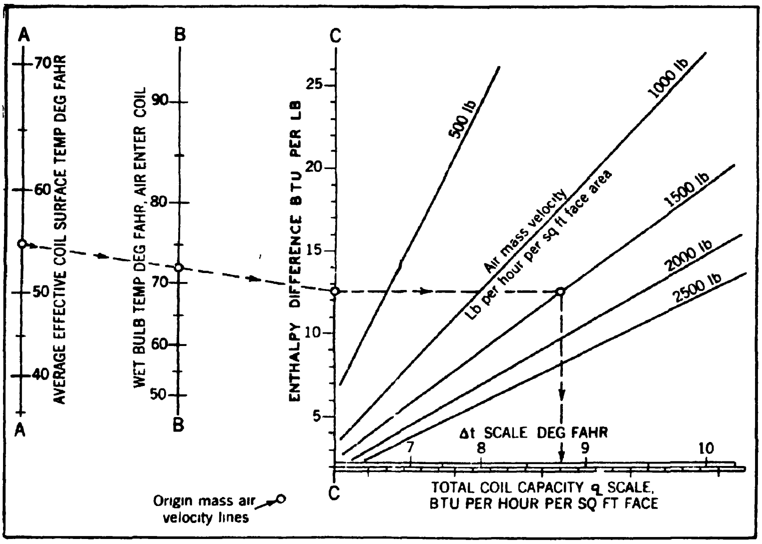

Fig. 16. Typical Coil Performance Chart

as shown in the following example, using Figs. 12, 13 and 15.

Example 2. Given: Total heat to be removed, 18,000 Btu per hour per square foot of coil face area; ratio of latent to total heat, 35 per cent; dry-bulb temperature of air entering coil, 83 F; dew-point temperature of air entering coil, 65 F. *Required:* Coil depth, air velocity and refrigerant temperature.

Solution. (1) Plot the entering air conditions at point *C* on Fig. 13. (2) Draw line *C-E*, parallel to the 35 per cent line *N-O* of the index chart, and obtain the required surface temperature, 55 F. (3) In Fig. 15, assume a coil depth of 4 rows, and obtain $\Delta t = 16$ F. Subtracting 16 deg from 55 deg gives a required refrigerant temperature of 39 F. (4) In Fig. 12, assume an air velocity of 2000 lb per hour and obtain a coil efficiency of 0.8. (5) Solving for q in Equation 12, a capacity of 17,400 Btu per hour is obtained, which is not the required capacity. It is necessary to try a higher air velocity until a balanced condition is found at an air velocity of 2080 pounds per hour. (6) By assuming a coil depth of 6 rows and repeating the same procedure, another solution can be obtained at a refrigerant temperature of 43.5 F and an air velocity of 1760.

The foregoing cut-and-try calculations can be eliminated by the use of

the type of graph shown in Fig. 16, which may be constructed as outlined herewith:

1. The three axes of the nomogram on the left side of the chart are drawn in such a manner that the C axis represents the differences in enthalpy between the air at wet-bulb temperature along B axis and air at wet-bulb temperature along A axis. Thus, the C axis represents the total heat (Btu per pound of air, sensible and latent) which could be removed from the air at some inlet wet-bulb temperature on B axis if the coil heat transfer efficiency were 100 per cent and the wet-bulb temperature of the air could be reduced to some average (effective) external coil temperature on A axis. For example if a straight line is drawn through 72 F wet-bulb temperature of entering air on axis B and the 55 F average effective coil (external surface) temperature on axis B, then this straight line will intersect the C axis at 12.6, which figure represents the difference in enthalpy of air between 72 and 55 F wet-bulb temperature.

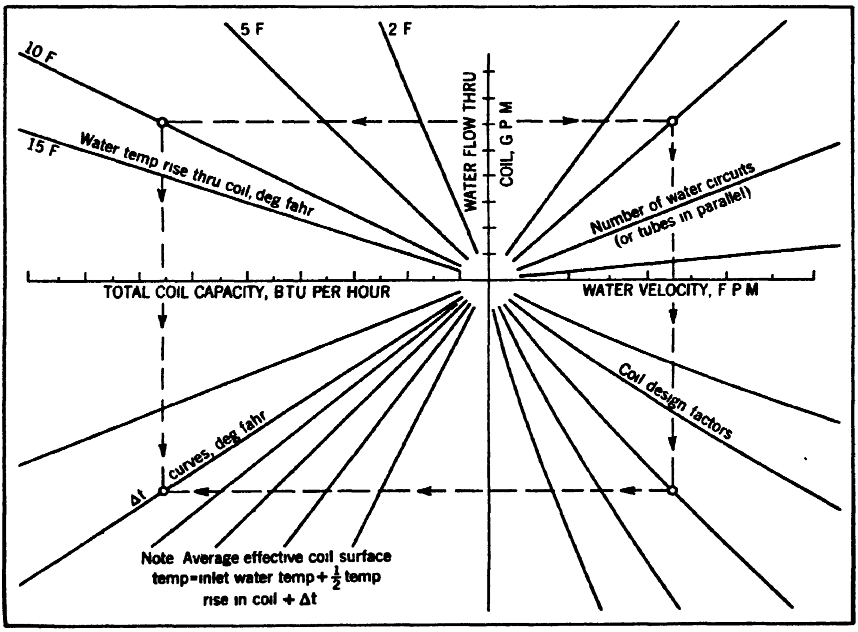

FIG. 17. WATER COIL PERFORMANCE CHART

2. Next scale q is drawn to cover the range of the likely practical loading for the given coil in Btu per hour per square foot coil face area.

3. Lastly, the diagonal mass air velocity lines are drawn in at the intersection of various values on C axis and the corresponding values on the q scale. The values on the q scale corresponding to various values on C axis are obtained by multiplying the values on C axis by mass air velocity and coil efficiency. In this way the calculations required by Equation 12 are performed.

4. Parallel to the q scale is drawn the Δt scale, so that the difference between average surface temperature and refrigerant temperature can be read directly, eliminating the use of Equation 16.

For coils using water as a cooling medium, the chart shown in Fig. 17 can be used for the purpose of eliminating calculations. Such a chart can embrace all sizes of coils of a particular design, but requires an index which gives the *coil design factor* for each size. The coil design factor is the number of square feet of internal tube surface of the entire coil. The curves shown in the lower right hand quarter of the chart perform

the calculations of Equation 17, by using an average water temperature and the actual tube diameter.

Performance of Coils and Refrigeration Compressor

Practically all data published by various makers of direct expansion cooling coils are based upon maintaining a predetermined refrigerant temperature within the coils. While it is often possible to maintain a definite refrigerant temperature within a given cooling coil, for the greater part it is either impossible or impractical. This is due to the fact that the capacity of standard refrigeration compressors is usually fixed and in matching a given cooling coil with a standard compressor the capacity of the latter is often somewhat smaller or greater than that of the former.

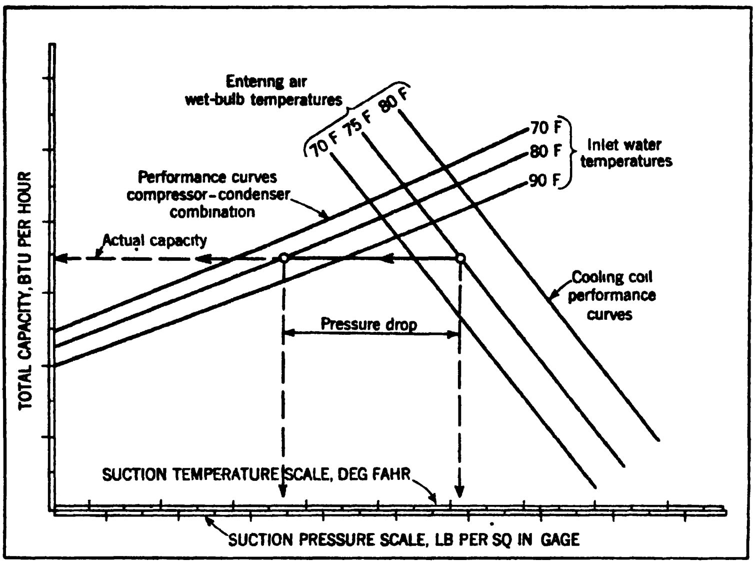

FIG. 18. GRAPHICAL ANALYSIS OF COIL-COMPRESSOR PERFORMANCE

Consequently, very often the refrigerant temperature resulting within a cooling coil and correspondingly the capacity of the coil-compressor combination are not what they were originally calculated to be.

In order to determine the actual performance of a given coil-compressor combination under varying conditions of operation, a graphical solution of the balance point is highly desirable. A typical method of graphical analysis of a coil-compressor combination performance is shown in Fig. 18, which is constructed in a manner described herewith:

1. On a piece of graph paper (with a uniform scale), the equipment capacity scale, total Btu per hour, is laid out along the vertical axis while the refrigerant suction temperature scale is laid out along the horizontal axis.

2. The performance curve of a given compressor with a definite condenser (combination usually called a *condensing unit*) is plotted as a function of suction temperature corresponding to the saturation suction pressure at the compressor suction service valve for a given inlet water temperature and quantity supplied to the condenser.

3. The performance curve of the given cooling coil is next plotted as a function of mean suction temperature within the coil, and the wet-bulb temperature of air entering the coil, for a given mass air velocity over the coil.

4. The refrigerant pressure drop between the center of the cooling coil and the compressor suction service vàlve is computed and converted into the terms of temperature-difference. This temperature difference is then fitted in horizontally between the performance curves of the cooling coil and the compressor, as shown, and the total capacity of the coil-compressor combination is read along the horizontal line upon which the above mentioned *temperature-difference* segment falls.

COIL SELECTION

In the selection of a coil it is necessary to consider several factors:

1. The duty required—heating, cooling, dehumidifying.
2. Temperature of entering air—dry-bulb only if there is no dehumidification, dry- and wet-bulb if moisture is to be removed.
3. Available heating and cooling media.
4. Space and dimensional limitations.
5. Air quantity limitations.
6. Allowable resistances in air circuit and through tubes.
7. Peculiarities of individual designs of coils.
8. Individual installation requirements, such, for example, as type of automatic control to be used.

The duties required may be determined from information in Chapters 4, 5, 6 and 7. There may or may not be a choice of cooling and heating media, as well as temperatures available, depending upon whether the installation is new or is in combination with present sources of heating or cooling. Space limitations are dictated by the requirements of individual cases. The air quantity is influenced by a number of considerations. The air quantity through heating coils is often made the same as that necessary to handle the summer cooling load. The air handled may be fixed by the use of old ventilating ducts as an air distribution system for new air conditioning apparatus, or may be dictated by requirements of satisfactory air distribution or ventilation. The resistance through the air circuit influences the fan horsepower and speed. This resistance may be limited to allow the use of a given size of fan motor, or to keep the operating expense low, or it may be limited by the maximum fan peripheral velocity which requirements of quietness may permit. The friction through the water or brine circuit may be dictated by the head available from a given size of pump and pump motor. As the fan and pump motor inputs represent a refrigerating load on cooling installations, it is economical to keep them low.

Proper performance of a surface heating or cooling coil depends upon correct choice of the original equipment and upon certain other factors. The usual coil ratings are based on a uniform face velocity of air. If the air is brought in at odd angles or if the fan is located so as to block part of the air flow, the performance as given in the manufacturer's ratings cannot usually be obtained. To obtain this performance it is necessary also that the air quantity be adjusted on the job to that used in determining the coil selection, and must also be kept at this value. The most common causes for a reduction of air quantity are the fouling of the filters and collection of dirt in the coils. These difficulties can be avoided by proper design and proper servicing. There are a number of ways in which coils may be cleaned. A common method is to wash them off with water. They can sometimes be brushed and cleaned with a vacuum cleaner. In bad cases of neglect, especially on restaurant jobs where grease and dirt have accumulated, it is sometimes necessary to remove the coils and wash

off the accumulation with steam, compressed air and water, or hot water. The most satisfactory solution, however, is to keep the filters serviced, and thus make the cleaning of the coils unnecessary.

The proper selection of coils requires an understanding of the necessities of each case and should be based on an economic analysis of the plant design as a whole. No general rule can, therefore, be laid down for the selection of heating or cooling coils. It is possible, however, to point out the limits of usual practice and to indicate the influence of the variables involved in the coil selection.

Heating Coils

Steam and hot water heating coils are usually rated within these limits:

Air Face Velocity—200 to 1200 fpm, sometimes up to 1500 fpm.

Steam Pressure—2 to 200 lb, sometimes up to 350 lb per square inch.

Hot Water Temperature—150 to 225 F.

Water Velocity—2 to 6 fps.

Individual cases may deviate widely, but the tabulation given herewith will serve as a guide to usual heating practice:

Air Face Velocity—500 to 800 fpm face, 500 being a common figure.

Delivered Air Temperature—varies from about 72 F for ventilation only to about 150 F for complete heating.

Steam Pressure—2 to 10 lb, 5 lb being common.

Hot Water Temperature—150 to 225 F.

Water Velocity—2 to 6 fps.

Water Quantity—Based on about 20 F temperature drop through a hot-water coil.

Air Resistance—The total resistance through heating coils is usually limited to from ⅜ to ⅝ in. of water gage for public buildings, to about 1 in. for factories.

The selection of heating coils is relatively simple as it involves dry-bulb temperatures and sensible heat only, without the complication of simultaneous latent heat loads, as in cooling coils. For a given duty, entering air temperature, and steam pressure, it is possible to select several arrangements of the same design of coil depending upon the relative importance of space, cross-sectional area, and air resistance.

Cooling Coils

The usual range of ratings for cooling and dehumidifying coils is enumerated herewith:

Entering Air Dry-Bulb—60 to 100 F.

Entering Air Wet-Bulb—50 to 80 F.

Air Face Velocities—300 to 800 fpm, (sometimes as low as 200 and as high as 1200).

Volatile Refrigerant Temperatures—25 to 55 F, at coil suction outlet.

Water Temperatures—40 to 65 F.

Water Quantities—2 to 6 gpm per ton, or equivalent to a water temperature range of from 4 to 12 F.

Water Velocity—2 to 6 fps.

The ratio of total to sensible heat removed varies in practice from 1.00 to about 1.65, *i.e.*, sensible heat is from 60 to 100 per cent of total, depending on the application. (See Chapter 20.) Required ratios may demand wide variations in air velocities, refrigerant temperatures, and coil depth, so that general rules as to these values may be misleading. On usual comfort installations air face velocities between 400 and 600 fpm

are frequent, 500 being a common value. Refrigerant temperatures will ordinarily vary between 40 and 50 F where cooling is accompanied by dehumidification. Water velocities will range from 2 to about 6 fps.

When no dehumidification is desired, for which condition the dew-point of the entering air will be equal to or lower than the cooling coil surface temperature, the coil selection is made on the basis of dry-bulb temperatures and sensible heat transfers only, the same as with heating coils. It is possible also to choose various arrangements of face area, depth, air velocity, etc., for the same duty.

Dehumidifying Coils

The selection of coils for combined cooling and dehumidifying duty is more involved than for heating or sensible cooling and requires consideration of both dry- and wet-bulb air temperatures. It is further complicated by the fact that the proportional amount of dehumidification required is also highly variable. The methods outlined previously under Heat Transfer and Resistance may be used to determine whether it is possible for a coil to perform the duty required. If entering and leaving air conditions are arbitrarily specified, the corresponding duty sometimes cannot be obtained at all without the use of reheat. As with heating and sensible cooling coils, there are combinations of face areas, depth, air velocity and refrigerant temperatures which will give the required performance. This is illustrated in Table 1.

TABLE 1. VARIOUS COOLING COIL ARRANGEMENTS

SELECTION	1	2	3	4
Total cooling capacity, tons	100	100	100	100
Sensible cooling capacity, tons	69	69	69	69
Latent cooling capacity, tons	31	31	31	31
Ratio total to sensible heat	1.45	1.45	1.45	1.45
Air quantity, cfm	47,800	41,700	37,100	46,800
Cfm per total ton	478	417	371	468
Face velocity, fpm	325	423	500	600
Resistance, in. water	0.11	0.27	0.51	0.37
Coil face area, sq ft	147	99.0	74.2	78.1
Coil rows deep	4	6	8	4
Coil evaporator temp. deg F	45	45	45	38

It is possible as shown in Table 1 to perform approximately the same duty at a given refrigerant temperature with small face area and large thickness or vice versa. The large face area coil will give low air velocity and resistance but high air quantities per ton. The coil of small face area and great depth will require small air quantities per ton of refrigeration, high resistance and high air velocities. As shown also in Table 1 the same sensible, latent and total cooling capacity may be obtained with various refrigerant temperatures by proper choice of coil. This makes it possible to keep the evaporating temperature high enough to carry the load with a chosen size of condensing unit. High evaporating temperatures with correspondingly small compressor operating expense can be attained but at the expense of coil surface, air quantity or both. The choice will be determined by the necessities of individual installations.

For a given quantity and condition of entering air the evaporating temperature of a volatile refrigerant coil will be determined by a balance

TABLE 2. CAPACITY BALANCES FOR MAXIMUM AND MINIMUM LOAD CONDITIONS

Conditions	Capacity in Tons			Ratio Total / Sensible
	Total	Sensible	Latent	
Required at peak load conditions	10.90	7.90	3.00	1.38
Required at minimum load conditions	6.62	3.36	3.26	1.98
Peak load equipment balance	10.90	7.90	3.00	1.38
Same equipment balanced at minimum load conditions	9.85	6.58	3.26	1.50
Same equipment balanced at maximum load conditions with 40 per cent by-pass	8.38	5.05	3.33	1.66
Same equipment balanced at minimum load conditions with 38,800 Btu per hour reheat	6.62	3.36	3.26	1.98

between the condensing unit and the coil. The total, sensible and latent cooling capacity can then be determined from the coil rating information. If the condensing unit and cooling coil have been properly balanced for the required load and, due to miscalculated duct resistance or improper choice of fan speed, the air quantity is reduced, the total cooling capacity will also be reduced. The decrease is generally in the sensible capacity. This is the effect also when the air by-pass or volume control is used.

It is necessary that not only the total capacity but also the sensible and latent cooling requirements both be met. The installation of an excess of coil will result in an increase in total capacity, but not a proportional gain in latent heat capacity. On installations controlled from dry-bulb temperature the operating time will be shortened because of the added sensible cooling capacity. The result will be less moisture pick-up than calculated, and higher relative humidity. If an oversize condensing unit is installed the opposite situation will take place. The relative humidity will be lower than estimated. This is not generally a disadvantage except that it results in a greater load from outside air than calculated, as well as in increased power consumption. If oversize equipment is furnished, a balance should be made to assure that the ratio of total to sensible capacity is the same as in the estimated load.

Sometimes arbitrary air quantities are specified for ventilation or other reasons independent of the selection of the cooling coil. As shown in Table 1, the coil selection can be altered to take care of various air quantities for the same duty.

Where coil and condensing unit are selected for the peak load condition, and the sensible load partially disappears due to fall of outside temperature or other cause, the condensing unit and coil rebalance. This may result in more sensible capacity than required at the light load condition and less latent in proportion, with an increased relative humidity in the conditioned space. Such a condition is shown in Table 2. If approximately 40 per cent of the total air is by-passed, the condition will be improved as indicated. The situation could be entirely avoided by using reheat, where it is possible to handle any ratio of sensible and latent loads and maintain the design temperature and humidity[7].

Care should be taken to avoid freezing at light loads. In general, freezing occurs when the coil surface temperature falls to 32 F. With usual coils for comfort installations, this will not occur unless the evapo-

[7]Reheating by Means of Refrigerant Compressor Discharge Gas, by S. F. Nicoll (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 239)

rating temperature at the coil outlet is about 20 to 25 F. The exact value depends on the design of coil and the amount of loading. Although it is not customary to choose coil and condensing units to balance at low temperatures at peak loads, there is danger of this occurring when the load decreases. This is further aggravated if a by-pass is used so that less air is passed through the coil at light loads. It may be even worse if the control is arranged for decrease of inside temperature with fall of that outside. Freezing can be avoided by making the full load balance a high evaporating temperature and checking the balance at the minimum load.

Care should be exercised in the design of humidity control to minimize the cycling of the refrigerating compressor because of re-evaporation of moisture from the fins. It is sometimes necessary to by-pass air around a coil when the compressor is not operating.

CHAPTER 26

Spray Equipment

Air Washers, Humidification with Air Washer, Apparatus for Direct Humidification, Air Dehumidification with Washers, Well and Water Main Temperatures, Atmospheric Water Cooling Equipment, Selection of Water Cooling Equipment, Design Wet-bulb Temperatures for Water Cooling, Cooling Ponds, Spray Towers, Cooling Tower Design, Cooling Tower Performance, Winter Freezing

AIR humidification is effected by the vaporization of water and always requires heat from some source. This heat may be added to the water prior to the time vaporization occurs or it may be secured by a transformation of sensible heat of the air being humidified to latent heat as the vapor is added to the air. The thermodynamics of the process are discussed in Chapter 1. The removal of moisture from air may or may not involve the removal of heat from the air-vapor mixture. With spray equipment dehumidification of air always necessitates the removal of heat.

AIR WASHERS

Air washers may be used as either humidifiers or dehumidifiers depending upon the method of operation and the temperature of the spray water. The functions of an air washer are to regulate the moisture and heat content of air passing through it and to remove dust and dirt from the air. Air washers are not as effective as air filters in the removal of dust and dirt.

The construction of commercial air washers is indicated in Figs. 1 and 2. Any air washer consists essentially of a chamber through which the air passes in intimate contact with water. The lower portion of the washer chamber serves as a sump for the spray water.

Contact between the air and the washer water is secured: (1) by breaking the water into a very fine mist, (2) by passing the air over surfaces which are continuously wetted by water, or (3) by a combination of water sprays and wetted plates. Scrubber-plate types of washers are used largely to wash heavy reclaimable products from the air, and are generally composed of one to three eliminator-type baffle scrubber plates across the air stream. Water is supplied at the tops of the scrubber plates by flooding nozzles placed across the top of the washer. Spray washers have one or more banks of water atomizing nozzles placed in the air stream above the level of the water in the sump. The direction of the water sprays may be against the air stream, with the air stream, or with one bank spraying with the air stream and one against it. The number of nozzles required depends upon their design, the quantity of air handled, and the arrangement of the nozzles.

Scrubbers generally consist of eliminator-type baffle plates placed in the air stream to cause several reversals of the direction of air flow. The scrubber plates are more effective as air cleaners than as humidifiers. All washer chambers should have inlet diffuser plates to aid in producing more uniform air flow through the washer spray chamber. These inlet vanes also aid in preventing spray water from being thrown into the air duct ahead of the washer. However, if the water spray is against the air flow, the ordinary perforated diffuser plate is not sufficient, and specially designed eliminator baffles must be used to prevent spray from passing

into the air inlet duct. At the outlet end of the washer suitable flooded eliminator plates, which will cause from 4 to 6 reversals of the direction of air flow, should be installed for the purpose of removing drops of unvaporized water from the leaving air. When the air carries certain substances mixed with it, the spray water may become acidulated and special consideration must be given to the materials used, to reduce the corrosive action.

Essential items in air washer operation are: uniform distribution of the air across the chamber section above the level of the water in the sump; moderate velocities of air flow, 300 to 600 fpm in the spray chamber; an adequate amount of spray water broken up into a fine mist throughout the air stream; sufficient length of air travel through the

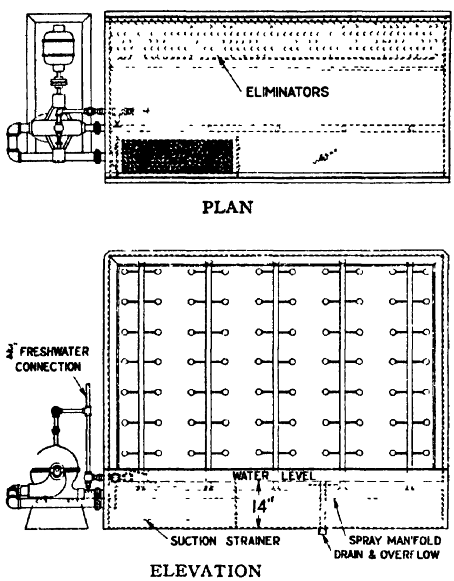

FIG. 1. TYPICAL SINGLE-BANK AIR WASHER

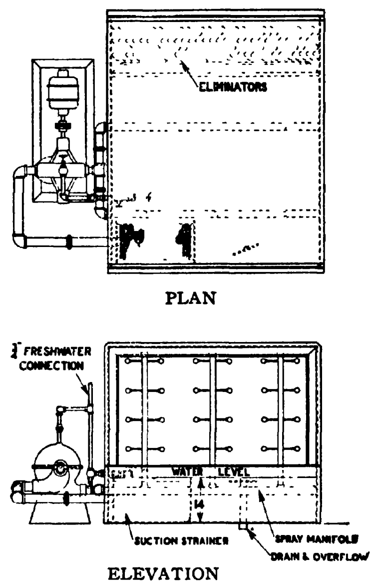

FIG. 2. TYPICAL TWO-BANK AIR WASHER

water spray and over thoroughly wetted surfaces; and the elimination of free moisture from the air as it leaves the unit.

Washers are sometimes arranged in two or more stages to cool through long ranges or to increase the over-all efficiency of heat transfer between the air and the heating or cooling medium. A multi-stage washer is equivalent to a number of washers in a series arrangement. Each stage is in effect a separate washer.

Usually the catalog capacity of a washer is expressed in cubic feet of air per minute and is based upon an air velocity of 500 fpm through the gross inlet area of the unit. At this rating spray type washers handle about 2½ gpm of water per bank per square foot of area, that is, about 5 gpm per bank per 1000 cfm. These proportions of air, water, area, and velocity may be departed from to meet the needs of some particular job, but certain limiting relationships should be observed.

For a single-stage air washer, a 15 F drop in dry-bulb temperature of the air passing through the washer is about the maximum that should be

anticipated. For greater decrease in dry-bulb temperature, multi-stage washers should be utilized. A rise of 6 F should be the calculated maximum for the spray water.

The width and height of a washer may be dictated by space limitations outside the washer, such as headroom, or by the inside space requirements, such as face area needed by a bank of cooling coils. The length of a washer is determined by the number of spray banks, or scrubber plates, and if cooling coils are installed in the unit, by the number of banks of coils. Roughly, a spray space of about 2 ft 6 in. in length is required for each bank of sprays; *leaving* eliminators require about 1 ft 6 in., and *entering* eliminators about 1 ft.

The resistance to air flow through an air washer varies with the type of eliminators, number of banks of sprays, direction of spray, air velocity, type of scrubber plates, and size and type of cooling coils if located in the washer. Manufacturers should be consulted to obtain the resistance for a particular installation.

HUMIDIFICATION WITH AIR WASHER

Air humidification can be accomplished in three ways with an air washer. These are: (1) use of recirculated spray water without prior treatment of the air, (2) preheating the air and washing it with recirculated spray water, and (3) using heated spray water. In any air washing installation the air should not enter the washer with a dry-bulb temperature less than 35 F in order to eliminate danger of freezing the spray water.

Method 1. Except for the small amount of energy added from outside by the recirculating pump in the form of shaft work, and for the small amount of heat leak from outside into the apparatus, including the pump and its connecting piping, the process would be strictly adiabatic. Evaporation from the liquid spray would therefore be expected to bring the air immediately in contact with it to saturation adiabatically; and, since the liquid is recirculated, its temperature would be expected to adjust to the thermodynamic wet-bulb temperature of the entering air.

It does not follow from the foregoing reasoning that the whole air stream is brought to complete saturation, but merely that its state point should move along a line of constant thermodynamic wet-bulb temperature as explained in Chapter 1. The extent to which the final temperature approaches the thermodynamic wet-bulb temperature of the entering air, or the extent to which complete saturation is approached is conveniently expressed by a ratio known as *humidifying effectiveness* or *saturating effectiveness* and is defined:

$$e_h = \frac{t_1 - t_2}{t_1 - t'} \tag{1}$$

where

e_h = humidifying effectiveness, per cent.
t_1 = dry-bulb temperature of the entering air, degrees Fahrenheit.
t_2 = dry-bulb temperature of the leaving air, degrees Fahrenheit.
t' = thermodynamic wet-bulb temperature of the entering air, degrees Fahrenheit.

The humidifying or saturating effectiveness of a washer is dependent upon the number of spray banks and nozzles, the effectiveness of the nozzles in breaking an adequate quantity of water into a fine spray, the velocity of air flow through the water sprays, and the time of the contact of the air with the spray water. Other conditions being the same, low velocity of air flow is more conducive to higher humidifying effectiveness.

The following may be taken as representative humidifying or saturating effectiveness of an air washer for the conditions stated:

1 bank—downstream	60–70 per cent
1 bank—upstream	65–75 per cent
2 banks—downstream	85–90 per cent
2 banks—1 upstream and 1 downstream	90–95 per cent
2 banks—upstream	90–95 per cent

The air leaving the washer may require reheating to produce the required dry-bulb temperature and relative humidity.

Method 2. The preheating of the air increases both the dry- and wet-bulb temperatures, lowers the relative humidity, but does not alter the humidity ratio (pound water vapor per pound dry air). At a higher wet-bulb temperature but the same humidity ratio, more water can be absorbed per pound of dry air in passing through the washer, assuming that the humidifying effectiveness of the washer is not adversely affected by operation at the higher wet-bulb temperature. The analysis of the process occurring in the washer itself is the same as that explained under *Method 1.* The final desired conditions are secured by adjusting the amount of preheating to give the required wet-bulb temperature at entrance to the washer and then reheating when necessary after passage through the washer.

Method 3. Even if heat is added to the spray water, the mixing occurring in the washer itself may still be regarded as adiabatic. The state point of the mixture should move in a direction determined by the specific enthalpy of the heated spray as explained in Chapter 1. By sufficiently elevating the spray water temperature it should be possible to completely saturate the air and even raise its temperature above the dry-bulb temperature of the entering air.

APPARATUS FOR DIRECT HUMIDIFICATION

Humidifiers may be divided into the following general types, according to the method of operation: (1) indirect, such as the air washer, which introduces moistened air; and (2) direct, which sprays moisture into the room or introduces moisture by means of steam jets.

As in the cases of humidification by use of an air washer, the heat necessary for the vaporization of the moisture added to the air by direct humidification is secured either from heat stored in the spray water or by a transformation of sensible to latent heat in the air humidified. In the latter case the enthalpy of the air remains constant but the dry-bulb temperature of the air is reduced.

Direct humidification is usually preferable where high relative humidities must be maintained, but where there is little cooling or ventilation required. In comfort air conditioning, where both humidification and ventilation are required, the indirect humidifier is preferable. In industrial applications, where the cooling or ventilation load is large and where very high relative humidities must be maintained, a combined system employing both direct and indirect humidifiers is sometimes used.

Spray Generation

Spray generation is obtained by (1) atomization, (2) impact, (3) hydraulic separation, and (4) mechanical separation.

Atomization involves the use of a compressed air jet to reduce the water particles to a fine spray. With the *impact* method, a jet of water under pressure impinges directly on the end of a small round wire. Where

hydraulic separation is employed, a jet of water enters a cylindrical chamber and escapes through an axial port with a rapid rotation which causes it immediately to separate in a fine cone-shaped spray. In the *mechanical separation* process, water is thrown by centrifugal force from the surface of a rapidly revolving disc and separates into particles sufficiently small to be utilized in certain types of mechanical humidifiers.

Spray Distribution

Spray distribution is obtained by (1) air jet, (2) induction, and (3) fan propulsion.

The air jet which generates the spray in atomizers also carries the spray through a space sufficient for its distribution and evaporation, and this method of distribution is termed *air jet*. Where distribution is obtained by *induction*, the aspirating effect of an impact or centrifugal spray jet is utilized to induce a current of air to flow through a duct or casing, and this air current distributes the spray. *Fan propulsion* obviously consists of the utilization of fans to entrain and distribute the spray.

Industrial type direct humidifiers are commonly classified as (1) atomizing, (2) high-duty, (3) spray and (4) self-contained or centrifugal.

Atomizing Humidifiers

There are several types of atomizing humidifiers, all of which rely upon compressed air as the atomizing and distributing agency, similar to the familiar method used in ordinary nasal atomizers. Compressed air (ordinarily about 30 lb per square inch) is supplied from a centrally-located air compressor through pipe lines to the atomizing units. The air lines are usually horizontal and parallel to water lines which supply water by gravity from a float tank. The water in the tank is maintained at a constant level slightly lower than the outlets of the atomizers themselves and is drawn constantly to the atomizer by aspiration when compressed air is supplied. This aspiration ceases and the flow of water stops when the air supply is cut off. The water should not be supplied under pressure to atomizers because of the possibility of leakage, drip, or coarse spray. These cannot occur when water is supplied by aspiration.

High-Duty Humidifiers

Water is supplied to high-duty humidifiers under high pressure (usually about 150 lb per square inch) through pipe lines from a centrally-located pumping unit. The spray-generating nozzle which is of the impact type is located in a cylindrical casing. A drainage pan provides for the collection and return of unevaporated water which flows through a return pipe to a filter tank, from which it is recirculated. A powerful air current is forced through the humidifier by means of a fan mounted above the unit.

The air enters from above, is drawn through the head, charged with moisture, and cooled. It then escapes from the opening below at a high velocity in a complete and nearly horizontal circle. The spray is evaporated and the resulting vapor diffused. This distribution of fine spray over the maximum possible area promotes complete and rapid vaporization even at high humidities.

Spray Humidifiers

This type of humidifier consists of an impact spray nozzle in a cylindrical casing with a drainage pan below it. The aspirating effect of the spray nozzle induces a moderate air current through the casing which distributes the entrained spray. The general method of circulating and

returning the water is similar to that employed for high-duty humidifiers. A suitable pump and centrally-located filter tank are required.

Self-Contained Humidifiers

The self-contained or centrifugal humidifier has the ability to generate and distribute spray without the use of air compressors, pumps, or other auxiliaries. These may be used either singly or in groups. In large installations, where suitable connections are provided to permit the cleaning and servicing of individual units without affecting the room as a whole, group control of the water and power may be employed.

AIR DEHUMIDIFICATION WITH WASHERS

Moisture removal from an air-vapor mixture can be accomplished by use of an air washer so long as the temperature of the spray medium is lower than the dew-point of the air passing through the unit. The final dry-bulb temperature and the relative humidity of the air leaving a dehumidifier washer are dependent upon: the air velocity, the length of air travel through the sprays, the dry- and wet-bulb temperatures of the entering air, the spray temperature, the number of spray banks and nozzles, the quantity of spray medium handled, and the effectiveness of the nozzles in breaking the spray into a fine mist.

Both sensible and latent heat are removed in the process of dehumidification by cooling. Abstraction of sensible heat occurs during the entire time that the air is in contact with the spray medium. Latent heat removal takes place as condensation occurs. Therefore, the lower the spray temperature the greater the amount of moisture removal per pound of dry air, all other conditions remaining the same. Washers with two or more banks of sprays are usually selected for comfort air conditioning installations. Such washers will cool the air to within 1 or 2 F of the leaving spray water temperature.

Where a limited supply of cold water is available multiple stage washers may be used to an advantage. The cool water is pumped through the multiple spray systems in series. By this arrangement the entering air is cooled first by the warmer water and finally by the cooler water which gives the maximum amount of cooling with the minimum amount of water. The approximate temperatures of water from wells at depths of 30 to 60 ft are given in Fig 3[1]. Frequently the temperature of the city water main supply is low enough during the summer to permit an appreciable cooling effect. Table 1 lists the maximum city water main temperatures for various localities in the United States and Canada.

Air washers using refrigerated spray generally have their own recirculating pumps. These pumps deliver to the sprays a mixture of water from the washer sump, which has not been re-cooled, and refrigerated water. The quantities of each are controlled by a three-way or mixing valve actuated by a dew-point thermostat located in the washer air outlet or by humidity controllers located in the conditioned space.

ATMOSPHERIC WATER COOLING EQUIPMENT

In the operation of a refrigerating plant or a condensing turbine, one of the main problems is the removal and dissipation of heat from the compressed refrigerant or the discharged steam. This is accomplished

[1]Temperature of Water Available for Industrial Use in the United States, by W. D. Collins (*U. S. Geological Survey*, Water Supply Paper No. 520 F).

Table 1. Average Maximum Water Main Temperatures[a]

State	City	Temp. F	State	City	Temp F
Ala.	Birmingham	84	Mass.	Lowell	50
	Mobile	73		Lynn	68
Ariz.	Phoenix	81		New Bedford	70
	Tucson	80		Salem	68
Calif.	Anaheim	60		Worcester	76
	Berkeley	69	Mich.	Detroit	77
	Fresno	72		Flint	70
	Fullerton	75		Grand Rapids	84
	Glendale	68		Highland Park	77
	Los Angeles	75		Jackson	56
	Oakland	69		Kalamazoo	53
	Ontario	70		Lansing	64
	Pasadena	82		Saginaw	82
	Pomona	75	Minn.	Duluth	55
	Riverside	78		Minneapolis	80
	Sacramento	72		St. Paul	77
	San Bernardino	65	Mo.	Jefferson City	82
	San Diego	82		Kansas City	84
	San Francisco	62		St. Joseph	84
	Whittier	75		St. Louis	85
Colo.	Denver	75		Springfield	70
Conn.	Bridgeport	66	Nebr.	Lincoln	63
	Hartford	73		Omaha	87
	New Haven	76	Nev.	Reno	61
	Waterbury	72	N. H.	Manchester	76
D. C.	Washington	84	N. J.	Jersey City	63
Del.	Wilmington	83		Newark	74
Fla.	Jacksonville	80		Paterson	78
	Miami	80		Trenton	79
	Tampa	77	N. Y.	Albany	68
Ga.	Atlanta	87		Buffalo	75
	Macon	80		Jamaica	56
Ill.	Chicago	76		Mt. Vernon	74
	Cicero	76		New Rochelle	75
	Evanston	73		New York	72
	Peoria	67		Rochester	70
	Rockford	59		Schenectady	60
	Springfield	82		Syracuse	74
Ind.	Evansville	86		Utica	69
	Gary	75		Yonkers	70
	Indianapolis	80	N. C.	Asheville	74
	South Bend	61		Charlotte	85
	Terre Haute	82		Winston-Salem	82
Iowa	Cedar Rapids	78	N. M.	Albuquerque	65
	Des Moines	77	Ohio	Akron	76
	Sioux City	62		Canton	50
Kans.	Concordia	57		Cincinnati	84
	Kansas City	86		Cleveland	74
	Topeka	88		Columbus	82
	Wichita	72		Dayton	60
Ky.	Louisville	85		Lakewood	82
La.	Baton Rouge	85		Springfield	72
	New Orleans	85		Toledo	83
Me.	Augusta	60	Okla.	Oklahoma City	82
Md.	Baltimore	75		Tulsa	85
Mass.	Boston	80	Ore.	Eugene	60
	Cambridge	70		Portland	64
	Fall River	76			

[a]These averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown.

TABLE 1. AVERAGE MAXIMUM WATER MAIN TEMPERATURES[a] (Concluded)

State	City	Temp. F	State	City	Temp. F
Pa.	Altoona	74	Va.	Fredericksburg	75
	Erie	75		Lynchburg	73
	Johnstown	74		Norfolk	80
	McKeesport	82	Wash.	Olympia	58
	Philadelphia	83		Seattle	62
	Pittsburgh	81		Spokane	51
R. I.	Providence	68		Tacoma	57
S. C.	Charleston	80	W. Va.	Charleston	85
	Greenville	81		Huntington	78
	Spartanburg	78		Wheeling	78
S. Dak.	Rapid City	55	Wis.	LaCrosse	54
Tenn.	Chattanooga	84		Madison	58
	Knoxville	89		Milwaukee	70
	Memphis	70		Racine	68
	Nashville	90			
Texas	Amarillo	65			
	Austin	90	Province		
	Beaumont	86			
	Dallas	86			
	Fort Worth	84			
	Galveston	90	Alta.	Calgary	64
	Houston	84	B. C.	Vancouver	60
	Port Arthur	83	Ont.	London	50
	San Antonio	76		Toronto	63
	Wichita Falls	85	P. E. I.	Charlottetown	48
Utah	Logan	44	Que.	Montreal	78
	Salt Lake City	60		Quebec	68

[a]These averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown.

ordinarily by first transferring the heat of the gas to water in a heat exchanger, from which water it may then be dissipated in a number of ways. If the plant is situated on the banks of a river or lake, an intake may be taken up-stream or at a considerable distance from the discharge, to prevent mixing of the heated discharged water with the inlet water. If the source of cooling water is a city supply or a well, the discharge water may be run into the nearest sewer or open waterway. Lacking an unlimited water supply, or in cases where purchased water is too expensive or where the water available contains dissolved salts which would form scale on the heat-exchanging apparatus, it is necessary to recirculate the water, and to cool it after each passage through the heat-exchanger by exposure to air in an atmospheric water cooling apparatus.

Air has a capacity for absorbing heat from water when the wet-bulb temperature of the air is lower than the temperature of the water with which it is in contact. The rapidity with which this transfer of heat occurs depends upon (1) the area of water in contact with the air, (2) the relative velocity of the air and water, and (3) the difference between the wet-bulb temperature of the air and the temperature of the water. The rate of heat dissipation is influenced further by many small factors[2] prevailing upon these primary ones, and complicating the design. Selection of equipment for any specified service must ultimately rest in over-all economic considerations established from reliable performance data.

[2]Cooling Tower Performance Studies, by L. M. K. Boelter (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 615).

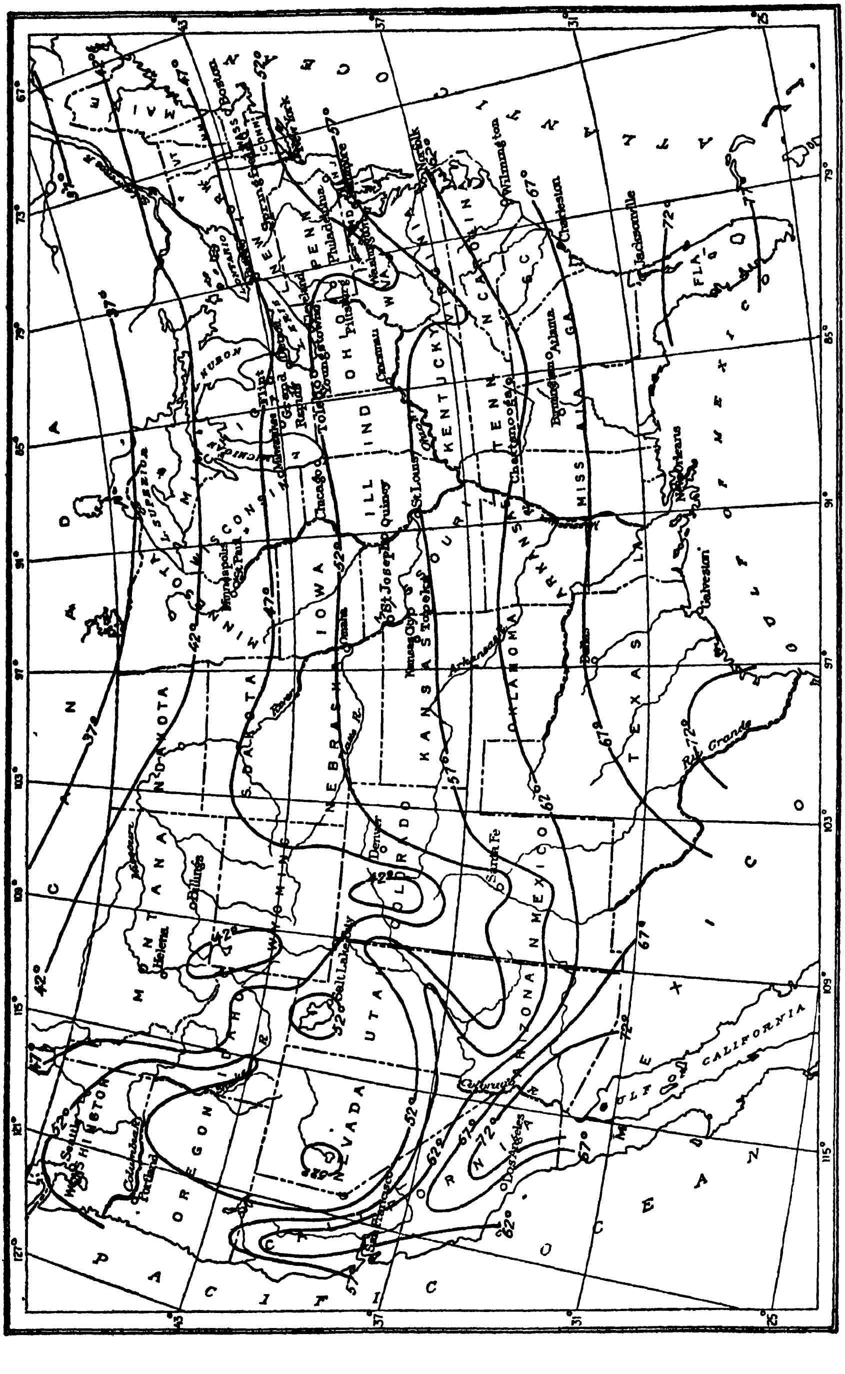

Fig. 3. Approximate Well Water Temperatures at Depths of 30 to 60 Ft

As the heat content of the air increases, its wet-bulb temperature rises. (See Chapter 1.) Because it is impractical to leave the air in contact with water for a long enough time to permit the wet-bulb temperature of the air and the temperature of the water to reach equilibrium, atmospheric water cooling equipment aims to circulate only enough air to cool the water to the desired temperature with the least expenditure of power.

In an air washer, humidifier or dehumidifier, the air is first conditioned by water to change its moisture and temperature, and it is then sent to the place where it is to be used. In water cooling equipment the temperature of the water is reduced by air, and the cooled water is carried to its point of usage. In the air washer, an excess of water is used to condition a fixed quantity of air, while in water cooling equipment, an excess of air is used to cool a fixed quantity of water.

Both types of equipment have a common basis of design in that the size of the equipment is determined by the quantity of air to be handled. With the air washer, the size of the equipment is fixed by the quantity of air to be conditioned, and the amount of conditioning is controlled by the quantity and temperature of the water supplied and its method of application. With water cooling apparatus, its size and the quantity of air required bear no direct relation to the quantity of water being cooled, but vary through a wide range for different services and conditions.

Selection of Water Cooling Equipment

The proper size and type of water cooling equipment for a given installation cannot be determined accurately without considering the characteristics of all types, together with all requirement factors and other matters which influence the selection. Very few installations are exactly alike in details of requirements, and conditions governing performance and operation of the several types of water cooling equipment vary widely with geographical location as well as other considerations. It necessarily holds, therefore, that requirement factors and equipment characteristics are closely interlocked and deserve careful study as a related whole.

Before the characteristics of a specific water cooling apparatus can be judged desirable or undesirable the survey must conclude definitely the importance of each of the following items: first cost including all necessary auxiliaries, area, height, weight, effect of wind velocity and direction, drift nuisance, make-up water requirements and cost of chemical treatment if needed, total power for pumping (plus fan operation in the case of mechanical draft), maintenance, available locations (with due thought to possible future expansion, wind restrictions, space cost, proximity and accessibility, etc.), appearance, the equipment's operating flexibility for most economical conformance to varying loads or seasonal changes, and other considerations occurring with regard to a specific application.

Sizes of Equipment

Assuming a definite quantity of water, the size and design of atmospheric cooling equipment are affected by the following primary conditions:

1. Temperature range through which the water must be cooled.
2. Number of degrees above the wet-bulb temperature of the entering air to which the water temperature must be reduced.
3. Temperature of the atmospheric wet-bulb at which the required cooling must be performed.
4. Time of contact of the air with the water. (This involves height or length of the apparatus and velocity of air.)

5. Surface of water exposed to each unit quantity of air.
6. Relative velocity of air and water.

Items 1, 2, and 3 are established by the type of service and geographical location, while items 4, 5, and 6 depend upon the design of the equipment.

The establishment of a proper cooling range depends upon:

1. Type of service (refrigerating, internal combustion engine and steam condensing).
2. Wet-bulb temperature at which the equipment must operate satisfactorily.
3. Type of condenser or heat-exchanger used.

Because the design of an entire plant is usually affected by the quantity and temperature of the cooling water supply, plants should be designed for cooling water conditions which can be most efficiently attained. The first consideration is usually the limiting temperature of the plant. For example, if an ammonia compressor refrigerating plant is to be designed for 185 lb head pressure as a normal maximum, the limiting temperature of the ammonia in the condenser is 96 F. Should the ammonia temperature go above this figure the head pressure will exceed 185 lb and power consumption increases. To obtain this head pressure, the temperature of the circulating water leaving the condenser must always be less than 96 F by an amount depending upon the size and design of the condenser, the quantity of water being circulated, and the refrigerating tonnage being produced. A condenser having a large surface per ton of refrigeration may be designed to operate satisfactorily with the leaving hot water temperature within 3 or 4 F of the ammonia temperature corresponding to the head pressure, while a small condenser might require a 10 F difference.

Table 2 lists several gases with data as to the temperatures and pressures for which commercial condensers are designed. Internal combustion engines have limiting hot water temperatures of 125 F to 140 F for closed systems, and 110 F to 120 F for open systems, depending upon the quality of the cooling water. The cooling of such fluids as milk or wort has variable requirements and is usually done in counter-flow heat-exchangers in which the leaving circulating water is at a much higher temperature than is the leaving fluid.

The temperature range, once the hot water temperature is approximately known, depends upon:

Table 2. Condenser Design Data

Gas	Maximum Pressure Desired in Condenser	Gas Temperature in Condenser Deg F	Leaving Hot Water Temperature Deg F	
			Best Condenser Design	Average Condenser Design
Steam	28 in. vacuum	101.2	97	93
Steam	27 in. vacuum	115.1	110	105
Steam	26 in. vacuum	125.9	120	114
Ammonia	185 lb gage[a]	96.0	92	88
Carbon dioxide	1030 lb gage[a]	86.0	83	81
Methyl chloride	102 lb gage[a]	100.0	96	92
Dichlorodifluoromethane	117 lb gage[a]	100.0	96	93

[a]Head pressure

1. Maximum wet-bulb temperature at which the full quantity of heat must be dissipated.

2. Efficiency of the atmospheric cooling equipment considered.

Design Wet-Bulb Temperatures

The maximum wet-bulb temperature at which the full quantity of water must be cooled through the entire range is never, in commercial design, the *maximum* wet-bulb temperature ever known to exist at the location nor the *average* wet-bulb temperature over any period. The former basis would require atmospheric cooling equipment several times greater than normal size, and the latter would result during a large part of the time, in higher condenser water temperatures than those for which the plant was designed. For instance, the maximum wet-bulb temperature recorded in New York City is 88 F, and the July noon average for 64 years is close to 68 F. Yet in the years 1925 to 1934, inclusive, there were but 8 hours per year when the wet-bulb temperature reached 80 F or more, and there were 975 hours in the average summer (June to September inclusive) when the wet-bulb temperature was 68 F or above. As these 975 hours represent a third of the summer period, cooling equipment based upon the noon average July wet-bulb of 68 F would be inadequate. Commercial practice is to choose a wet-bulb temperature for air conditioning design purposes which is not exceeded during more than 5 to 8 per cent of the summer hours (75 F for New York City) with somewhat lower requirements for steam turbines and internal combustion engines. This difference is made because the heaviest load on an air conditioning plant is coincident with high wet-bulb temperatures, whereas the heaviest electric power demand occurs either in the winter or after nightfall in summer, when the wet-bulb temperature is low. Table 1, Chapter 7, shows design wet-bulb temperatures which will not be exceeded more than 8 per cent of the time in an average summer.

Knowing the hot water temperature and the wet-bulb temperature for which the equipment must be designed, the cold water temperature must be chosen to place the requirement within the effectiveness range of the type of atmospheric water cooling apparatus to be used. This effectiveness is expressed as the percentage ratio of the actual cooling effect to the maximum possible cooling effect. Since the wet-bulb temperature of the entering air is the equilibrium temperature to which the water could be cooled, the effectiveness of water cooling apparatus can be indicated thus:

$$\frac{(\text{hot water temperature} - \text{cold water temperature}) \times 100}{\text{hot water temperature} - \text{wet-bulb temperature of entering air}}.$$

Magnitudes of this effectiveness ratio will vary through wide limits in accordance with construction and conditions of operation. Values indicative of the commercial range of the effectiveness ratio are given in Table 3, although unusual designs may operate outside these ranges.

TABLE 3. EFFECTIVENESS OF ATMOSPHERIC WATER COOLING EQUIPMENT

Equipment	Cooling Effectiveness—Per Cent		
	Minimum	Usual	Maximum
Spray Ponds	30	40 to 50	60
Spray Towers	40	45 to 55	60
Natural Draft Deck Towers	50	60 to 75	90
Mechanical Draft Towers	50	60 to 75	90

From consideration of the factors which include the cooling range and design wet-bulb temperature, the quantity of water required can be calculated from the amount of heat to be dissipated. The normal amounts of heat to be removed from various processes of the cooling equipment are:

Compressor Refrigeration: 220 to 270 Btu per minute per ton. Usual practice is to assume: 250 Btu per minute per ton which is equivalent to 30 gal per degree Fahrenheit per minute per ton.

Steam Turbine Condensers: 950 to 980 Btu per pound of steam. Usual practice is to assume 970 Btu per pound of steam.

Steam Jet Refrigerating Condensers: 1030 to 1150 Btu per pound of steam. Exact value depends upon initial steam conditions.

Diesel Engine Jackets: 2500 to 4000 Btu per BHP per hour. Usual practice is to assume 3500 Btu per BHP per hour.

Natural Gas or Gasoline Engines: 4500 to 6000 Btu per BHP per hour. Usual practice is to assume 5000 Btu per BHP per hour.

Cooling Ponds

A natural pond is often used as a source of condensing water. The hot water should be discharged close to the surface at the shore line. Natural air movement over the surface of the water will cause evaporation and carry away heat. Because increased density due to the loss of heat causes the cooled water to sink to the bottom of the pond, the suction connection for intake water should be placed as far below the surface as possible, and at as great a distance from the discharge as practicable.

Spray Cooling Ponds

The spray pond consists of a basin, above which nozzles are located to spray water up into the air. Properly designed spray nozzles break up the water into small drops, but not into a mist because the individual drops must be heavy enough to fall back into the basin and not drift away with the air movement. The water surface exposed to the air for cooling is the combined area of all the small drops. Since the rate of heat removal by atmospheric water cooling is a function of the area of water exposed to the air, the difference in temperature between the water and the wet-bulb temperature of the air, the relative velocity of air and water, and the duration of contact of the air with the water, a much larger quantity of heat may be dissipated in a given area with the spray pond than with the cooling pond, because of (1) the speed with which the drops travel as they are propelled into the air and fall back into the water basin, (2) the increased wind velocity at a point above the surrounding structures or terrain, (3) the increased volume of air used, and (4) the vastly increased area of contact between air and water[3].

Spray pond effectiveness is increased by (1) elevating the nozzles to a higher point above the surface of the water in the basin, (2) increasing the spacing between nozzles of any one capacity, (3) using smaller capacity nozzles to decrease the concentration of water per unit area, and (4) using smaller nozzles and increasing the pressure to maintain the same concentration of water per unit area. Usual practice is to locate the nozzles from 5 to 7 ft above the edge of the basin, to supply from 5 to 12 lb pressure at the nozzles, using nozzles spraying from 20 gpm to 60 gpm each and spacing them so the average water delivered to the surface of the pond is from 0.5 to 0.7 gpm per square foot. Best results are obtained by placing the nozzles in a long relatively narrow area located broadside to the wind.

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[3]A.S.H.V.E. Research Paper—Design of Spray Cooling Ponds, by S. Hori, U. A. Patchett and L. M. K. Boelter (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, October, 1942, p. 624).

Spray ponds may be located on the ground, or they may be placed on roofs. To prevent excessive drift loss, or the carrying of entrained water beyond the edge of the pond by the air on the leeward side, louver fences are required for roof locations and for those ground locations where space is so restricted that the outer nozzles cannot be located at least 20 ft to 25 ft from the edge of the basin. Such fences usually are constructed of horizontal louvers overlapping so the air is forced to turn a corner in passing through the fence, and the heavier drops of water are thrown back, owing to their inertia. The louvers also restrict the flow of air, particularly at the higher wind velocities, and thus further reduce the possibility of water being carried off. The height of an effective fence should be equal to the height of the spray cloud. Louver boards are preferably of red gulf cypress or California redwood supported on cast-iron, steel or wood posts. Where building ordinances forbid the use of combustible materials, sheet metal is customarily used.

Algae growths, during warm weather, in cooling towers and spray ponds may be eliminated while the plant is in operation by the use of potassium permanganate. This chemical can be dissolved at the rate of 1 lb in 1¼ to 1½ gal of hot water. About 10 parts of permanganate should be used per million parts of cooling water. Enough of the permanganate solution should be added periodically to cause the water to have a pink color for a period of from 15 to 20 min. The best results are obtained when sufficient quantities are added periodically at intervals of several weeks, the time intervals being dependent upon local operating conditions. The chemical is non-poisonous and non-corrosive when used as directed.

Natural Draft Spray Type Towers

Where not more than 30,000 Btu per minute are to be dissipated, the natural draft spray type tower is a satisfactory apparatus. The word *tower* in this connection is somewhat of a misnomer as the apparatus is essentially a narrow spray pond with a high louver fence. As usually built, the nozzles spray down from the top of the structure and the distance from the center of the nozzle system to the fence on either side is not more than half the distance that the nozzles are elevated above the water basin. Heights range from 6 ft to 15 ft and the total width of a structure is not usually greater than its height. Spray towers occupy less space on small jobs than spray ponds of equivalent capacities because the towers have a capacity of from 0.6 gpm to 1.5 gpm per square foot of tower area. The louvers are continually wet, and so add to the surface of water exposed to the cooling air.

Natural Draft Deck Type Towers

In past years much of the atmospheric water cooling on refrigeration work has been done with natural draft deck type towers, which are also referred to as *wind* or *atmospheric* towers. These towers consist of heavy wooden or steel framework from 20 to 40 ft high and from 10 to 20 ft wide, having open horizontal lattice-work platforms or decks at regular intervals from top to bottom, and a catch basin at the foot. The hot water is distributed over the upper part of the structure by means of troughs, splash heads, or nozzles, and it drips from deck to deck down to the basin. The object of the decks is to arrest the fall of the water so as to present efficient cooling surfaces to the air, which passes through the tower parallel to the decks. The decks add to the area of water surface exposed to the air both by causing it to splash into fine droplets and to

spread over the deck surfaces, but since some deck designs furnish a resistance to air flow, too many decks may be a detriment.

To prevent the loss of water on the leeward side of the tower, wide splash boards are attached at regular intervals from top to bottom. In some towers these boards or louvers extend outward and upward, usually with lower edge (which is attached to the frame of the tower) somewhat below the level of the outer and upper edge of the next louver board below it. Some louvering designs add *secondary* louvering to these *primary* louvers. In such designs the primary louvers are generally broader and do not slope so sharply nor overlap one another, since their function is merely to return to the interior of the tower the water caught by the secondary louvering which extends upward from each primary louver's outer edge to the next primary louver above. The individual slats which comprise the secondary louvering may be arranged either vertically in an overlapping staggered manner with air passage between, or set horizontally with their flat surfaces sloping inward and spaced so their edges over-reach but permit free air passage. Secondary louvering reduces drift loss on this type of tower materially.

Efficiency of a deck tower is improved, within limits, by increased height, increased length, or increased width. The first two increase the area of water exposed to the wind, and the latter increases the time of contact of the air with the water. Performance is best when the wind blows crosswise to the tower and decreases with obliqueness of the angle until the effect of a lengthwise wind is virtually nil

Wind Velocities on Natural Draft Equipment

Since natural air movement is the prime requirement for a deck type tower, spray cooling tower, or spray pond, the apparatus must be designed to produce the desired cooling at times when the wind velocity is below average, when the wet-bulb temperature is at the maximum chosen for design, and when the plant is operating at full load. The apparatus must also, for best results, be located with its longest axis at right angles to the direction of the prevailing hot weather breeze. Table 1, Chapter 7, gives the average summer wind velocities and directions in representative cities. Natural draft cooling equipment should be designed to operate properly with *not more than one-half* of the *average* wind velocity, and in no case for a wind velocity of more than 5 mph. Natural draft equipment must not be obstructed by trees, buildings, or other wind deflectors.

Mechanical Draft Towers

Mechanical draft towers usually consist of box-like shells, constructed of wood, metal, or masonry, in which water is distributed uniformly at the top and falls to a collecting basin at the bottom. The inside of the tower may be filled with wood lattice work over which the water drips, or the water surface may be presented to the air by filling the entire inside of the structure with spray from nozzles.

Air is drawn through the tower by induced draft fans or propelled through it by forced draft fans. Since the air flows at a constantly sustained and comparatively high velocity (by contrast with natural draft equipment which varies in performance with every change of the wind's direction or velocity) the air and water are brought into contact under controlled conditions resulting in more efficient transfer of heat from a given quantity of water into a given volume of air. The least possible operating cost is achieved by that tower design which delivers the desired performance for the lowest total power input, both to fans

for air movement and to pumps for raising and distributing the water.

The effectiveness of a mechanical draft tower is improved by increasing height, area, or air quantity. Increasing the height increases the length of time the air is in contact with the water without affecting seriously the fan power required, but it increases the pumping power needed. Increasing the area while maintaining constant fan power increases the air quantity somewhat and because of lowered velocities it increases the time this air is in contact with the water. The surface area of water in contact with the air is increased in both cases. Increasing the air quantity decreases the time the air is in contact with the water, but, since a greater quantity is passing through, the average differential between the water temperature and the wet-bulb temperature of the air is increased, and this speeds up the heat transfer rate. Increased air quantities are obtained only at the expense of increased fan power, which increases approximately as the cube of the air quantity. Air velocities through mechanical draft towers vary from 250 to 450 fpm over the gross area of the structure.

Mechanical draft water cooling equipment may be set up inside buildings, where it usually draws its air supply from the general space in which it is installed, and discharges its exhaust air through a duct to the outside. Indoor cooling towers may be either of the wood-filled or the spray-filled type. In many cases where little height but considerable area is available, water is cooled in a spray-filled structure similar to an air washer, with the air passing horizontally through the apparatus and being discharged through a duct to the outside.

Cooling Tower Design

The method of design of equipment for energy transfer from water to an air-water vapor mixture is similar to that used for absorption equipment. Details of this procedure are available[4, 5, 6] and its application to the problem of the cooling tower operating at atmospheric pressure is illustrated by the following development. The nomenclature used is as follows:

a = over-all average wetted area (surface of water drops plus wetted tower surface) square feet per cubic foot of tower volume
c = specific heat of liquid water, Btu per (pound) (degree Fahrenheit).
e = effectiveness
ε = natural base
G = weight rate of flow of air, pounds of dry air per hour
h = enthalpy, Btu per pound of dry air
h_a = enthalpy of air-vapor mixture, Btu per pound of dry air
h'' = enthalpy of saturated air-vapor mixture at water temperature, Btu per pound of dry air
K = over-all energy unit conductance, Btu per (hour) (square foot over-all average wetted area) (Btu enthalpy difference per pound of dry air)
Ka = over-all rate coefficient, Btu per (hour) (cubic foot of tower volume) (Btu enthalpy difference per pound of dry air).
L = water rate, pounds per hour.
lm = logarithmic mean
S = average cross-sectional area of cooling tower for air flow, square feet.
t = water-main body temperature, degrees Fahrenheit
t_{wb} = wet-bulb temperature, degrees Fahrenheit.
V = tower volume, cubic feet.

Note: Subscripts 1 and 2 when used in equations refer to water entrance and exit sections respectively, for the counter-flow tower.

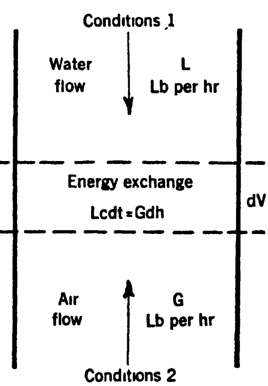

FIG. 4. SECTION OF TYPICAL COUNTER-FLOW TOWER

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[4]Principles of Chemical Engineering, by W. H. Walker, W. K. Lewis, W. H. McAdams and E. R. Gilliland (McGraw-Hill Co., 1937, p. 480).

[5]Absorption and Extraction, by T. K. Sherwood (McGraw-Hill Co., 1937, p. 91).

[6]Performance Characteristics of a Mechanically Induced Draft, Counterflow, Packed Cooling Tower, by A. L. London, W. E. Mason and L. M. K. Boelter (*A.S.M.E. Transactions*, January, 1940, Vol. 62, p. 41).

A section of a typical counter-flow tower is shown in Fig. 4. If the reduction in water rate due to evaporation within the volume is neglected, the energy balance for this differential section of the exchanger volume may be written as:

$$L\, c\, dt = G\, dh \tag{2}$$

The potential for net energy transfer due to heat and mass transfer from the water to the mixture in contact with it may be expressed with reasonable accuracy as the difference between the enthalpy of saturated air at the water temperature, h'', and the enthalpy of the main stream air vapor mixture[7], h_a. The rate of energy transfer is given by the expression:

$$Ka\,(h'' - h_a)\, dV \tag{3}$$

which equation defines the over-all rate coefficient, Ka; the latter being

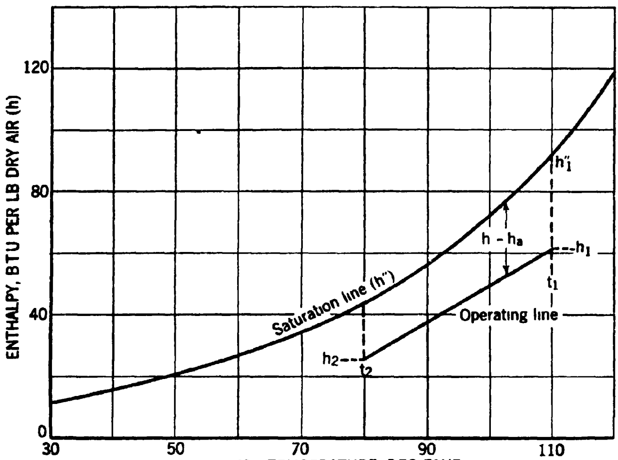

FIG. 5 · TEMPERATURE ENTHALPY DIAGRAM FOR AIR WATER VAPOR MIXTURE SHOWING THE OPERATING LINE FOR EXAMPLE 1

the product of the over-all energy unit conductance, K, and the ratio of the transfer surface to the exchanger volume, a.

Equations 2 and 3 are conveniently illustrated by means of the temperature-enthalpy diagram of Fig. 5. Equation 2 indicates that the succession of air and water states existing in the exchanger sections must combine to form a straight line (for L = constant) on the temperature enthalpy diagram. The slope of this operating line is $\frac{dh}{dt} = \frac{Lc}{G}$. Since the heat capacity of water is approximately unity, this slope is the ratio of the water to the air rate. Equation 3 indicates that the potential for energy transfer at any section is the difference between the enthalpy of saturated air at the main-body water temperature at that section and the enthalpy of the air stream in contact with that water. This potential is the difference in the ordinates of the saturation and operating lines for the water temperature at the plane in the tower which is under consideration.

[7]Determination of Unit Conductances for Heat and Mass Transfer by the Transient Method, by A. L. London, H. B. Nottage and L. M. K. Boelter (*Industrial and Engineering Chemistry*, April, 1941, Vol. 33, p. 467).

Combination of Equations 2 and 3 results in the expression:

$$Gdh = Ka\,(h'' - h_a)\,dV \tag{4}$$

Integrating this equation over the length of the exchanger:

$$\int_2^1 \frac{G}{Ka}\,\frac{dh}{h'' - h_a} = V \tag{5}$$

The integration of the left side of Equation 5 determines the tower volume required to achieve the desired energy exchange. This summation is readily accomplished for counter and parallel flow arrangements. G and Ka are usually independent of the tower volume and Equation 5 then becomes:

$$NTU = \int_2^1 \frac{dh}{h'' - h_a} = \frac{KaV}{G} \tag{6}$$

Where NTU is defined as the *Number of Transfer Units* and is a measure of the difficulty of the cooling process.

The integration is made numerically or graphically. In the graphical integration $\frac{1}{h'' - h_a}$ is evaluated as a function of h_a. This determination involves the use of the energy balance equation integrated from one section to the section in question. The area under the curve between any two abscissae is the number of transfer units required to change the air state from h_2 to h_1.

An approximate value for the number of transfer units can also be determined by a simple graphical method of direct construction on the temperature enthalpy diagram[8]. This method cannot be applied very satisfactorily to cooling towers as the operating range is small and the value of the NTU is near unity.

When the relationship between the enthalpy of the saturated air and the temperature is linear over the range of water temperatures involved, it can be shown[9] that the logarithmic mean of the terminal potentials, h_{lm}, is the correct driving force. This is true to a good approximation when the water cooling does not exceed 15 F. The approximation to linearity may be determined by inspection of Table 6, Chapter 1, or the temperature enthalpy diagram of Fig. 5. If the logarithmic mean is a valid potential, Equation 6 may be written:

$$\frac{h_1 - h_2}{\Delta h_{lm}} = \frac{KaV}{G} \tag{7}$$

and the need for the numerical integration for the determination of the tower volume is eliminated.

The over-all rate coefficient, Ka, must be known if the tower volume is to be determined. Experiments conducted on towers containing different packing construction have yielded some magnitudes of this coefficient, evaluated on an over-all basis. These data are presented in Fig. 6 as a function of the gas mass velocity through the packing, and apply only to the particular packing structure for which they were obtained. The over-all rate coefficient (Ka) may also be a function of the water rate,

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[8]Graphical Method of Determining Number Transfer Units, by T. Baker (*Industrial and Engineering Chemistry*, August, 1935, Vol 27, p. 977)

[9]Loc. Cit Note 4, p 79

since a reduction of the water rate may reduce the wetted area within the exchanger[10]. The results included in Fig. 6 probably represent magnitudes of *Ka* which were obtained for complete wetting. Within the cooling tower operating range the over-all rate coefficient for energy transfer is nearly the same numerically as the over-all rate coefficient for mass transfer. The conditions of test corresponding to the data presented in Fig. 6 are not well enough known in most cases to warrant recomputation of *Ka*. Therefore the magnitudes of the over-all rate coefficient for mass

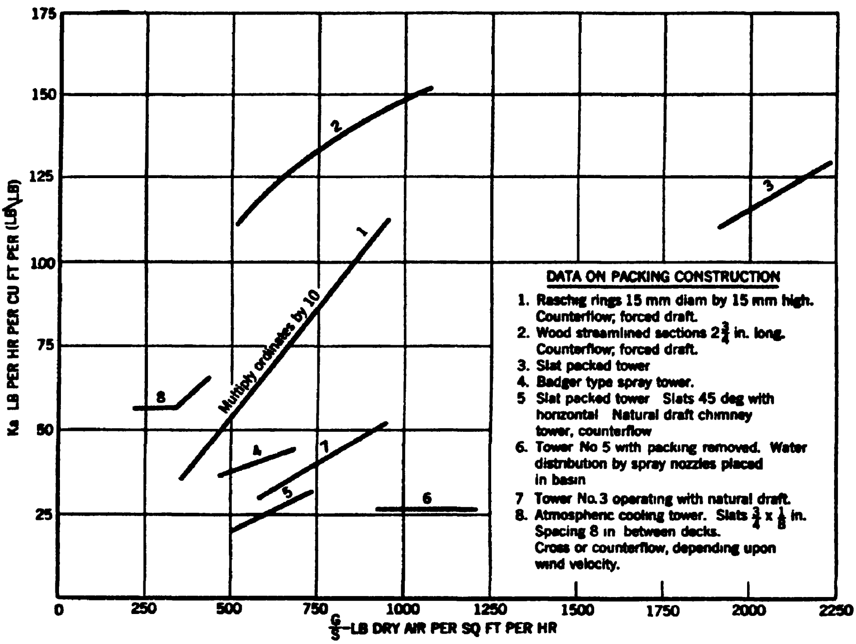

FIG. 6. UNIT CONDUCTANCES FOR VARIOUS TYPES OF PACKING CONSTRUCTION

transfer presented in Fig. 6 may be used directly in Equations 5 and 6, the units of *Ka* in these equations being Btu per (hour) (cubic foot per pound of dry air).

A typical design procedure is outlined in an illustrative example:

Example 1. The rate of air flow, arbitrarily assumed in the data given, is related to the tower volume by economic considerations. A balance between air rate and tower volume rests on consideration of the costs of producing air flow and of the tower construction[11]. A counter-flow forced draft cooling tower is to cool 36,000 lb of water per hour from an initial temperature of 110 F to a final temperature of 80 F. Air having an initial condition of 65 F dry-bulb and 58 F wet-bulb temperature will be forced through the tower counter to the direction of water flow at the rate of 30,000 lb of dry air per hour.

The cross-section of the tower is to be 8 ft x 8 ft and the packing is to be of the type producing a rate coefficient as indicated in curve No. 2 of Fig. 6. For this type of packing the average cross-sectional area for air flow will be 36 sq ft.

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[10] Loc. Cit. Note 5.
[11] Loc. Cit. Note 4, p. 142.

Solution:

Initial air enthalpy = 25.1 Btu per pound of dry air.

Final air enthalpy:

$$(h_1 - h_2) = \frac{Lc}{G}(t_1 - t_2)$$

$$(h_1 - 25.1) = \frac{36{,}000 \times 1}{30{,}000}(110 - 80)$$

$$h_1 = 61.1 \text{ Btu per pound dry air.}$$

A numerical integration (Table 4) is employed to determine the Number of Transfer Units (*NTU*) required. Temperature increments of 2 F are used between successive determinations of the quantity $\frac{1}{h'' - h_a}$. The energy balance indicates that the enthalpy

TABLE 4. NUMERICAL INTEGRATION FOR THE NUMBER OF TRANSFER UNITS

Water Temperature Interval Deg F	Mean Water Temperature Deg F	Mean Air Enthalpy, h_a Btu per Lb Dry Air	Saturated Air Enthalpy, h'' Btu per Lb Dry Air	Enthalpy Potential $h'' - h_a$	$\frac{\Delta h}{h'' - h_a}$
80–82	81	26.3	44.6	18.3	0.131
82–84	83	28.7	46.9	18.2	0.132
84–86	85	31.1	49.2	18.1	0.133
86–88	87	33.5	51.7	18.2	0.132
88–90	89	35.9	54.4	18.5	0.130
90–92	91	38.3	57.1	18.8	0.128
92–94	93	40.7	60.0	19.3	0.124
94–96	95	43.1	63.0	19.9	0.121
96–98	97	45.5	66.2	20.7	0.116
98–100	99	47.9	69.6	21.7	0.111
100–102	101	50.3	73.2	22.9	0.105
102–104	103	52.7	77.0	24.3	0.099
104–106	105	55.1	80.9	25.8	0.093
106–108	107	57.5	85.1	27.6	0.087
108–110	109	59.9	89.5	29.6	0.081
					1.723

increments corresponding to these temperature increments are:

$$\Delta h = \frac{Lc}{G}\Delta t = \frac{36{,}000}{30{,}000} \times 2 = 2.4 \text{ Btu per pound of dry air.}$$

The result of the integration is that the Number of Transfer Units required (*NTU*) = 1.72.

Unit gas mass velocity, $\frac{G}{S} = \frac{30{,}000}{36} = 830$ lb per (hour) (square foot average cross-sectional air flow area).

From Curve 2, Fig. 6, Ka = 138 Btu.

The tower volume required is:

$$V = \frac{G}{Ka}(NTU) = \frac{30{,}000}{138} \times 1.72 = 217 \times 1.72 = 374 \text{ cu ft.}$$

Height of the packed section is:

$$\frac{374}{8 \times 8} = 5.9 \text{ ft.}$$

The graphical solution for the Number of Transfer Units required for the desired performance is plotted in Fig. 7. $\frac{1}{h'' - h_a}$ is plotted as a function of h, and the area

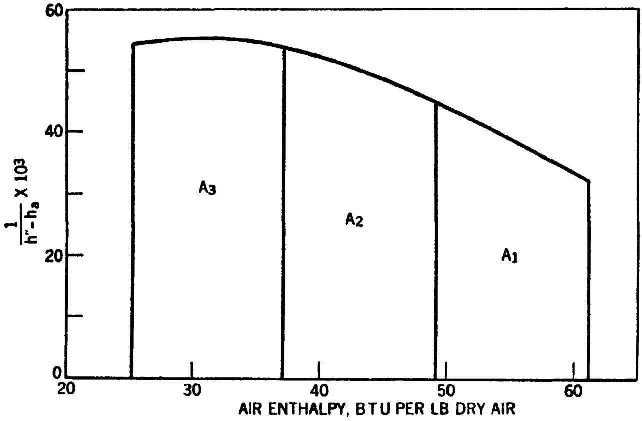

FIG. 7 GRAPHICAL INTEGRATION TO DETERMINE NUMBER OF TRANSFER UNITS REQUIRED FOR DESIRED OPERATING CONDITIONS OF EXAMPLE 1

under the curve from the initial enthalpy of the air, 25 1 Btu per pound, to the final enthalpy of the air, 61.1 Btu per pound is 1 72, the Number of Transfer Units required. The effect of the rapid decrease in potential due to the cooling of the water is indicated by comparison of area A_1, A_2, and A_3 of Fig. 7. Each represents the Number of Transfer Units required to achieve a water temperature reduction of about 10 F.

$$A_1 = 0.471\ NTU\ (110 \text{ to } 100 \text{ F})$$
$$A_2 = 0.595\ NTU\ (100 \text{ to } 90 \text{ F})$$
$$A_3 = 0.657\ NTU\ (90 \text{ to } 80 \text{ F})$$

The use of the logarithmic mean driving potential is illustrated by applying Equation 7 to Example 1:

$$NTU = \frac{(Ka)\ V}{G} = \frac{h_1 - h_2}{\Delta h_{lm}}$$

$$\Delta h_{lm} = \frac{(h_1'' - h_1) - (h_2'' - h_2)}{ln_e \frac{h_1'' - h_1}{h_2'' - h_2}} = \frac{30.7 - 18.4}{ln_e \frac{30\ 7}{18.4}} = 24$$

$$NTU = \frac{61.1 - 25.1}{24} = 1.5$$

The Number of Transfer Units required as determined by use of the logarithmic mean driving potential equals 1.5 which compares favorably with the correct magnitude, 1.72.

For small size counterflow spray towers intended for a low cooling range only considerable data[12, 13] are available on the performance of the spray system in terms of the logarithmic mean enthalpy potential. These data show the importance of proper nozzle arrangement and spray distribution within the tower. The greatest cooling effect is shown to be obtained in a region close to the nozzles; and the beneficial effects of packing added in the lower section of the tower follow as increasing the cooling range.

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[12]A.S.H.V.E. RESEARCH REPORT No. 1189—Performance Characteristics of a Forced Draft, Counterflow Spray Cooling Tower, by H. H. Niederman, E. D. Howe, J. P. Longwell, R. A. Seban and L. M. K. Boelter (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 413).

[13]A.S.H.V.E. RESEARCH REPORT No. 1240—Spray Nozzle Performance in a Cooling Tower, by L. M. K. Boelter and S. Hori (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p 309).

A variation of the design method according to Equation 6 is further possible[14] through the introduction of Equation 2 to yield

$$\int_{t_2}^{t_1} \frac{c\,dt}{h'' - h_a} = \frac{KaV}{L} \tag{8}$$

The advantage of this form is in the fact that the left side of the equation may be dealt with as a thermodynamic function only, while the right side depends upon the tower construction and operation. A series of performance curves may be determined from a graphic evaluation of the integral, which will serve either as a convenient means of analyzing experimental data to determine the conductance Ka, or as a rapid method of performance prediction, once the tower characteristic, $\frac{KaV}{L}$, is known. These performance curves may be expressed in terms of the following variables; approach to wet bulb, cooling range, L/G, and wet-bulb temperature of the entering air; and as such they will serve to accurately cover the tower performance in consistent terms.

The application of the foregoing design method to atmospheric towers is difficult because rate coefficients and flow conditions are not yet well defined for such equipment. If these are known, application of Equation 7 to sections of the tower small enough to justify use of the logarithmic mean potential will yield the tower volume required for each section. The sections must be taken perpendicular to the path of water flow. A correction to adjust the logarithmic mean potential, evaluated as for counter-flow, to the reduced effectiveness of cross-flow, has been derived for heat transfer and may be applied to this case[15].

Atmospheric towers operate with natural draft, produced in a vertical direction by the stack action of the tower structure, at zero velocity of the approach wind. Approach wind of sufficient magnitude (the magnitude depending on the baffle arrangement which is designed to reduce drift) will cause cross-flow augmenting the natural draft. An adequate design requires the consideration of both flow conditions.

Expression for Cooling Tower Performance

The performance of a cooling tower is described in terms of its effectiveness as an energy exchanger. The effectiveness is defined as the ratio of the energy actually exchanged to the energy available for exchange.

Effectiveness expressions:

Case 1. The slope of the operating line on the $t - h$ diagram exceeds the slope of the saturation line in the region of water temperatures considered.

$$e = \frac{h_1 - h_2}{h_1'' - h_2} \tag{9}$$

Case 2. The slope of the saturation line exceeds that of the operating line.

$$e = \frac{h_1 - h_2}{\frac{Lc}{G}(t_1 - t_{wb})} = \frac{t_1 - t_2}{t_1 - t_{wb}} \tag{10}$$

This equation represents the ***approach to wet-bulb.***

[14]Performance and Selection of Mechanical-Draft Cooling Towers, by Joseph Lichtenstein (*A S M E. Transactions*, October, 1943, Vol 65, No. 7, p 779).

[15]Heat Transmission, by W. H. McAdams (McGraw-Hill Co., 1933, p. 157).

Usual tower operating conditions conform to *Case 1*. Because of the curvature of the saturation line, operating conditions may present themselves to which neither *Case 1* nor *2* applies. Since a simple expression for the intermediate case is not available, the expression of *Case 1* may be utilized for the small number of operating conditions falling into the intermediate classification.

Make-Up Water

Since the atmospheric water cooling equipment performs its functions chiefly by evaporating a portion of the water in order to cool the remainder, there is a continual drain on the quantity of water in the system, and this loss must be replaced. Approximately 1 gal of water is lost for every 1000 gal of water cooled per degree of cooling range; so if 1000 gpm of water are cooled through a 10 F range, 10 gpm of water will be required to replace evaporated water. Replacement supply is usually regulated by a float control valve. Chemical treatment of the make-up water may be necessary to avoid excessive deposits in the condensers.

Winter Freezing

If atmospheric water cooling equipment is operated in freezing weather, the water may be cooled to freezing temperature so ice forms and collects until its weight causes damage. To obviate freezing during continued operation, the efficiency of the apparatus may be lowered. This is done on the spray pond and the spray cooling tower by reducing the quantity of water fed to the apparatus, thereby lowering the pressure at the nozzles and increasing the size of the drops produced. On the deck tower the upper system may be shut off and a secondary distribution system put in service midway down the height of the tower. The water will be kept above freezing because it will have shorter contact with the air. The mechanical draft tower can be protected by reducing the air flow through the tower, by stopping or reducing the speed of the fans, or by partially closing dampers.

If the system is operated intermittently in freezing weather, water in the basin may freeze and the expansion of the ice may do harm. Freezing during intermittent operation can be prevented only by draining the water basin when it is out of service. On small roof installations, a tank large enough to hold all the water in the system is often installed inside the building and the basin is drained into this by gravity, the pump suction being taken from this inside tank.

CHAPTER 27

Air Pollution

Classification of Air Impurities, Dust Concentrations, Air Pollution and Health, Occlusion of Solar Radiation, Smoke and Air Pollution Abatement, Dust and Cinders, Nature's Dust Catcher

THE particulate impurities which contribute to atmospheric pollution include carbon from the combustion of fuels, particles of earth, sand, ash, rubber tires, leather, animal excretion, stone, wood, rust, paper, threads of cotton, wool, and silks, bits of animal and vegetable matter, and pollen. Microscopic examination of the impurities in city air shows that a large percentage of the particles are carbon.

Gaseous pollution consists of gases originating in combustion, manufacturing and biochemical processes.

CLASSIFICATION OF AIR IMPURITIES

The most conspicuous sources of atmospheric pollution may be classified in various ways, as dusts, fumes and smoke. In Fig. 1, the classification is by particle size, but recent practice favors differentiation by method of formation. Thus, *dusts* are composed of particles produced by disintegration of larger material, as by crushing or grinding, whereas *fumes* are produced by condensation, and *smoke* consists of the finer carbon particles resulting from incomplete combustion. Similarly, *mists* are formed by the breaking up of liquids and *fogs* by condensation of vapors. There is as yet, however, no general agreement on these terms.

Dusts tend to settle without agglomeration, fumes to aggregate and smoke to diffuse. Particles which approach the common bacteria in size—from 1 to 10 microns—are difficult to remove from air and are apt to remain in suspension unless they can be agglomerated by artificial means. The term fly-ash is applied to solid ashy material, usually finely divided, that is a constituent of the effluent gases from coal-fired furnaces. Cinders denote the larger solid constituents which may be entrained by furnace gases.

Particles larger than 10 microns are unlikely to remain suspended in air currents of moderate strength, but settle out by gravity at speeds dependent upon the shape, size and specific gravity of the particle and upon the wind velocity. These larger particles are of major interest to the engineer in the solution of nuisance problems; on the other hand, it is mainly the smaller particles that are of hygienic significance. A notable exception to this size limitation in the latter case is the common hay-fever producing pollen such as that from ragweed. Pollen grains may be anything from fragments 15 microns or less in diameter to whole pollens 25 microns or more in size.

The lower limit of size of particle visible to the naked eye cannot be stated definitely. It depends not only upon the individual, but also upon the shape and color of the particle and upon the intensity of light. Under ideal conditions a particle 10 microns, or even less, may be recognized, while under less favorable conditions it may be difficult or even impossible to recognize a particle of 50 microns. The lower limit of visibility should, therefore, be considered as within a range (as above) of optimum conditions stated.

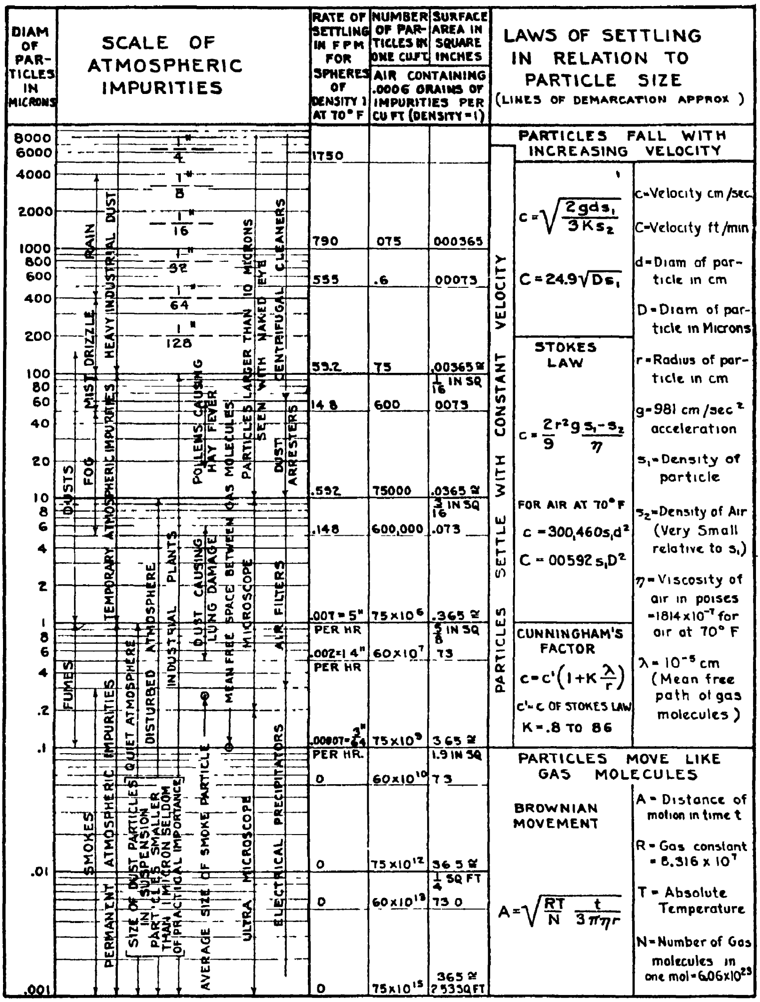

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FIG. 1. SIZES AND CHARACTERISTICS OF AIR-BORNE SOLIDS

Mineral particles, such as grains of sand, bits of rock, volcanic ash, or fly-ash, can be transported long distances under unusual circumstances. Thus, the dust storms of 1935 in the Kansas district resulted in vast amounts of fine top soil being thrown high into the air. Solar illumination as far east as Boston was affected noticeably and particles as large as 40 to 50 microns were actually carried half way across the continent before they settled out. In similar manner volcanic ash has been carried even further. It is not surprising, therefore, that fly-ash from furnace gases, cement dust and the like, can be carried for considerable distances and occasionally the engineer is confronted with the problem of removing such

TABLE 1. APPROXIMATE LIMITS OF INFLAMMABILITY OF SINGLE GASES AND VAPORS IN AIR AT ORDINARY TEMPERATURES AND PRESSURES[a]

Gas or Vapor	Lower Limit Volume in Per Cent	Higher Limit Volume in Per Cent	Gas or Vapor	Lower Limit Volume in Per Cent	Higher Limit Volume in Per Cent
Acetaldehyde	4.0	57.0	Hexane	1.2	6.9
Acetone	3.0	13.0	Hydrocyanic acid	5.6	40.0
Acetone[b]	2.5	......	Hydrogen	4.1	74.0
Acetylene	2.5	80.0	Hydrogen sulphide	4.3	45.5
Acetylene[b]	2.3	......	Illuminating gas	5.3	31.0
Allyl alcohol	2.4	......	Iso-amyl alcohol	1.2	......
Ammonia	16.0	27.0	Iso-butane	1.8	8.4
Amyl alcohol	1.2	......	Iso-butyl alcohol	1.8	......
Amyl chloride	1.4	......	Iso-pentane	1.3	......
Amylene	1.6	......	Iso-propyl acetate	1.8	7.8
Benzene	1.4	6 7	Iso-propyl alcohol	2.6	......
Benzine	1.1	......	Methane	5.3	14.0
Blast-furnace gas	35.0	74.0	Methane[b]	5 0	15.0
Butane	1.9	8.4	Methyl acetate	3.1	15.5
Butyl acetate (30 C)	1.7	7.6	Methyl alcohol	6.7	36.0
Butyl alcohol	1.7	......	Methyl bromide	13.5	14.5
Butylene	1.7	9.0	Methylbutyl ketone	1.2	8.0
Carbon disulphide	1.2	50.0	Methyl chloride	8.0	19.0
Carbon monoxide	12.5	74.0	Methyl cyclohexane	1.2	......
Croton aldehyde	2.1	15.5	Methyl ethyl ether	2.0	10.1
Cyclohexane	1.3	8.3	Methyl ethyl ketone	1.8	10.0
Cyclopropane	2.4	10.3	Methyl formate	5.0	23.0
Decane	0.7	......	Methyl propyl ketone	1.5	8.5
Dichlorethylene	9.7	12.8	Natural gas	4.8	13.5
Diethyl selenide	2.5	......	Nonane	0.8	......
Dioxan	2.0	22 0	Octane	0.9	......
Ethane	3 2	12 5	O-Xylene	1.0	6.0
Ethyl acetate	2.0	11.5	Paraldehyde	1.3	......
Ethyl alcohol	3.3	19 0	Pentane	1.4	7.8
Ethyl bromide	6.7	11.2	Propane	2.4	9.5
Ethyl chloride	4.0	14.8	Propyl acetate	1.8	8.0
Ethylene dichloride	6 2	15 9	Propyl alcohol	2.5	......
Ethylene	3 0	29.0	Propylene	2.0	11.1
Ethyl ether	1 8	36.5	Propylene dichloride	3.4	14.5
Ethyl formate	2.7	16.5	Propylene oxide	2.1	21.5
Ethyl nitrite	3.0	......	Pyridine (70 C)	1.8	12.4
Ethylene oxide	3.0	80.0	Toluene	1.3	6.7
Furfural (125 C)	2 1	......	Vinyl ether	1.7	27.0
Gasoline	1.4	6.5	Vinyl chloride	4.0	22.0
Heptane	1 0	......	Water gas	6 to 9	55 to 70

[a]Limits of Inflammability of Gases and Vapors, by H. F Coward and G. W. Jones (*U. S. Bureau of Mines, Bulletin* No. 279, 1939).

[b]Turbulent mixture.

material before the air in question is suitable for use in building ventilation.

The physical properties of the particulate impurities of air are summarized conveniently in the chart of Fig. 1.

In the case of gases, the objectionable features are physiological effects, inflammability (see Table 1) and odors.

Dust Concentrations

It is customary to report dust concentrations as grains per 1000 cu ft or milligrams per cubic meter (except for dusts that may cause pneumoconiosis, which are reported as so many particles per cubic foot of air). Gas concentrations are commonly recorded as milligrams per cubic meter

or as parts per million or as per cent by volume. Typical ranges in dust concentrations as now found in practical applications are given in Table 2.

The engineer frequently desires information regarding the effects of various concentrations of gases or dusts upon man, as the success of a particular installation may depend upon the maintenance of air which is adequately clean. At the present time there are several organizations working on this problem all of them publishing literature of various kinds.[1] References to books covering the hygienic significance, determination and control of dust are listed at the end of this chapter.

AIR POLLUTION AND HEALTH

The prevention of various diseases which result from exposure to atmospheric impurities is an engineering problem. It is important for the engineer to insure, by proper ventilation, suitable environments for working or for general living. If the equipment used is to be successful, it must operate automatically as in the modern air conditioned theater or railroad train.

In Table 3 are given data on permissible accepted standards for toxicity of gases and vapors which occur in industry. The prudent

TABLE 2. DUST CONCENTRATION RANGES IN PRACTICAL APPLICATIONS[a]

Application	Grains Per 1000 Cu Ft	Mgs Per Cu M
Rural and suburban districts	0 2 to 0 4	0 4 to 0 8
Metropolitan districts	0 4 to 0 8	0 9 to 1.8
Industrial districts	0 8 to 1 5	1 8 to 3 5
Dusty factories or mines	4 0 to 80 0	10 to 200
Explosive concentrations (as of flour or soft coal)	4000 to 8000	10,000 to 20,000

[a]1 grain per 1000 cu ft = 2.3 mgs per cubic meter; 1 oz per cubic foot = 1 g per liter.

engineer will design equipment using these bench marks as the upper limits of pollution. In general it is good practice to avoid recirculation of air which originally contains toxic substances. Obviously there are exceptions to this rule.

Affections of the respiratory tract are associated with exposure to thick dust, and may follow inhalation of practically any kind of insoluble and non-colloidal dust. Atmospheric dust in itself cannot be blamed for causing tuberculosis, but it may aggravate the disease once it has started [2]

The sulphurous fumes and tarry matter in smoke are more dangerous than the carbon. In foggy weather the accumulation of these substances in the lower strata may be such as to cause irritation of the eyes, nose, and respiratory passages. The Meuse Valley fog disaster will probably become a classic example in the history of gaseous air pollution. Released in a rare combination of atmospheric calm and dense fog, it is believed that sulphur dioxide and other toxic gases from the industrial region of the valley caused 63 sudden deaths, and injuries to several hundred persons.

Carbon monoxide from automobiles and from chimney gases constitutes another important source of aerial pollution in busy cities. During

[1]National Institute for Health, U. S. Public Health Service; Division of Labor Standards, U S Department of Labor; University of Toronto Medical School, Canada; Saranac Laboratories, Saranac Lake, N. Y.; Industrial Hygiene Foundation, Inc., Pittsburgh, Pa.; Harvard School of Public Health, Boston, Mass.; Haskell Laboratory, Wilmington, Del.; and the Departments of Health and of Labor in the United States and in various provinces of Canada.

[2]Physiological Response of the Peritoneal Tissue to Dusts Introduced as Foreign Bodies, by Miller and Sayers (*U. S. Public Health Reports*, 49:80, 1934).

heavy traffic hours and under atmospheric conditions favorable to concentration, the air of congested streets may contain enough *CO* to affect those exposed over a period of several hours, particularly if their activities call for deep and rapid breathing. In open air under ordinary conditions the concentration of *CO* in city air is insufficient to affect the average city dweller or pedestrian.

TABLE 3. ACCEPTED STANDARDS FOR TOXICITY OF GASES AND VAPORS[a]

SUBSTANCE	SINGLE EXPOSURES			MAX ALLOWABLE AVERAGE CONCENTRATION FOR REPEATED EXPOSURES
	Rapidly Fatal	Dangerous for from ½ to 1 Hour	Max. Safe Concentration for ½ to 1 Hour	
	(ppm)	(ppm)	(ppm)	(ppm)
Ammonia	5,000–10,000	2500	300	100
Amyl acetate	....	....	....	400
Aniline	....	....	105–160	5
Arsine	250	15	6	1
Benzene	190	....	31–47	100
Butyl acetate	....	....	....	400
Carbon bisulfide	4800	3200–3850	960–1600	20
Carbon dioxide	80,000–100,000	....	....	....
Carbon monoxide	4000	1500–2000	400	100
Carbon tetrachloride	10,000	....	1000	100
Chlorine	900	14–21	3.5	1
Dichlorbenzene	....	....	....	75
Dichlorethyl ether	....	....	....	15
Ether	....	....	....	400
Ethylene dichloride	....	....	....	100
Formaldehyde	....	....	....	20
Gasoline	....	....	....	1000
Hydrochloric acid	....	....	....	10
Hydrogen cyanide	270	110–135	45–54	20
Hydrogen chloride	1250–1750	1000–1350	40–90	10
Hydrogen fluoride	660	50–250	10	3
Hydrogen sulfide	1000–2000	360–500	200–300	20
Methanol	....	....	....	200
Methyl bromide	20,000–40,000	2000–4000	1000	....
Methyl chloride	150,000–300,000	20,000–40,000	7000	....
Nitrobenzene	....	....	200	5
Oxides of nitrogen	320–530	117–154	....	10
Phosgene	90	12.5	....	1
Phosphine	2000	400–600	100–200	2
Sulfur dioxide	400–500	150–190	50–100	10
Tetrachlorethane	7300	....	....	10
Tetrachlorethylene	....	....	....	200
Toluene and xylene	....	....	....	200
Trichlorethylene	7800	....	3700	200
Turpentine	....	....	....	200

[a]Adapted from The Prevention of Occupational Diseases, by R. R. Sayers and J. M DallaValle (*Mechanical Engineering*, Vol. 57, No 4, April, 1935); Safe Concentrations of Certain Common Toxic Substances used in Industry, by M. Bowditch, C K Drinker, P. Drinker, H A. Haggard and H Hamilton (*Journal of Industrial Hygiene and Toxicology*, Vol 22, No. 6, June, 1940); and other authoritative sources.

Occlusion of Solar Radiation

The loss of light, particularly the occlusion of solar ultra-violet light due to smoke and soot, is beginning to be recognized as a health problem in many industrial cities. Measurements of solar radiation in Baltimore[3] by actinic methods show that the ultra-violet light in the country was 50 per cent greater than in the city. In New York City[4] a loss as great as

[3]Effects of Atmospheric Pollution Upon Incidence of Solar Ultra-Violet Light, by J. H. Shrader, M. H. Coblentz and F. A. Korff (*American Journal of Public Health*, p. 7, Vol. 19, 1929).

[4]Studies in Illumination, by J. E. Ives (*U. S. Public Health Service Bulletin* No. 197, 1930).

50 per cent in visible light was found by the photo-electric cell method.

The aesthetic and economic objections to air pollution are so definite, and the effect of air-borne pollen can be shown so readily as the cause of hay-fever and other allergic diseases, that means and expenses of prevention or elimination of this pollution are justified.

SMOKE AND AIR POLLUTION ABATEMENT

Successful abatement of atmospheric pollution requires the combined efforts of the combustion engineer, the public health officer, and the public itself. The complete electrification of industry and railroads, and the separation of industrial and residential communities would aid materially in the effective solution of the problem.

In the large cities where the nuisance from smoke, dust and cinders is the most serious, limited areas obtain some relief by the use of district heating. The boilers in these plants are of large size designed and operated to burn the fuel without smoke, and some of them are equipped with dust catching devices. The gases of combustion are usually discharged at a much higher level than is possible in the case of buildings that operate their own boiler plants.

In general, time, temperature and turbulence are the essential requirements for smokeless combustion. Anything that can be done to increase any one of these factors will reduce the quantity of smoke discharged. Especial care must be taken in hand-firing bituminous coals. (See Chapter 8.)

Checker or alternate firing, in which the fuel is fired alternately on separate parts of the grate, maintains a higher furnace temperature and thereby decreases the amount of smoke.

Coking and firing, in which the fuel is first fired close to the firing door and the coke pushed back into the furnace just before firing again, produces the same effect. The volatiles as they are distilled thus have to pass over the hot fuel bed where they will be burned if they are mixed with sufficient air and are not cooled too quickly by the heat-absorbing surfaces of the boiler.

Steam or compressed air jets, admitted over the fire, create turbulence in the furnace and bring the volatiles of the fuel more quickly into contact with the air required for combustion. These jets are especially helpful for the first few minutes after each firing. *Frequent firings* of small charges shorten the smoking period and reduce the density. *Thinner fuel beds* on the grate increase the effective combustion space in the furnace, supply more air for combustion, and are sometimes effective in reducing the smoke emitted, but care should be taken that holes are not formed in the fire. A *lower volatile coal* or a higher gravity oil always produces less smoke than a high volatile coal or low gravity oil used in the same furnace and fired in the same manner.

The installation of more modern or better designed fuel burning equipment, or a change in the construction of the furnace, will often reduce smoke. The installation of a Dutch oven which will increase the furnace volume and raise the furnace temperature often produces satisfactory results.

In the case of new installations, the problem of smoke abatement can be solved by the selection of the proper fuel-burning equipment and furnace design for the particular fuel to be burned and by the proper operation of that equipment. Constant vigilance is necessary to make certain that the equipment is properly operated. In old installations the

solution of the problem presents many difficulties, and a considerable investment in special apparatus is often necessary.

Legislative measures at the present time are largely concerned with the smoke discharged from the chimneys of boiler plants. Practically all of the ordinances limit the number of minutes in any one hour that smoke of a specified density, as measured by comparison with a Ringelmann Chart (Chapter 34), may be discharged.

These ordinances do not cover the smoke discharged at low levels by automobiles, and, although they have been instrumental in reducing the smoke emitted by boiler plants, they have, in many instances, increased the output of chimney dust and cinders due to the use of more excess air and to greater turbulence in the furnaces.

Legislative measures in general have not as yet covered the noxious gases, such as sulphur dioxide, nor sulphuric acid fog, which are discharged with the gases of combustion. Where high sulphur coals are burned, these sulphur gases present a serious problem.

DUST AND CINDERS

The impurities in the air other than smoke come from so many sources that they are difficult to control. Only those which are produced in large quantities at a comparatively few points, such as the dust, cinders and fly-ash discharged to the atmosphere along with the gases of combustion from burning solid fuel, can be readily controlled.

Dusts and cinders in flue gas may be caught by various devices on the market, such as fabric filters, dust traps, settling chambers, centrifugal separators, electrical precipitators, and gas scrubbers, described in Chapter 28.

The cinder particles are usually larger in size than the dust particles; they are gray or black in color, and are abrasive. Being of a larger size, the range within which they may annoy is limited.

The dust particles are usually extremely fine; they are light gray or yellow in color, and are not as abrasive as cinder particles. Being extremely fine, they are readily distributed over a large area by air currents.

The nuisance created by the solid particles in the air is dependent on the size and physical characteristics of the individual particles. The difficulty of catching the dust and cinder particles is principally a function of the size and specific gravity of the particles.

Lower rates of combustion per square foot of grate area will reduce the quantity of solid matter discharged from the chimney with the gases of combustion. The burning of coke, coking coal, and sized coal from which the extremely fine coal has been removed will not as a general rule produce as much dust and cinders as will result from the burning of non-coking coals and slack coals when they are burned on a grate.

Modern boiler installations are usually designed for high capacity per square foot of ground area because such designs give the lowest cost of construction per unit of capacity. Designs of this type discharge a large quantity of dust and cinders with the gases of combustion, and if pollution of the atmosphere is to be prevented, some type of catcher must be installed.

NATURE'S DUST CATCHER

Nature has provided means for catching solid particles in the air and depositing them upon the earth. A dust particle forms the nucleus for

each rain drop and the rain picks up dust as it falls from the clouds to the earth. However, it was found in recent studies[5] that rain was not a good air cleaner of the material below about 0.7 micron.

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CHAPTER 28

Air Cleaning Devices

Air Cleaners, Dust and Lint, Classification of Air Cleaning Media, Viscous Impingement Type Cleaners, Automatic Viscous Filters, Dry Air Filters, Electric Precipitators, Performance and Testing, Selection and Maintenance, Safety Requirements, Adsorption of Vapors

FOR the purposes of this discussion, an air cleaner is defined as a device for capturing and removing solid matter from a stream of air. This solid matter includes fibrous material such as lint as well as particulate matter such as dust, fumes, smoke, cinders, etc. Air cleaning is distinct from air purification in that the latter deals with removal of harmful or unpleasant gases, vapors, or bacteria from an occupied space. Air pollution and purification are discussed in Chapters 2 and 27 and Air Sterilization in Chapter 36.

In general, air cleaners are not installed in buildings for the specific purpose of improving the health of the occupants. However, in some cases air borne solid matter does affect health and the removal of such matter from the air by means of cleaners is beneficial, such as pollen, house dust, and similar allergens which motivate attacks on persons having allergeric sensitivity[1]. The toxic elements in some war gases are in reality fine particles capable of air flotation.

It is known by experience that air cleaners greatly reduce the rate of dirt accumulation in buildings, but, in the present state of knowledge, an exact numerical expression for the cleaning effect of the filter cannot be given because of the many unknown or variable factors involved. For this reason air cleaners are usually selected on a basis of experience or judgment, guided by the results of various test procedures.

AIR CLEANERS

A typical air cleaner consists of a frame, which may be of metal, wood, cardboard, etc., and a filtering medium. The frame is designed to support the medium in a duct or chamber forming part of the air conditioning or ventilating system, so that the air passes through the medium while en route to spaces to be ventilated.

Unit Air Cleaners or Filters

Many air cleaners are available in the form of units of convenient size for handling during installation, cleaning or replacement. Such units are usually designated as filters or unit filters. A typical unit filter may be 20 in. square and from one to several inches thick, depending on the manufacture and proposed use. In large systems, a number of such units are installed adjacent to each other and collectively are called a bank of filters.

Air cleaners are commonly installed in the outdoor air intake ducts of buildings and, often, in the recirculating air ducts as well. Cleaners are logically placed ahead of heating or cooling coils and other air conditioning equipment in the system to protect them from dust. The character of the dust arrested by the filters in an air intake duct is likely to be mostly

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[1]Bronchial Asthma and Allied Allergic Disorders, by S. S. Leopold and C. S. Leopold (*Journal of the American Medical Association*, March 7, 1925, Vol. 84, p. 731-734).

particulate matter of a greasy nature, while lint may predominate in dust from within the building.

Settling chambers, air washers and electrostatic precipitators are also air cleaners. Settling chambers are used in boiler plants for capturing cinders, but they occupy too much space for general use in air conditioning or ventilating systems. Unless they are inordinately large, they are not effective in capturing small particles, since the air velocity is not sufficiently reduced to permit such particles to settle.

Air washers have generally become recognized for the purpose of adjusting the temperature and humidity of the air, rather than for cleaning air. It happens that insofar as dirtiness is concerned, carbon is the most troublesome dust. Carbon particles are likely to be greasy and since there is a natural repulsion between grease and water, the water spray in an air washer is not effective to a desirable degree in capturing them. However, air washers do capture considerable amounts of dust and lint which become sludge in the sump. Much of the cleaning action occurs at the eliminator plates where the dust particles are thrown against the film of water on the plates by their momentum.

The electrostatic precipitator is an effective available means of capturing the finer dust particles. This device does not entirely displace other types of air cleaners at the present time because of relatively high cost, large space requirements, and the fact that the air must enter and leave the apparatus in substantially parallel and straight flow.

Dust and Lint

Air-borne solid matter may fall into two classifications, namely, lint and dust or particulate matter. Some lint originates outdoors, as animal hair, vegetable fibers, etc., but much is generated within buildings by the wear and brushing of fabrics in the form of clothes, draperies, carpets, etc. Lint is comparatively easy to capture in an air filter because of its comparatively great length. So far as air filter performance and testing are concerned, lint is chiefly important because of its tendency to impede or stop the flow of air through the filter. In general, lint, if not captured, will accumulate in corners and under furniture in a building in areas of slight air motion and in some cases may seriously obstruct heating and cooling coils. Dust settles, or is precipitated by heat or air motion, upon furniture, fixtures and walls and the only satisfactory treatment is washing or re-painting. Dust is more difficult to capture than lint, and obviously, small particles are more difficult to capture than large ones. The air cleaning problem is complicated by the vast difference in size of dust particles, the range of which is shown in Fig. 1, Chapter 27.

Even if the discussion is limited to the range from 0.1 micron to 50 microns, that is between the smallest particle observable in the microscope and the smallest particle distinguishable to the naked eye, this range is so far outside the usual experience that it is difficult to visualize. If particles could be examined through a super microscope having a magnification of 250,000 diameters, a tobacco smoke particle of 0.1 micron would appear to be 1 in. in diameter, or approximately the size of a golf ball; a soft coal smoke particle 0.3 micron in diameter would appear like a baseball; a ragweed pollen grain 20 microns in diameter would appear 16.5 ft in diameter, while the 50 micron particle, just visible to the naked eye and able to pass through a 270 mesh screen, would appear to be 50 ft in diameter. Picturing this range in particle size from a golf ball to a sphere 50 ft in diameter will emphasize the difficulty of cleaning air

or of devising any single test to adequately measure the performance of air cleaning devices under all conditions of service.

It may be contended that the removal of the finer particles from ventilating air is relatively unimportant insofar as cleanliness in a house or building is concerned, since it is possible that such particles may remain suspended in the air and in large part be removed from the building without settling by the circulating air. However, since some of the particles are undoubtedly deposited by contact and by thermal precipitation, the ability to remove small particles is desirable in an air cleaner.

The migration of dust particles from a warm region toward a cool surface, to which they will adhere, is called thermal precipitation[2]. This fact is responsible for the *lath marks* often observed on walls or ceilings. The laths form barriers to the passage of heat so that the surface of the plaster in front of them is warmer than the surface between them. Dust is therefore deposited more rapidly between joists or laths than it is in front of them, resulting in the streaks observed. If the entire ceiling or wall is insulated, the differences in temperature across the surfaces cease to exist, the lath marks do not form, and the deposition of dust is much slower.

A laboratory apparatus has been designed to employ thermal precipitation in capturing dust particles for microscopic examination[3]. So far as is known thermal precipitation has not yet been used as a practical means of cleaning air.

CLASSIFICATION OF AIR CLEANING MEDIA

Air cleaners of such a variety of types have been used in the past that a single classification is difficult. Considering the wide diversification of materials and particle sizes to be removed and the varying requirements which have to be met in the field, it is natural that many kinds of air cleaning devices are used which cannot be shown satisfactorily in a simple outline. Classifications on three different bases are enumerated herewith:

1 Principle of air cleaning.
   - a. Viscous-impingement filters.
   - b. Dry filters
   - c. Washers.
   - d. Centrifugal devices.
   - e. Electrical precipitators.

2. Methods of servicing.
   - a. Automatic.
   - b. Non-automatic.
     - (1) Throw-away (replaceable elements).
     - (2) Manually cleaned in place (including one type of electrostatic).
     - (3) Removable for cleaning.

3 Classification according to application.
   - a. General air conditioning.
     - (1) Central cleaning system.
     - (2) Unit ventilator.
     - (3) Window installation
     - (4) Warm air furnace.
   - b. Removal of smoke and fumes from stack gases.
   - c. Collection of dusts from exhaust systems.

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[2]Dirt Patterns on Walls, by R. A. Nielsen (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1941, p. 247).

[3]Industrial Dust, by Philip Drinker and Theodore Hatch (McGraw-Hill Co., New York, N. Y.).

VISCOUS IMPINGEMENT TYPE CLEANERS

The medium in a viscous impingement type filter is usually a fiber pack for non-automatic types or a series of metal plates for automatic self-cleaning types. In either case, the medium is treated with a viscous substance, often an oil or grease, called the *adhesive* or the *saturant*, intended to retain dust particles which come in contact with it. Also, in either case, the arrangement is such that the air stream is broken up into many small air streams and these are caused to abruptly change direction a number of times in order to throw the dust particles, by momentum, against the adhesive. Several desirable characteristics of an adhesive for air cleaners of this type are:

1. Its surface tension should be such as to produce a homogeneous film or coating on the filter medium.
2. The viscosity should vary only slightly with normal changes of temperature.
3. It should prevent the development of mold spores and bacteria on the filter medium.
4. The liquid should have high capillarity, or ability to wet and retain the dust at all operating temperatures.
5. Evaporation should be slight.
6. It should be fire resistant.
7. It should be odorless.

Various fibrous materials have been used as filtering media in unit filters of the viscous impingement type. This includes glass fiber, steel wool, similar *wool* of non-ferrous metals, wire screen, animal hair, hemp fibers, and other materials. In such filters, the medium is often packed more densely on the discharge than on the approach-side in order to increase the dust holding capacity. This results in a selective arrestance of dust with the larger particles nearer the approach face. The arrangement also permits some penetration of lint into (but not through) the filter, so that the amount of lint which can be tolerated on the filter is also increased. Due to plane surface area the viscous impingement type filter, however, may be inferior to some dry types where the air carries a high percentage of lint.

The resistance of air filters obviously increases with the air flow through them. Face velocities of about 300 fpm and resistances in the range from 0.1 to 0.2 in. water, when the device is new and clean, are usual for ventilation system filters. Special filters with low resistances are available for use with gravity warm air furnaces and for other uses where only low pressure is available.

The resistance of these filters increases with dust or lint loading and it is the resistance due to this cause which ordinarily necessitates servicing. The rate of loading obviously depends upon the amount as well as the kind of dust in the air and for this reason, periods between servicing cannot be predicted. Manometers are often installed to indicate the pressure drop across filter banks and they serve to indicate when the filter requires cleaning. The pressure drop tolerated differs between operators and system designs. The resistance of a filter bank can be kept desirably low by periodically servicing some but not all of the units in the bank at one time, providing the difference in resistance between the clean and dirty filters is small.

The method of cleaning viscous impingement unit filters differs for different types of filters and kinds of dust. Much dry dust or lint can often be removed by rapping the filter.

Throw-away filters are constructed of inexpensive materials and are designed to be discarded after one use. The frame is frequently a combination of cardboard and wire.

Cleanable types usually have metal frames. Various cleaning methods have been recommended including: air jet, water jet, steam jet, washing in kerosene, and dipping in an oil. The latter may serve both to clean the filter and add the necessary adhesive.

Automatic Viscous Filters

In an automatic air filter, means are provided to remove the dust from the medium mechanically. Automatic filters with moving cloth media have been constructed. The media is supported on rollers and moves slowly and continuously across the air stream and then through some cleaning mechanism, such as a beater, a vacuum cleaner, or a brushing arrangement. Such filters, however, are not now in wide use, possibly because of mechanical difficulties and the rapid and sometimes permanent increase of resistance when oily matter is present in the air.

The medium in a typical automatic filter at present consists of a series of specially formed metal plates mounted on a pair of chains. The chains are mounted on sprockets located at the top and bottom of the filter housing, so that the filtering medium can be moved as a continuous curtain up one side and down the other side of the sprockets. The arrangement is such that, at the bottom, the medium passes through a bath of special oil which both serves to remove the dirt from the plates and acts as an adhesive when the cleaned plates next pass through the air stream. The plates forming the filtering medium or curtain usually overlap each other and due to their special shape many small air passages are formed between them. These air passages turn abruptly one or more times in order to give the impingement effect.

An electrically driven rotating device is usually supplied with an automatic filter. The device may be set to move the curtain periodically or a special switch, actuated by pressure drop, may be used to govern its motion. Such a switch will cause the gear to move the curtain when the resistance of the filter to air flow becomes excessive and will stop it when the resistance becomes sufficiently low.

In operation, the resistance of an automatic filter will remain approximately constant as long as proper operation is obtained. A resistance of $\frac{3}{8}$ in. water at a face velocity of 500 fpm is typical of this class.

DRY AIR FILTERS

As the name implies, adhesives are not used on dry air filters. The media in such filters are usually fabrics or fabric-like materials. Media of wool felt, cotton batting (both glazed and unglazed), celulose fiber and other materials have been used commercially.

The medium in a filter of this class is usually supported by a wire frame in the form of pockets or V-shaped pleats in order to increase the area exposed to the passage of air. A 2 ft square unit may contain from 15 to 30 sq ft of medium.

Dry air filters are likely to have a comparatively high lint-holding capacity on account of the large area of medium used. Wool felt media are troublesome to clean when impregnated with greasy dust and they are too expensive to discard frequently. Both vacuum cleaning and dry cleaning have been used for reconditioning wool felt filters.

ELECTRIC PRECIPITATORS

The fact that a particle exposed to an electric field will assume a charge and migrate toward one of the electrodes has been utilized for some years in boiler plants as a means of smoke abatement. More recently, means were developed whereby the phenomenon could be employed in air cleaning in connection with air conditioning without generating ozone in intolerable quantities. The air stream in a precipitator passes first through a relatively high-tension electric field, known as the ionizing field and then through a secondary field where the precipitation of the dust occurs. The arrangement is as shown in Fig. 1.

In a typical case, a potential of 12,000 volts may be used to create the

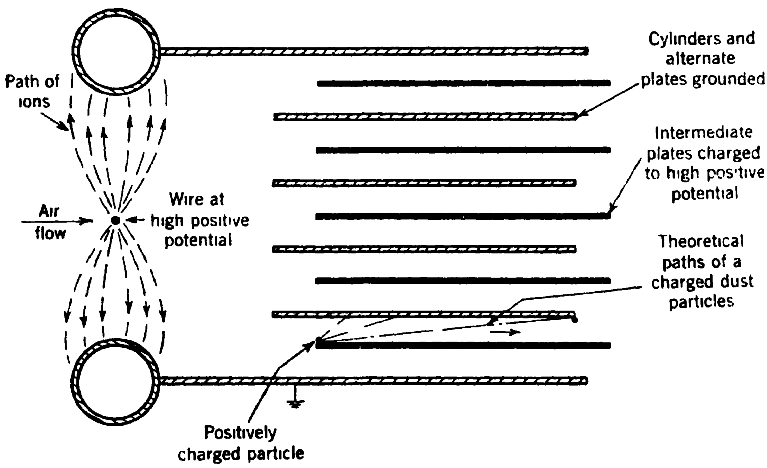

Fig. 1. Diagrammatic Cross-Section of Electrostatic Precipitator

ionizing field, and some 5000 volts between the plates upon which the precipitation of dust occurs. These voltages, which are capable of shock to personnel similar to that of a spark plug, necessitate some safety measures. A typical arrangement provides means for automatically making the unit inoperative when a door to the precipitator is opened. To resume operation the procedure necessitates closing the door and turning an electric switch, the latter of which should be located at a reasonable distance from the equipment. The voltages necessary for the operation of the precipitator are usually obtained from an alternating current building service line by means of a step-up transformer. Precipitation with alternating current is possible but is not nearly so effective; so the current is usually rectified by means of vacuum tubes. The transformer and tubes are collectively termed the power pack.

Only a very small amount of electric energy is necessary to operate an electric precipitator and the resistance to air flow through the device is practically negligible. Some care is necessary in arranging the duct approaches on the entering and leaving sides of precipitators to assure that the air flow is distributed uniformly over the cross-sectional area. The efficiency of the precipitator is sensitive to air velocity and the device itself has much less tendency to rectify the air stream than filters, which have much higher resistances.

Electric precipitators are available in both automatic and non-automatic types. The plates of non-automatic precipitators are commonly coated with a light oil as an adhesive. Cleaning is accomplished with a water hose and, for this reason, the bottom of the equipment is made water tight and provided with a drain. In one automatic type, precipitation units are mounted on chains and are alternately dipped in oil and exposed to the air stream with an action similar to that of an automatic impingement filter. An arrangement of sliding contacts maintains the necessary electric circuits.

PERFORMANCE AND TESTING

The rating of an air cleaner is the air flow for which it is designed expressed in cubic feet per minute. Face velocity is defined as the average velocity of the air entering the cleaner, and it is determined by taking the air flow and dividing it by the area of the duct connection to the cleaner in square feet. Cleaners are often rated at a face velocity in the range of 250 to 500 fpm. The resistance of an air cleaner to air flow is usually measured in inches water. The resistances of filters when new and clean and when operated at rated capacity are generally available from the manufacturer (see *Catalog Data Section*).

The ability of air cleaners to clean air is called the efficiency or the arrestance, and may be denoted by the symbol E. The efficiency of an air cleaner differs with the size and nature of the dust on which the cleaner operates. Obviously, large particles and lint are more easily captured than minute particles which are small in all dimensions. The efficiency of an air cleaner, algebraically expressed, is:

$$E = \frac{D_1 - D_2}{D_1} \tag{1}$$

where

D_1 = amount of dust per unit volume in uncleaned air.
D_2 = amount of dust per unit volume in cleaned air.

Several methods have been investigated for evaluating D_1 and D_2. The particle count method is no longer used for efficiency evaluations except in rough field measurements or in investigation of filter performance on specific and comparatively large particles such as pollen. Dust particles can be captured on microscope slides by means of one of the various kinds of impingement devices. The process is useful if an inspection and analysis of dust is desired, but particle counting is not sufficiently precise for evaluating the efficiency of a cleaner operating on a heterogeneous dust.

The weight method of evaluating efficiency has found wide utility and was recognized by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and incorporated in a code[4]. For this test, a known weight of a prepared dust is injected into air supplied to the filter and the quantity of dust in the cleaned air is determined by extracting and weighing the dust from a known volume of the cleaned air. Dust extraction from the air is accomplished by drawing the air through a porous crucible or thimble by means of a high vacuum.

The *dust-spot* or *blackness* test for cleaner efficiency was developed at the *National Bureau of Standards*[5]. The test consists of drawing samples

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[4]A.S H V.E Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work (A.S H.V.E. TRANSACTIONS, Vol. 39, 1933, p 225).

[5]A Test Method for Air Filters, by Richard S. Dill (A S.H V.E. TRANSACTIONS, Vol. 44, 1938, p. 379).

of cleaned air and of uncleaned air through filter papers simultaneously. The ratio of the areas of paper through which the air samples are drawn and the ratio of the amount of air drawn through the papers are adjusted during successive trials to yield spots of approximately equal blackness on the papers. The ratios of the areas and of the volumes of the air samples are then indicators of the filters effectiveness. A special photometer is provided for comparing the blackness or opacity of the papers by transmitted light. For tests of ordinary air filters by this method, a dust is injected into the air stream. The dust consists of precipitated smoke particles from a Cottrell precipitator used in a local power plant for smoke abatement. For tests of electrostatic air cleaners, no dust is added to the air. Tests are commonly made with the dust existing in the air at the location of the installation on a clear day. Some specifications for this type cleaner have required that in dust spots of equal area the downstream spot shall not be any dirtier than the upstream spot when 10 times as much air is drawn through it as is drawn through the upstream

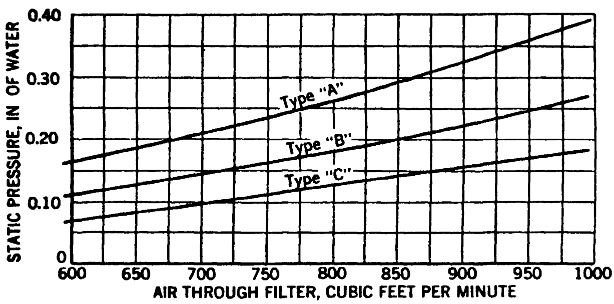

Fig. 2. Resistance to Air Flow of Typical Unit Air Filters

spot. When this condition is met the cleaner is said to have an efficiency or arrestance of 90 per cent or better on atmospheric air.

Dust-holding capacity is defined as the amount of dust which a filter can retain and have a resistance less than some arbitrary value. The term applies only to non-automatic air cleaners. Determination of dust-holding capacity is an objective of each test under the A.S.H.V.E. Standard Code[6]. Curves are obtained during such tests to show the relation between dust load and resistance. Typical curves are shown in Fig. 2. Type A is a dense pack used in bacterium control; Type B is a medium pack used for general ventilation work; and Type C is a low resistance unit, for use where low resistance is the important factor and maximum cleaning efficiencies are not essential.

At the *National Bureau of Standards* two injectors are provided on the air cleaner testing apparatus. One injector is used to contaminate the air stream with Cottrell precipitate, previously described. This dust is used to make both efficiency determination and dust-holding capacity tests. The other injector contaminates the air stream with cotton linters with which lint-holding capacity tests are made. The curves in Fig. 3 illustrate the difference in the characteristics of two filters, one a viscous-impingement type and the other a dry filter with a celulose fiber medium. The two injectors can be operated either separately or simultaneously.

[6]Loc. Cit. Note 4.

A total dust deposit of 4 per cent cotton linters and 96 per cent Cottrell precipitate gives a deposit on a filter closely resembling those that occur in Washington, D. C.

SELECTION AND MAINTENANCE

If effectiveness in arresting dust were always the primary consideration, an electrostatic cleaner might be used for all air cleaner installations. Where the dust load is heavy, a filter bank may be installed ahead of the precipitator. At the present time electrostatic equipment is comparatively expensive and bulky and is therefore only used when the expense is justified by the need for the cleanest air possible.

The advantage of the automatic impingement type filter consists in the small amount of attention which it requires. Such devices are therefore to be recommended where labor is scarce or where reliable and frequent

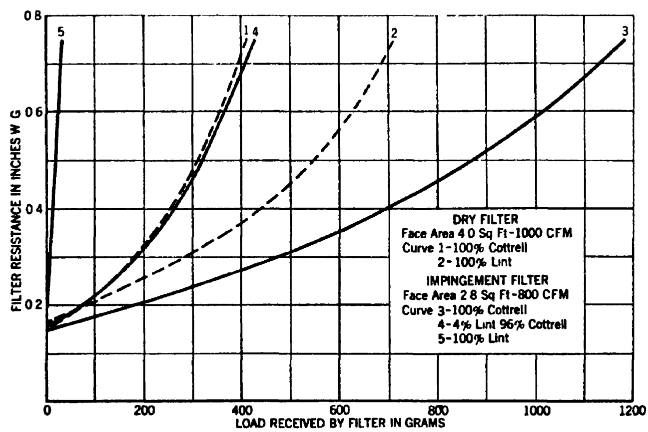

FIG. 3. DUST AND LINT HOLDING CAPACITY OF TWO FILTERS

attention to filters cannot be assumed. This type of equipment is not any better in dust arrestance than some unit filters, and it ranks next to precipitators in first cost.

Unit filters constitute the majority of air cleaners now in use, and some choice is possible between the types available. Where lint in an eminently dry state predominates, a dry filter obviously may be preferable to other types on account of its lint-holding capacity. If the lint is greasy or if oil vapor exists in the air, the dry filter if it is of the cleanable type may be troublesome, since grease tends to make it difficult to clean. The cleaning difficulty is avoided if a throw-away type of medium is used. Some dry filters are capable of high efficiencies, compared to other unit filters on fine particles, but their dust-holding capacity for such dust may be inferior to that of the viscous impingement type.

Viscous impingement unit filters represent the general type of air cleaner now in use. They have approached standards in size and their over-all dimensions are small when compared with their ratings.

Throw-away units are often installed in series so that the one in front, which usually becomes plugged with lint, can be discarded after which the downstream unit is moved to the front and replaced by a new unit.

Viscous impingement unit filters do not have efficiencies as high as can be expected with some other types of unit filters, but their first cost and upkeep are generally lower, whether of the cleanable or the throw-away type. They require more careful attention than the automatic oil type if the resistance is to be maintained within reasonable limits.

Safety Requirements

An investigation of safety ordinances should be made by the engineer when the installation of an air cleaner of any considerable size is contemplated. It is possible that a combustible filtering media may not be permitted in accordance with some existing local regulations. Combus-

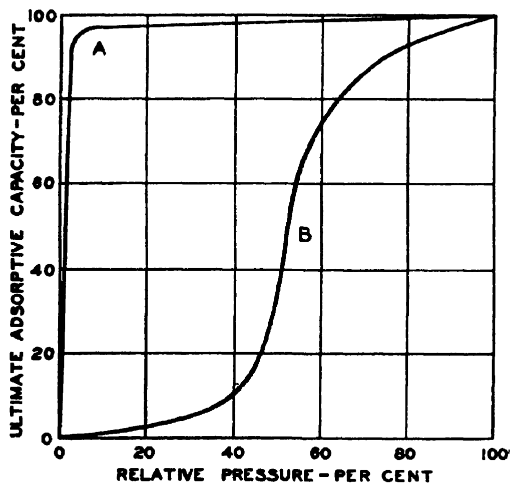

Fig. 4. Adsorption at Relative Pressures by Charcoal of a Typical Organic Gas (A) and Water Vapor (B)

tion of dust and lint on a filtering medium is possible, though the medium itself may not burn.

ADSORPTION OF VAPORS OTHER THAN WATER

Many of the foreign gases in the atmosphere are selectively adsorbed by charcoal. Included are many of the organic gases, such as those emanating from animals and people, some of the gaseous constituents of combustion, alcohols, ketones, esters, and gaseous products of putrefaction.

The selective action of charcoal with a mixture of water vapor and organic gases is shown in Fig. 4.

Charcoals differ widely in their adsorptive capacity. Those which have marked adsorption characteristics, such as properly prepared coconut shell charcoals, are sometimes called *activated charcoals* or *activated carbon*. These materials can adsorb approximately 50 per cent of their own weight in many organic gases at 70 F. The charcoal may be used for a long time by reactivation at high temperatures, under which condition it gives up the adsorbed gases. Temperatures of approximately 1000 F are desirable for reactivation. Charcoals for use in air handling systems should be

able to stand physical handling, including reactivation without excessive loss by breakage or dusting.

As applied in air handling systems, the charcoal is placed in perforated metal containers which are grouped in frames and set in the air stream. The percentage removal of an organic gas, such as carbon tetrachloride, is 95 per cent or above when placed in intimate contact with the carbon at 70 F. In commercial apparatus there may be a by-pass effect which depends on the physical arrangement of the charcoal containers. This by-passing reduces the percentage removed in the total gas passing through the adsorber. Resistance to air flow is usually selected within the general range of resistance of impingement filters.

The required quantity of recirculated air to be treated is determined by the requirements for contaminant-free air, minus the outdoor air, divided by the fraction denoting the percentage removal of the gas in question in the adsorber bank which is to be used.

Adsorbers may be applied to reduce objectionable gases entering through the outdoor air inlet. They may also be used to reduce the odors caused by exhausts from processing. Adsorber beds, in all cases, should be protected from dust, free oil and grease.

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CHAPTER 29

Fans

Classification, Performance, Fan Efficiency, Characteristic Curves, System Characteristics, Selection of Fans, Fan Designations, Control, Motive Power

IN heating, ventilating, and air conditioning practice, fans and blowers are used to produce air flow. All fans or blowers are classified according to the direction of air flow through the fan with relation to the axis of rotation and are either of the (1) *axial flow* or *propeller* type, in which the flow is parallel to the axis, or (2) *radial flow* or *centrifugal* type, in which the flow is at right angles to the axis.

Axial flow fans are made in various designs and sizes. The form and number of blades may vary for impellers of the same or different diameters. The blades may be of uniform thickness and made of cast or sheet metal, and either flat or cambered or of screw form; or they may vary in thickness, in the latter case usually being designed to conform to so-called *airfoil* sections of known characteristics, similar to those which have been developed for airplane propellers. Likewise, blade angle, or the angular relation of the blades to the plane of rotation, varies over a wide range For operation against comparatively high pressures, it is customary to resort to enlarged hubs in proportion to fan diameter (large hub ratio) and correspondingly short blade length. The term *disc fan* has sometimes been loosely applied to such large hub fans, though it has long been generally used in connection with any propeller fan of comparatively short axial length whose blades are relatively flat; in other words, for fan wheels which occupy a space which is more or less disc-shaped. Single stage air foil axial flow fans can operate at pressures up to 4 in. at a quite low noise level and, at considerably higher pressures, they are comparable in regard to noise to the centrifugal fan type.

Radial flow or *centrifugal fans* used in heating, ventilating, and air conditioning practice, are in general of two types; one with forward-sloped blades, and the other with backward-sloped blades. Many modifications may be made in the proportions, the curvature, and the slope or angularity of the blades. The slope or angularity of the blades determines the operating characteristics of a fan; a forward-curved or sloped blade is found in a fan having low speed operating characteristics, while a backward-curved or sloped blade is found in a fan having higher speed operating characteristics.

A wide variety of heating, ventilating, and air conditioning systems creates a wide variety in the demands that have to be met by the fan or blower. The requirement may be to move small or large quantities of air against little or no resistance, or to move small or large quantities of air against higher resistance. Between the two limits innumerable specific requirements must be met. Although fans of any class in either of the two general types can in general be made to perform the same duty, mechanical difficulties, space and noise limitations, efficiency and power limitations usually determine the selection.

FAN PERFORMANCE

Fans of all types follow certain laws of performance which are useful in predicting the effect upon performance of changes in the conditions of operation, the duty required of the installation, or the size of the equip-

ment due to the space, power, or speed limitations. The following laws in which Q = air volume and P = static, velocity or total pressure, apply to all types of fans:

1. Variation in Fan Speed:
   Constant Air Density—Constant System
   (*a*) Q: Varies as fan speed.
   (*b*) P: Varies as square of fan speed.
   (*c*) Power: Varies as cube of fan speed.

2. Variation in Fan Size:
   Constant Tip Speed—Constant Air Density
   Constant Fan Proportions—Fixed Point of Rating
   (*a*) Q: Varies as square of wheel diameter.
   (*b*) P: Remains constant.
   (*c*) RPM: Varies inversely as wheel diameter.
   (*d*) Power: Varies as square of wheel diameter.

3. Variation in Fan Size:
   At Constant RPM—Constant Air Density
   Constant Fan Proportions—Fixed Point of Rating
   (*a*) Q: Varies as cube of wheel diameter.
   (*b*) P: Varies as square of wheel diameter.
   (*c*) Tip Speed: Varies as square of wheel diameter.
   (*d*) Power: Varies as fifth power of diameter.

4. Variation in Air Density:
   Constant Volume—Constant System
   Fixed Fan Size—Constant Fan Speed
   (*a*) Q: Constant.
   (*b*) P: Varies as density.
   (*c*) Power: Varies as density.

5. Variation in Air Density:
   Constant Pressure—Constant System
   Fixed Fan Size—Variable Fan Speed
   (*a*) Q: Varies inversely as square root of density.
   (*b*) P: Constant.
   (*c*) RPM: Varies inversely as square root of density.
   (*d*) Power: Varies inversely as square root of density.

6. Variation in Air Density:
   Constant Weight of Air—Constant System
   Fixed Fan Size—Variable Fan Speed
   (*a*) Q: Varies inversely as density.
   (*b*) P: Varies inversely as density.
   (*c*) RPM: Varies inversely as density.
   (*d*) Power: Varies inversely as square of density.

Examples 1 to 4 illustrate the application of the preceding fan laws.

Example 1. A certain fan delivers 12,000 cfm at a static pressure of 1 in. of water when operating at a speed of 400 rpm and requires an input of 4 hp. If in the same installation 15,000 cfm are desired, what will be the speed, static pressure, and power?

$$\text{Speed} = 400 \times \frac{15{,}000}{12{,}000} = 500 \text{ rpm}$$

$$\text{Static pressure} = 1 \times \left(\frac{500}{400}\right)^2 = 1.56 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{500}{400}\right)^3 = 7.81 \text{ hp.}$$

Example 2. A certain fan delivers 12,000 cfm at 70 F and normal barometric pressure (density 0 075 lb per cubic foot) at a static pressure of 1 in. of water when operating at

400 rpm, and requires 4 hp. If the air temperature is increased to 200 F (density 0.0602 lb) and the speed of the fan remains the same, what will be the static pressure and power?

$$\text{Static pressure} = 1 \times \frac{0.0602}{0.075} = 0\ 80 \text{ in}$$

$$\text{Power} = 4 \times \frac{0.0602}{0.075} = 3.20 \text{ hp}$$

Example 3. If the speed of the fan of *Example 2* is increased so as to produce a static pressure of 1 in. of water at the 200 F temperature, what will be the speed, capacity, and power?

$$\text{Speed} = 400 \times \sqrt{\frac{0.075}{0.0602}} = 446 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \sqrt{\frac{0.075}{0.0602}} = 13{,}392 \text{ cfm (measured at 200 F)}$$

$$\text{Power} = 4 \times \sqrt{\frac{0.075}{0.0602}} = 4.46 \text{ hp.}$$

Example 4. If the speed of the fan of the previous examples is increased so as to deliver the same weight of air at 200 F as at 70 F, what will be the speed, capacity, static pressure, and power?

$$\text{Speed} = 400 \times \frac{0.075}{0.0602} = 498 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \frac{0.075}{0.0602} = 14{,}945 \text{ cfm (measured at 200 F)}$$

$$\text{Static pressure} = 1 \times \frac{0.075}{0.0602} = 1.25 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{0.075}{0.0602}\right)^2 = 6.20 \text{ hp.}$$

Laws of Homologous Fans

The laws applying to different sizes of homologous fans are as follows:

Capacity varies as the ratio of size cubed, times the ratio of the rpm

Pressure varies as the ratio of size squared, times the ratio of the rpm squared.

Horsepower varies as the ratio of the size to fifth power, times the ratio of the rpm cubed.

Example 5. Assuming that a fan with a 36 in. diameter blast wheel will deliver 12,000 cfm at 70 F at 1 in. static pressure, requiring 4.0 brake hp when operating at 400 rpm, what is the capacity, pressure and horsepower of a homologous fan having a 45 in wheel at the same speed?

$$\text{Capacity} = \left(\frac{45}{36}\right)^3 \times \left(\frac{400}{400}\right) \times 12{,}000 = 23{,}400 \text{ cfm}$$

$$\text{Static Pressure} = \left(\frac{45}{36}\right)^2 \times \left(\frac{400}{400}\right)^2 \times 1 = 1\ 56 \text{ in}$$

$$\text{Horsepower} = \left(\frac{45}{36}\right)^5 \times \left(\frac{400}{400}\right)^3 \times 4 = 12.2 \text{ hp}$$

FAN EFFICIENCY

The efficiency of a fan may be defined as the ratio of the horsepower output to the horsepower input.

The horsepower output is expressed by the formula:

$$\text{Air Horsepower}^1 = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6356} \tag{1}$$

When the static pressure is used in the computation in place of total pressure it is assumed that this represents the useful pressure and that the velocity pressure is lost in the piping system and in the air which leaves the system. Since in most installations a higher velocity exists at the fan outlet than at the point of delivery into the atmosphere, some of the velocity pressure at the fan outlet may be utilized by conversion to static pressure within the system, but, owing to the uncertainty of friction losses which occur at the places where changes in velocity take place, the amount of velocity pressure which is actually utilized is seldom known, and the static pressure alone may best represent the useful pressure. In the standards for published capacity tables as adopted by the *National Association of Fan Manufacturers*, the term *static pressure* refers to the true resistance to air flow. Such tables charge both the inlet and outlet velocity of the fan to the fan performance, and may be used directly where the static pressure of the system as calculated represents only the actual resistance to flow of the air.

The efficiency based upon static pressure is known as the static efficiency and may be expressed as follows:

$$\text{Static Efficiency}^1 = \frac{\text{cfm} \times \text{static pressure in inches of water}}{6356 \times \text{Horsepower input}} \tag{2}$$

Different fans may develop the same capacity against the same static pressure and with the same power input, and therefore operate at the same static efficiency, while maintaining different outlet velocities. Where a high outlet velocity is desirable or can be utilized effectively, the static efficiency fails to be a satisfactory measurement of the performance. In many applications of propeller fans, air is circulated without encountering resistance and no static pressure is developed. The static efficiency is zero and its calculation is meaningless. Because of such situations where the static efficiency fails to indicate the true performance, many engineers prefer to base the calculation of efficiency upon the total pressure. This efficiency is variously known as the total, or mechanical efficiency, and may be expressed as follows:

$$\text{Mechanical or Total Efficiency}^1 = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6356 \times \text{Horsepower input}} \tag{3}$$

CHARACTERISTIC CURVES

In the operation of a fan at a fixed speed the static and total efficiencies vary with any change in the resistance which is imposed. With different designs the peak of efficiency occurs when the fans deliver different percentages of their wide-open capacity. Variations in efficiency accompany variations in pressures and power consumption which are characteristic of the individual designs and which are influenced particularly by the shape and angularity of the blades. Such variations in pressure, power, and efficiency are shown by characteristic curves.

Characteristic curves of fans based upon tests performed in accordance with the Standard Test Code for Centrifugal and Axial Fans[1] prepared jointly by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers* are generally

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[1]See Standard Test Code for Centrifugal and Axial Fans, 3rd Edition, 1938, Bulletin No. 103, *National Association of Fan Manufacturers*

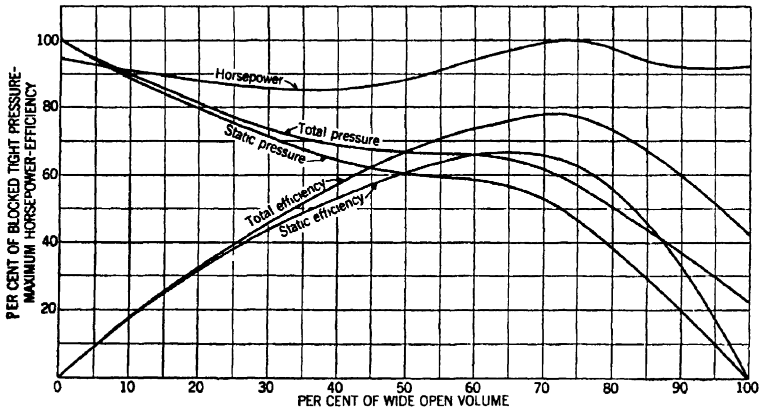


FIG. 1. OPERATING CHARACTERISTICS OF AXIAL FLOW AIRFOIL TYPE FAN

plotted to show total and static pressure, mechanical and static efficiency, and horsepower in relation to air delivery as a basis. Results may also be plotted against per cent of wide open volume or discharge. Examples of fan performance curves are shown in Figs 1, 2 and 3.

In the selection of all but very small fans, power consumption is usually a major consideration. It must be borne in mind that the horsepower at peak efficiency alone may be misleading, as actual operation is apt to occur at some point on the pressure-volume curve varying considerably from that specified, due to inaccuracies of the estimated system resistance or to fluctuating resistance caused by damper or louver adjustments. To cope with such variations a fan should be selected having a high efficiency over a wide range, that is, a *flat* or broad efficiency curve is more desirable than a sharp or narrow curve which, though reaching a high peak, falls off rapidly to either side of a narrow range. When the point of operation varies only within narrow limits and both volume and pressure requirements are accurately known in advance, the designer can select a fan

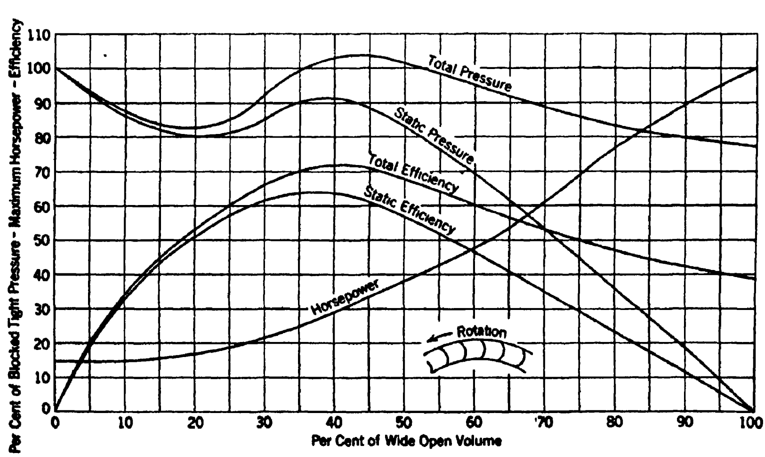


FIG. 2. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED FORWARD

operating at maximum efficiency, irrespective of performance over the entire range.

Generally, fans are selected either at the peak of the static efficiency or to the right of the peak depending on the requirements of the particular installation. Fans selected to the right of the peak will be smaller but will require more power, run at higher speeds and may have a higher sound rating. Where first cost is important and added horsepower and noise are not important, smaller fans may be used. Where efficient and quiet operations are most important, fans are selected at or near the peak of the static efficiency curve. Fans are not ordinarily selected to the left of the peak of the static efficiency curve as this results in larger, more costly fans, requiring more power and in some cases producing objectionable noise.

The curves in Figs. 1, 2, and 3 show operating characteristics for axial flow and two classes of centrifugal fans, namely, forward-curved multiple blade and backward-inclined blade design, for comparison purposes.

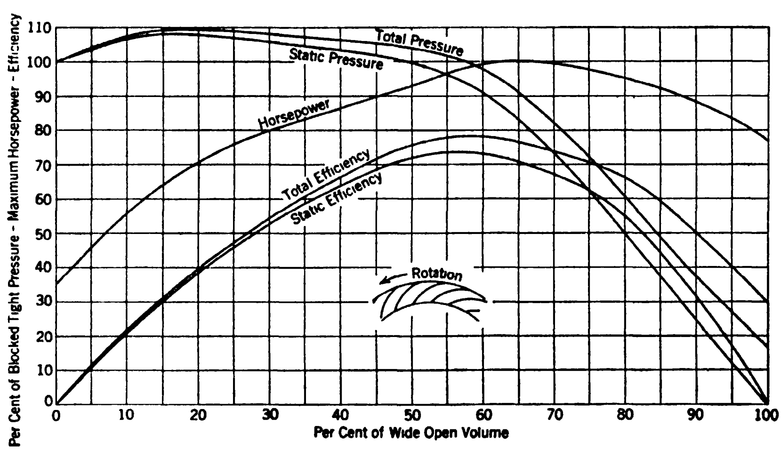


FIG. 3. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED BACKWARD

These curves are not applicable for rigid comparison or actual selection, but are shown to indicate variations in operating characteristics.

The curves in Fig. 1 of a *typical axial flow* fan show characteristics of non-overloading horsepower and high efficiency. These results are obtained by producing a more uniform pressure throughout the blade annulus, so that back flow does not occur except at high pressures. This avoidance in turbulence has a tendency to reduce noise. Fans of this type are operating against static pressures as high as 4 in. water. The capacity and efficiency of axial flow fans when operating above the low pressure range can be improved by the use of either inlet or outlet guide vanes or both. The effect of such vanes is to increase the level of the pressure volume curve, and properly designed vanes on the discharge side of the fan have the advantage of eliminating the rotational component of the air stream, thus restoring uniform axial flow. As high pressures usually require large hubs in proportion to the fan diameter, performance is improved by the use of round-nosed or conical forms mounted coaxially with the direct-connected fan (sometimes partly or wholly enclosing the motor) so as to make the changes in velocity to and from the fan blade annulus as uniform as space conditions permit. When axial

flow fans are installed in ducts, provisions may be made to install the driving motor outside the duct by employing slots in the duct to permit a belt drive from the motor to the fan sheave, or by extending the shaft for a direct-connected motor placed outside of a Y fitting or elbow in the duct system.

The *forward-curved multiblade* fan and the *backward-curved* type are used extensively in heating, ventilating, and air conditioning work. The forward-curved type has a low peripheral speed and a large capacity. (See Fig. 2). The point of maximum efficiency for this fan occurs near the point of maximum pressure. The static pressure drops consistently from the point of maximum efficiency to full open operation. The power curve rises continuously from low to peak capacity and, if reasonable care is exercised in calculating resistance, a moderate reserve in power in the motor selection will prevent overloading.

The *backward-sloped type* includes the full backward-curved blade and the double-curved blade having a forward-curved heel and a backward-curved tip. This type has steep pressure curves, non-overloading power characteristics, and relatively high speed (see Fig. 3). This fan operates at a peripheral speed approximately 175 to 200 per cent of that of the forward-curved multiblade fan for like performance. Pressure curves for this type begin to drop at very low capacity, with the most rapid drop beginning at about 60 per cent of wide open volume. The steep portions of the pressure curves tend to produce nearly constant capacity under changing pressures. Where wide fluctuations in demand occur, especially where the regulation is obtained by damper control and particularly through by-passes, this type of fan is desirable to prevent overloading of motor. The maximum power requirement occurs at about the maximum efficiency. Consequently a motor selected to carry the load at this point will be of sufficient capacity to drive the fan over its full range of capacities at a given speed. The high speed of this type makes it adaptable for direct connected electric motor drives. The dimensional bulk is usually greater than that of the forward-curved multiblade type.

Between the extremes of the forward and backward curved blade type centrifugal fans there exists a number of modified designs differing in angularity and in the shape of the blades. Characteristic curves of these types show varying degrees of similarity to the curves in Figs. 2 and 3.

SYSTEM CHARACTERISTICS

Any ventilating system consisting of duct work, heaters, air washers, filters, etc., has a system characteristic which is individual to that system and is independent of any fan which may be applied to the system. This characteristic may be expressed in curve form in exactly the same manner that fan characteristics may be shown. Typical system characteristic curves are shown as *A*, *B* and *C* in Fig. 4. These curves are drawn to follow the simple parabolic law in which the static pressure or resistance to flow of air varies as the square of the volume flowing through the system. Heating and ventilating systems follow this law very closely and no serious error is introduced by its use.

When a constant speed fan curve for a given size fan is super-imposed upon a system characteristic curve, the relation between the two is at once apparent. The only point common to the two curves is the point at the intersection of the system characteristic curve and the fan characteristic curve, and it is at this point that the combination will operate. In

Fig. 4, curves A, B and C cross the fan characteristic curve at points X, Y and Z. This means that when the fan whose curve is shown is applied to system A, 10,000 cfm will flow through the system. If it is applied to system B, 13,000 cfm will flow, and applied to system C, 16,400 cfm will flow through that system.

The curves in Fig. 4 also illustrate the effect of errors which may be determined by calculating the resistance of a ventilating system. For instance, a given system requires 13,000 cfm and the resistance to flow of the system has been computed as 1.25 in. static pressure. Such a

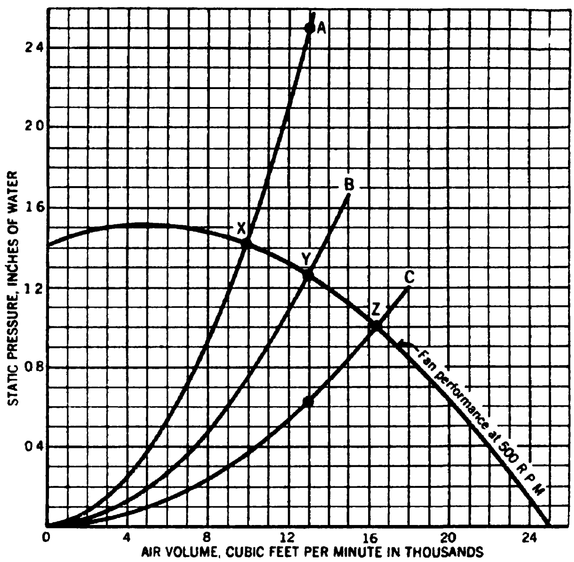


FIG. 4. PARABOLIC SYSTEM CHARACTERISTIC CURVES

system may be represented by curve B in Fig. 4. Assume that 100 per cent error has been made and the resistance calculated should have been 2.5 in. instead of 1.25 in. Then the system would be as shown in curve A. This new system curve crosses the fan curve at 10,000 cfm. Such an error would result in the flow of air being decreased from a design volume of 13,000 cfm to 10,000 cfm. In case the resistance to flow had been over estimated and instead of 1.25 in. being required, the resistance actually should have been 0.625 in., this would correspond to a system curve as shown at C and on this curve the fan would deliver 16,400 cfm to the system instead of the design volume of 13,000 cfm.

In this example extreme errors have been selected to emphasize the effect the square function of the system characteristic has in maintaining the fan performance within comparatively narrow limits. In the first example a system estimated at half what it should have been, resulted in a drop of 23 per cent in volume; and in the second example, a system estimated at twice what it should have been resulted in an increase of 26 per cent in volume.

In some instances fans may be applied to variable flow systems. In such cases the limiting systems may be plotted and the effect on fan performance examined. For instance, a system might vary between

system *A*, shown in Fig. 4 as one limit; and system *B* as the other limit. The fan performance will then fall between points *X* and *Y* on the fan curve at a point determined by the system characteristics at that particular time. If curves *A* and *B* are the limiting systems, the fan performance will never be outside the points *X* or *Y*.

SELECTION OF FANS

The following information is required to select the proper type of fan:

1. Cubic feet of air per minute to be moved.
2. Static pressure required to move the air through the system.
3. Type of motive power available.
4. Whether fans are to operate singly or in parallel on any one duct.
5. What degree of noise is permissible.
6. Nature of the load, such as variable air quantities or pressures.

In order to facilitate the choice of apparatus, the various fan manufacturers supply fan tables or curves which usually show the following factors for each size of fan operating against a wide range of static pressures: (1) volume of air in cubic feet per minute (68 F, 50 per cent relative humidity, 0.075 lb per cubic foot), (2) outlet velocity, (3) revolutions per minute, (4) brake horsepower, (5) tip or peripheral speed, and (6) static pressure. The most efficient operating point of the fan is usually shown by either bold-face or italicized figures in the capacity tables.

Other important factors to be considered in selecting fans are: (1) efficiency, (2) space occupied, (3) sound emission, (4) first cost, and (5) speed (both peripheral and revolutions per minute). These factors are not necessarily shown in the order of importance. In some installations space occupied may be of first importance. In others lowest power consumption is desirable. In many cases quietness of operation of the entire system is essential. Practically all fans operate at their lowest sound level when selected at or near the peak of the static efficiency so that in selecting a fan for highest static efficiency the quietest operating range of the fan will also be obtained. Table 1 shows desirable outlet

TABLE 1. GOOD OPERATING VELOCITIES AND TIP SPEEDS FOR MULTIBLADE VENTILATING FANS

Static Pressure Inches of Water	Forward Curved Blade Fans		Backward Tipped and Double Curved Blade Fans	
	Outlet Velocity Feet per Minute	Tip Speed Feet per Minute	Outlet Velocity Feet per Minute	Tip Speed Feet per Minute
1/4	1000–1100	1520–1700	800–1100	2600–3100
3/8	1000–1100	1760–1900	800–1150	3000–3500
1/2	1000–1200	1970–2150	900–1300	3400–4000
5/8	1200–1400	2225–2450	1000–1500	3800–4500
3/4	1300–1500	2480–2700	1100–1650	4200–5000
7/8	1400–1700	2660–2910	1200–1750	4500–5300
1	1500–1800	2820–3120	1200–1900	4800–5750
1 1/4	1600–1900	3162–3450	1300–2100	5300–6350
1 1/2	1800–2100	3480–3810	1400–2300	5750–6950
1 3/4	1900–2200	3760–4205	1500–2500	6200–7550
2	2000–2400	4000–4500	1600–2700	6650–8050
2 1/4	2200–2600	4250–4740	1700–2800	7050–8550
2 1/2	2300–2600	4475–4970	1800–2950	7450–9000
3	2500–2800	4900–5365	2000–3200	8200–9850

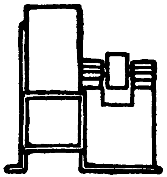
velocities and tip speeds, or peripheral velocities, for various static pressures. Fans selected accordingly will operate at or near the peak of the static efficiency with resulting low power consumption and noise levels. Smaller fans with higher outlet velocities may be used if the installation requirements are such as to warrant the additional power and increased sound level. When space for duct expansion from a fan outlet is not available there may be advantages in selecting a larger fan for reducing duct noises, although lower outlet velocities generally result in lower fan efficiencies which cannot always be justified on the basis of increased cost and space requirements. Fans for schools, churches, residences, and all public buildings should be selected for lower outlet velocities and tip speeds than would be required for other types of installations.

Having selected a fan for its quietest operating point consistent with the requirements of the installation, it must be recognized that ventilating fans, even so selected, emit noise and precautions must be taken in the installation of the fans to prevent this noise from being transmitted to occupied portions of the building. Fans operating against high static pressures produce more noise than fans operating against low static pressures. Consequently, from a noise standpoint, the system should be designed to operate against the lowest static pressure possible. In many modern air conditioning systems it is necessary to introduce devices into the air stream for conditioning the air in various ways, the result of which is to set up a rather high static pressure against which the fan must operate. In such cases the sound level at the fan may be too high to be neglected and special sound treatment of the installation must be considered. When a fan is operating against higher pressures it should be located in a room either removed from the occupied areas, or in a room which has been acoustically treated to prevent sound being carried through the walls to adjoining spaces. The fan should be mounted on a resilient base along with its driving motor to absorb any noise or vibration which might be transmitted to the floor and thence to the building structure. All ducts should be connected to fans with unpainted canvas, or other flexible material, to prevent any vibrations being transmitted to the duct work. Ducts leading into the fan room or from the fan, should be acoustically treated on the interior and in special cases, should be provided with sound traps or filters. Many ventilating systems encounter noises which are connected with the fan in no way. Noises due to high duct velocities, abrupt turns, grilles, etc., may be present. Treatment of such problems is covered in Chapter 32.

If double width, double inlet fans are selected, care must be taken that both inlets have the same free area. If one inlet of a fan is obstructed more than the other, the fan will not operate properly, as one half of the wheel will deliver more air than the other half. Proper installation and location of fans is as important as the selection of the size of fan for that particular installation. All fans will work according to their characteristic curves if they are properly selected and properly installed and operated.

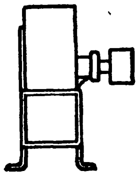
FAN DESIGNATIONS

In order to prevent misunderstandings, which may cause delays and losses, the arrangements of fan drives adopted by the *National Association of Fan Manufacturers* and indicated in Fig. 5 are suggested.



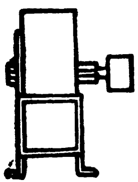
Arr. 1.
For belt drive.

Wheel overhung Bearings on pedestal



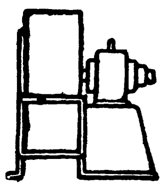
Arr. 2.
For belt drive.

Pulley and wheel overhung. Bearings in bracket on fan housing. Made only in smaller sizes for reversible discharge.



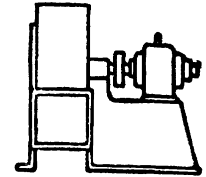
Arr. 3.
For belt drive.

Pulley overhung Bearings supported on fan housing



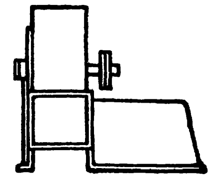
Arr. 4
For direct drive.

Wheel overhung. No bearings on fan Wheel mounted on motor or engine shaft Pedestal for motor or engine



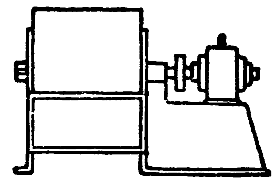
Arr. 5.
For direct drive.

Wheel overhung Includes housing, wheel, shaft, one intermediate bearing, flanged coupling and pedestal only for motor or engine



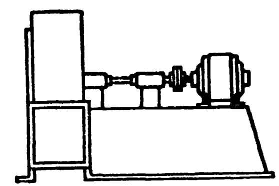
Arr. 6.
For direct drive.

Three-bearing arrangement with fan bearing at inlet side. Includes housing, wheel, shaft, one bearing (in inlet), rigid coupling, and pedestal only for motor or engine.



Arr. 7.
For direct drive.

Similar to Arr 6, but with two bearings on fan, and flexible instead of rigid coupling



Arr. 8.
Similar to Arr. 5.

But with two bearings on pedestal with motor, and flexible instead of rigid coupling

Fig. 5. Arrangement of Fan Drives

Facing the driving side of the fan, blower, or blast wheel, if the proper direction of rotation is clockwise, the fan, blower, or blast wheel will be designated as ***clockwise***. If the proper direction of rotation is counter-clockwise, the designation will be ***counter-clockwise***. (The driving side of a single inlet fan is considered to be the side opposite the inlet regardless of the actual location of the drive.)[2]

This method of designation will apply to all centrifugal fans, single or double width, and single or double inlet. Do not use the word ***hand*** but specify ***clockwise*** or ***counter-clockwise***.

The discharge of a fan will be determined by the direction of the line of air discharge and its relation to the fan shaft, as follows:

Bottom horizontal: If the line of air discharge is horizontal and below the shaft.

Top horizontal: If the line of air discharge is horizontal and above the shaft.

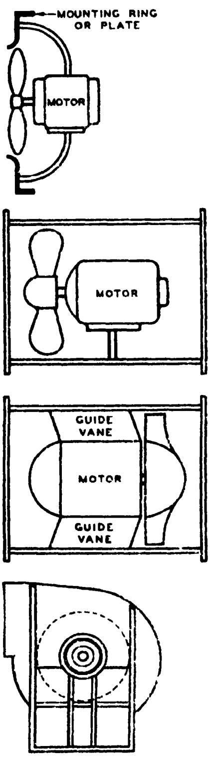
Up blast: If the line of air discharge is vertically up.

Down blast: If the line of air discharge is vertically down.

All intermediate discharges will be indicated as angular discharge as follows:

Either top or bottom angular up discharge or top or bottom angular down discharge, the smallest angle made by the line of air discharge with the horizontal being specified.

Fig. 6 shows the names and definitions of types of fans recently distributed by the *National Association of Fan Manufacturers*[3].



Propeller Fan

A propeller fan consists of a propeller or disc type wheel within a mounting ring or plate and including driving mechanism supports either for belt drive or direct connection

Tubeaxial Fan

A tubeaxial fan consists of a propeller or disc type wheel within a cylinder and including driving mechanism supports either for belt drive or direct connection.

Vaneaxial Fan

A vaneaxial fan consists of a disc type wheel within a cylinder, a set of air guide vanes located either before or after the wheel and including driving mechanism supports either for belt drive or direct connection.

Centrifugal Fan

A centrifugal fan consists of a fan rotor or wheel within a scroll type of housing and including driving mechanism supports either for belt drive or direct connection.

FIG. 6 NAMES AND DEFINITIONS OF TYPES OF FANS

[2]Recommendations adopted by the *National Association of Fan Manufacturers*

[3]Supplement No B to Form X-12, *National Association of Fan Manufacturers*

FAN CONTROL

In some heating and ventilating systems it is desirable to vary the volume of air handled by the fan, and this may be accomplished by a number of methods. Where the change is made infrequently, the pulley or sheave on the driving motor, or fan, may be changed to vary the speed of the fan and alter the air volume. Dampers may be placed in the duct system to vary the volume. Variable speed pulleys or transmissions, such as fan belt change boxes, electric or hydraulic couplings, may be used to vary the fan speed. Variable speed motors and variable fan inlet vanes may also be used to adjust the fan volume. All of these methods will give control. From a power consumption standpoint, a reduction of the fan speed is most efficient. Inlet vanes save some power and dampers save the least. From the standpoint of first cost, dampers usually are the lowest in cost. In some installations adjustments of volume are desirable at various times during the day or continuously. In others an increased supply of air in summer over that needed in winter is demanded. The demands of each case will dictate which type of control is most desirable. Where noise is a factor, lowering the fan speed if possible is preferred as a control means, because of the resulting reduction in sound level.

MOTIVE POWER

Heating, ventilating and air conditioning fans are usually driven by electric motors, although other prime movers may be used. The small sizes of fans, and especially those operating in the higher speed range, are equipped with direct-connected motors. For larger size fans and those operating at lower speed V-belt drives are generally used.

In selecting the size of motor for operating a fan, it is advisable to select at least the standard size next larger than the fan requirements. Direct-connected motors do not require so great a safety factor as belted units. Justification for liberal power provision exists only in systems where it is possible that larger volumes of air may be required at intervals and made available by use of by-pass dampers, thus greatly reducing the system resistance. If such a system includes a fan with forward-curved blades, it would be necessary that the motor be sized for the maximum volume and duty. If such a system includes fans with backward-curved blades, the volume peak would not make it necessary to provide additional motor power. In selecting fans for such a system, sound ratings should be given careful consideration.

Where a system is constant, and has no provision for volume change that would materially reduce the resistance, and when the resistance calculations are reasonably accurate, there is no necessity for too liberal a motor allowance, even where fans with forward-curved blades are used, if the fan has been properly selected. Fig. 4 shows that the system resistance varies as the square of the volume and the fan static pressure varies approximately inversely as the volume, thus greatly offsetting the trend toward both increase in air delivery and motor load. Reference to Fig. 4 indicates that there is no justification for allowing large spare motor capacity, and it is generally more economical to operate motors well loaded.

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CHAPTER 30

Air Distribution

Standards for Satisfactory Conditions, Definitions, Mechanics of Air Distribution, Types of Supply and Return Openings, Outlet Locations, Return and Exhaust Intakes, Specific Applications, Balancing the System

CORRECT air distribution contributes as much or more to the success of a forced air heating, ventilating, cooling or air conditioning system as does any other single factor. The scope of the chapter is limited to the air distribution within the conditioned space. Reference is made to the distributing duct system only insofar as it affects the performance of the air distribution outlet. See Chapter 31 for information on air duct design.

STANDARDS FOR SATISFACTORY CONDITIONS

The air distribution problem consists of distributing air as a cooling, heating, drying, moistening or ventilating medium in a designated space within accepted limits of air motion, temperature variation, temperature fluctuation, direction, humidity and noise. Reference should be made to Chapter 2, Physiological Principles, for the accepted standards on room temperature, humidity, air motion and direction. Material in Chapter 32, Sound Control, covers acceptable room noise levels and noise generated by air outlets.

Variations from accepted standard limits of each element may result in discomfort to the occupants. Neglecting noise, discomfort complaints usually arise from draftiness, or stuffiness. A draft may be defined as an air current which, due to its temperature, humidity, or motion, removes more heat from a body surface than is usually dissipated. Although stuffiness may be attributed to odors, the complaint of stuffiness usually results from a person feeling too warm. Outside of localized sensation, such as caused by a single down draft, draftiness and stuffiness may be considered functions of effective temperature, which of course takes in the factors of air temperature, motion and humidity. Draftiness can be associated with too low an effective temperature, and stuffiness with too high an effective temperature. Therefore, satisfactory comfort conditions are the result of minimizing the factors of temperature variation, temperature fluctuation, gusts, air motion and noise.

Definitions

1. *Supply Opening or Outlet:* Any opening through which air is delivered into a space which is being heated, or cooled, or humidified, or dehumidified, or ventilated.

2. *Exhaust Opening or Return Intake:* Any opening through which air is removed from a space which is being heated, or cooled, or humidified, or dehumidified, or ventilated

3. *Outside Air Opening:* Any opening used as an entry for air from outdoors.

4. *Grille:* A covering for any opening and through which air passes.

5. *Damper:* A device used to vary the volume of air passing through a confined cross-section by varying the cross-sectional area.

6. *Multiple Louver Damper:* A damper having a number of adjustable blades.

7. *Single Louver Damper:* A damper having one adjustable blade.

8. *Face:* A grille with provision for attaching a damper.

9. *Register:* A face with a damper attached.

10. *Flange:* The portion (either integral or separate) of a grille, face, or register extending into the duct opening for the purpose of mounting.

11. *Frame:* The portion (either integral or separate) of a grille, face, or register extending around the duct opening for the purpose of mounting.

12. *Margin:* The margin of a grille, face, or register is one-half of the difference between the duct dimension and over-all dimension measured either horizontally or vertically.

13. *Fret:* The member separating the openings of a grille, face, or register.

14. *Free Area:* The total minimum area of the openings in the grille, face, or register through which air can pass.

15. *Core Area:* The total plane area of the portion of a grille, face, or register bounded by a line tangent to the outer edges of the outer openings through which air can pass.

16. *Mean Area:* The total of the core and free areas divided by two.

17. *Duct Area:* The area of a cross-section of the duct based on the inside dimensions at the point where the grille, face or register is mounted.

18. *Percentage Free Area:* The ratio of the free area to the core area expressed in percentage.

19. *Aspect Ratio:* The ratio of length of the core of a grille, face or register to the width.

20. *Throw:* The distance air will carry measured along the axis of an air stream from the supply opening to the position in the stream at which air motion reduces to 50 fpm.

21. *Envelope:* The outer boundary of an air stream.

22 *Drop:* The vertical distance the lower edge of the air stream drops between the time it leaves the outlet and reaches the end of its throw (h in ft).

23. *Rise:* The converse of drop.

24 *Induction:* The entrainment of room air by the air ejected from the outlet.

25. *Primary Air:* The air leaving an outlet (Q_1 in cfm).

26. *Secondary Air:* The room air picked up by the primary air through induction (Q_2 in cfm).

27. *Total Air:* The mixture of primary and secondary air (Q_3 in cfm).

28 *Induction Ratio:* The total air divided by the primary air equals r, or Q_3/Q_1

29. *Outlet Velocity:* The average air velocity emerging from the outlet (V_1 in fpm) measured at the plane of the opening.

30. *Terminal Velocity:* The average air stream velocity at the end of the throw (V_T in fpm).

31 *Horizontal Spread:* The divergence of the air stream in the horizontal plane after it leaves the outlet (Degrees).

32. *Vertical Spread:* The divergence in the vertical plane (Degrees).

33. *Temperature Differential:* Temperature difference between primary and room air ($t_r - t_{as}$).

34. *Vane Ratio:* The ratio of depth of vane to shortest opening width between two adjacent grille bars.

MECHANICS OF AIR DISTRIBUTION

In the mechanics of air distribution, two major problems are involved: (1) complete mixing of the primary air and room air outside of the zone of occupancy and (2) counteraction of the natural convection and radiation effects within the room.

Induction

When air is discharged from an outlet into a free open space, the primary air stream entrains room air as it traverses the space. This entraining effect increases the cross-sectional area and reduces the velocity of the resulting air stream. Induction takes place with the conservation of linear momentum; this has been confirmed by tests which indicate that the momentum remains almost constant throughout the entire measureable length of the air stream. This relationship may be expressed by the Equation 1:

$$M_1 V_1 + M_2 V_2 = (M_1 + M_2) V_3 \tag{1}$$

where

M_1 = mass of primary air.
M_2 = mass of secondary air.
V_1 = velocity of primary air.
V_2 = velocity of secondary air (normally = 0).
V_3 = velocity of the mixture.

Substituting zero for V_2 and the volume rate Q for the mass (M), and solving for the induction ratio (r):

$$r = \frac{V_1}{V_3} = \frac{Q_1 + Q_2}{Q_1} \tag{2}$$

The total air entrained by an air stream is in direct proportion to the distance from the discharge of the outlet. For a given blow from a wall in which a number of outlets are located, the induction ratio may be increased by increasing the aspect ratio, diverging the vanes of the outlet, or by simultaneously increasing the number of outlets, reducing their size, and increasing the velocity but maintaining a constant blow. The aspect ratio must be increased considerably from that of a rectangle to a slot before marked changes in the entrainment ratio take place.

Spread

The induction effect results in the spreading of the air stream. Equation 3 derived from induction Equation 1 gives spread or cross-sectional area of the stream as a function of induction ratio, volume of primary air, and primary air velocity.

$$A = \frac{r^2 Q_1}{V_1} = \frac{V_1 Q_1}{(V_3)^2} \tag{3}$$

where

Q_1 = primary air quantity, cubic feet per minute.
V_1 = primary air velocity, feet per minute.
V_3 = air velocity of mixture, feet per minute.
r = induction ratio.

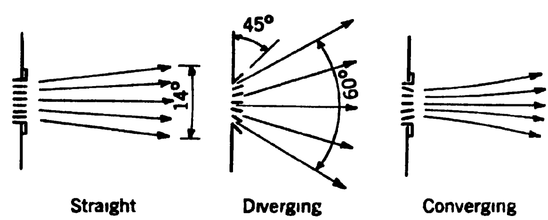

Fig. 1. Spread of Air Stream with Various Vanes

The average jet angle (included angle in both planes, see Fig. 1) for an air stream as it emerges from a rectangular outlet of any shape without spreading vanes is about 19 deg, plus or minus 5 deg, depending on the type of approach, type of outlet and velocity. The spread increases slightly with velocity. A vaned outlet discharging air uniformly forward will result in a spread of about 14 deg. This is equivalent to a spread in any direction of about one foot in every 8 ft of blow.

Throw

The distance air will carry measured along the axis of an air stream from the supply opening to the position in the stream at which the average

frontal air velocity reduces to 50 fpm is termed the *throw*. The throw distance is based on an assumed terminal velocity, which can be assigned any arbitrary value. Since air striking a wall at too high a velocity may bring the air stream down within the occupied zone, the terminal velocity should be limited to 50 fpm. The maximum transverse velocity of the air stream is usually from 2.5 to 3.5 times the average frontal velocity. Assuming no obstructions, the blow is affected by face velocity, core area, aspect ratio and included angle of effluent stream as determined by vanes. For low aspect ratios, the major variables of velocity, area and effluent angle are related[1] approximately as given in Equation 4 when the air stream is unaffected by obstructions of any kind.

$$X_a = \frac{kQ}{\sqrt{a_0 b_0}} \tag{4}$$

where

X_a = throw, feet.

Q = air volume flow rate, cubic feet per minute.

a_0 and b_0 = grille width and height, inches.

k = dimensionless constant with the following approximate empirical values:

Vanes set straight ahead		= 0.77
Vanes causing a spread on each horizontal side of	15 deg	= 0.66
	30 deg	= 0.45
	45 deg	= 0.34

Vanes

For vanes to be mechanically satisfactory, the depth of the vane should be between one and two times the spacing between the vanes. If the ratio of vane depth to spacing is less than one, effective turning by means of the vanes cannot be obtained. Little improvement is obtained by increasing the ratio beyond two.

Straight Vanes As mentioned previously, the included angle between both planes will be in the neighborhood of 14 deg, for a straight setting of the vanes as shown in Fig. 1.

Diverging Vanes. Such vanes set for an angular spread will have a marked effect on the direction and distance of travel of an air stream. An outlet having vertical vanes set straight forward in the center, with uniformly increasing angular deflection to a maximum at each end of 45 deg, will produce an air stream with a horizontal included angle of approximately 60 deg as shown in Fig. 1. The throw will be reduced one-half for such a vane setting. Increasing the divergence of the vanes reduces the air quantity handled by an outlet for a given duct static pressure. The primary function of the vanes is to spread the air horizontally. Little is gained by spreading the air vertically.

Converging Vanes. The blow of an outlet may be somewhat increased by converging the vanes of an outlet as illustrated in Fig. 1. Even with converging vanes, the resultant angle of spread of an air stream will not be less than 14 deg. The air converges for a few feet in front of the outlet, and then diverges more than if the vanes had been set straight.

Both the horizontal and vertical vanes of an outlet are important. After an installation has been made, many conditions of draftiness or stuffiness can be alleviated by some vane adjustment, provided an independent means for regulation of static pressure behind the vanes is included.

Room Air Motion

The air motion in the occupied zone is usually such that the air travels across the room in reverse direction to the blow of the outlet. The cross-

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[1]The Rationale of Air Distribution and Grille Performance, by C O Mackey (*Refrigerating Engineering*, Vol 35, No 6, June, 1938, p 417)

sectional area of this stream is equal to the outlet wall area less the stream area and the area obstructed by furnishings. Equation 5 gives the average room velocity in the occupied zone as a function of the air volume supplied per square foot of outlet wall area and the outlet velocity.

$$V_r = \frac{Q_3}{AZ} \tag{5}$$

where

V_r = average room velocity, feet per minute.
A = outlet wall area, square feet.
Z = 0.6 (reduction factor to allow for supply air stream 20 per cent, furniture obstruction 20 per cent, at point where supply air stream occupies 20 per cent of the room cross-section).

Since $Q_3 = Q_1 r$ by definition, and $r = \frac{V_1}{V_3}$ from Equation 2, and V_3 is assumed to be about 200 fpm for total induction in actual practice, then Equation 5 results in:

$$V_r = \frac{Q_1}{A} \times \frac{V_1}{V_3} \times \frac{1}{Z} = \frac{FV_1}{120}$$

or

$$F = \frac{120\,V_r}{V_1} \tag{6}$$

where F is the room circulation factor expressed in cubic feet per minute per square foot of outlet wall area. Thus room air motion is directly a function of outlet velocity and air volume per square foot of outlet wall area. Hence Equation 6 and Table 1 can be used to determine the probable acceptability of a particular installation from the standpoint of proposed air volume, outlet wall area, and outlet velocity.

Vertical Drop and Rise

The vertical distance the lower edge of an air stream moves between the outlet and the end of the blow is termed the drop or rise (*H*). This drop or rise is influenced by the difference in density between the air stream and the room air, resulting from the temperature difference and the spread of the air stream. For air emerging at room temperature, the drop or rise will be a function of the spread only and will be equal to:

$$H = L \times \tan\left(\frac{\text{Spread Angle}}{2}\right) \tag{7}$$

where

H = drop due to spread, feet.
L = throw, feet.

TABLE 1. VALUES OF ROOM CIRCULATION FACTOR (F) IN EQUATION 6

Outlet Velocity Fpm V_1	Average Room Velocity Fpm, V_r					Outlet Velocity Fpm V_1	Average Room Velocity Fpm, V_r				
	10	20	30	40	50		10	20	30	40	50
200	6.0	12.0	18.0	24.0	30.0	900	1.3	2.7	4.0	5 3	6.7
300	4.0	8.0	12.0	16.0	20.0	1000	1.2	2.4	3.6	4.8	6.0
400	3.0	6.0	9.0	12.0	15.0	1200	1.0	2.0	3.0	4.0	5 0
500	2.4	4.8	7.2	9.6	12.0	1400	0.9	1.7	2.6	3.4	4 3
600	2.0	4.0	6.0	8.0	10.0	1600	0.8	1.5	2.3	3.0	3 8
700	1.7	3.4	5.1	6.8	8.5	1800	0.7	1.4	2.0	2.7	3.4
800	1.5	3.0	4.5	6.0	7.5	2000	0.6	1 2	1 8	2 4	3 0

When there is a temperature difference between the air stream and the room, the additional drop or rise is approximately given by Equation 8:

$$H = \frac{n_1 (t_r - t_{as}) L n_2}{V_1} \tag{8}$$

where

n_1 and n_2 = constants (tentative suggested values $n_1 = 5$, $n_2 = 1.2$).
t_r = room temperature, degrees Fahrenheit.
t_{as} = supply air temperature, degrees Fahrenheit.

For cooling application H is subtracted from the outlet height, for heating H is added.

Duct Approaches to Outlets

Assuming that proper supply openings for a given installation have been selected, unsatisfactory performance may still result due to the construction of the duct work immediately back of the supply openings. Performance data on the grilles and registers of various manufacturers are based upon results obtained with the air approaching the grille perpendicularly and at uniform velocity over the entire duct cross-section. Where this condition does not exist in practice, performance predictions based on published data cannot be realized. Every precaution should be taken to secure as nearly ideal conditions in the approaching air stream as are possible.

In addition to disturbances due to the construction of the duct work itself are those which may be created by dampers immediately behind the grille. Where either multiple louver or single blade dampers are used for throttling, considerable deflection of the air stream may result. This is particularly true when the fins of the register core are perpendicular to the damper blades. If the core has sufficient depth and the fins are parallel to the blades, there is a marked tendency to straighten the air stream, although some deflection may still result.

Any attempt to secure a low face velocity and high duct velocity by the construction of any expanding chamber immediately behind the grille is likely to be unsuccessful. In order to expand from a small duct to a larger one, and have the air stream fill the duct at the end of the diverging section without turbulence, angle A in Fig. 2 should be about 7 deg. From this it is apparent that an attempt to secure equivalent results with a short connection would be futile. What actually happens when this is attempted is illustrated by the arrows in Fig. 2. When localized high velocities through the supply opening exist from this cause or any other, the noise produced will naturally exceed that which would be expected

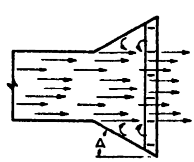


FIG. 2. EFFECTS OF EXPANDING DUCT

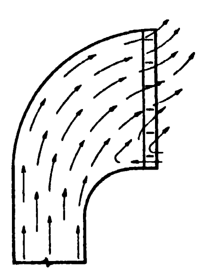


FIG. 3. UNEQUAL FACE VELOCITIES

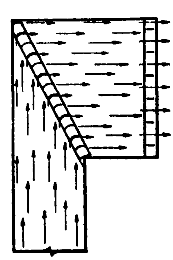


FIG. 4. EFFECT OF TURNING MEMBER

from the supply opening area and average face velocity. This fact should be remembered in considering the use of register dampers, particularly in those cases where there must be considerable throttling with the damper to balance a poorly designed system. Where reduction of noise is important, it is recommended that balancing dampers be placed in the duct *ahead* of the acoustic duct lining.

Similar unequal face velocities, aggravated by a deflection of the air stream, are obtained with the arrangement shown in Fig. 3. The latter may be corrected by inserting a turning member in the elbow back of the outlet face as shown in Fig. 4. The importance of straightening the air stream and effecting uniform distribution over the entire face of the supply opening cannot be over-emphasized[2].

Dampers of special construction, as illustrated in Fig. 5, may be used to maintain a constant direction of blow, approximate distance of blow, and constant outlet velocity regardless of the damper's position. The capacity of dampers diagrammed as *A* and *B* will be roughly in proportion to the position of the operating lever. They are particularly effective for

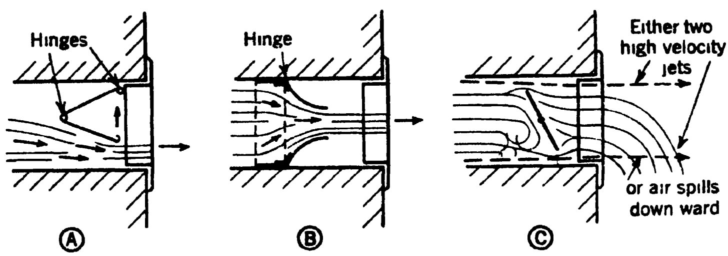


Fig. 5. Effect of Various Damper Arrangements Designed for Straight Blow

cooling work with oversized grilles. The single leaf damper shown as *C* in Fig. 5 is objectionable in that it frequently results in a condition whereby two high velocity jets are created along the sides of the duct, or the air spills immediately downward on the occupants below the outlet.

Operation of Ceiling Outlets

The relationships presented for sidewall outlets will apply with the proper modification to the operation of ceiling outlets. With the ceiling outlets, the angle of distribution may be a full 360 deg. This extreme angle of distribution results in a high rate of induction and short blow, and also permits air to be introduced into a space satisfactorily at a relatively high temperature differential and high velocity. Ceiling outlets may handle greater quantities of air than a comparable sidewall outlet without the creation of excessive air motion. In contrast to wall outlets, the ceiling outlet does not necessarily have to be equipped with adjustable features, other than means to regulate air quantity. Frequently sections of ceiling outlets are blanked off to avoid having the air stream strike wall or column obstructions.

TYPES OF SUPPLY AND RETURN OPENINGS

Perforated Outlets. Due to the non-adjustability and small vane ratio of perforated sheet metal outlets, they have not met with favor as supply

[2]A.S.H.V.E. Research Report No. 1155—The Performance of Stack Heads, by D. W. Nelson, D H. Krans and A. F. Tuthill (A S H V.E. Transactions, Vol. 46, 1940, p. 205) A S H V E. Research Report No. 1226—Performance of Side Outlets on Horizontal Ducts, by D W Nelson and G. E. Smedberg (A S H.V.E Transactions, Vol 49, 1943, p 58).

openings. They are useful primarily where directional air control is unnecessary, and for return air openings.

Vaned Outlets. Outlets equipped with both vertical and horizontal adjustable vanes are particularly suited to sidewall distribution. For proper control over the air flow, the vane ratio should be from 1 to 2. Outlets with non-adjustable vanes may be employed; however, they should only be used where the performance is not critical or can be adequately predicted.

Registers. Fixed vanes or perforated grilles equipped with a single blade damper are termed registers. They are primarily used for residential heating systems, where the outlet distribution is not critical and low cost is of importance.

Slotted Outlets. Slotted outlets essentially consist of either flat steel plates containing a number of long narrow slots or a single long narrow slot, with a free area of approximately 10 per cent. In order to give a good conversion from static pressure to velocity pressure, the sides of the slots are rounded to give a venturi effect. Due to their high aspect ratio, the slotted outlets have a greater induction effect than the comparable vaned outlets of equal area; thus, the throw of the slot is slightly less. They are primarily useful where an unobtrusive means of distribution is desired, and where it is desirable to submerge the outlets into the room decoration. Since these outlets offer greater induction at a given noise level, they are useful in obtaining proper air motion when otherwise limited by design.

Ejector Nozzles refer to outlets which operate at high static pressures, and which are constructed to give a high conversion from static in the duct to velocity pressure in the outlet, and have a high induction effect due to their high outlet velocity. The ejector is chiefly used for long throws and industrial process installations, such as drying, freezing, cooking, etc. Another type of ejector is sometimes referred to as a *louver nozzle* having a 45 to 90 deg elbow, which can be rotated similarly to a universal joint about an axis perpendicular to the surface to which it is fastened. These outlets give a considerable degree of adjustability and are, therefore, desirable for use in confined spaces where spot cooling is employed.

Ceiling Outlets. There are two general classifications of ceiling outlets, one the simple ceiling plaque, usually with limited induction effect, and the other a concentrically vaned ceiling outlet with high induction effect. Plaque outlets, although cheap and simple in construction, are difficult to control and are not generally satisfactory for comfort cooling where the entering air is more than 12 F below the room temperature. The vaned ceiling outlet with marked induction will distribute air uniformly over an entire half sphere. The induction effect is greatest in the direction of the axis of the outlet, and least in the plane perpendicular to the axis and located at the ceiling level. Thus the induction is greatest in the vertical direction where the least blow can be tolerated, and least in the horizontal plane at the ceiling where the greatest blow is both desired and permissible.

Perforated Ceilings. Such outlets consist of metal or composition board ceiling with small perforations through which air may be supplied to a room. The free area of the ceiling is about 10 to 15 per cent. The perforated ceiling can be designed to give a lower rate of room air motion for a given air supply than any other type of outlet. For this reason, the perforated ceiling is particularly applicable in installations requiring a

low room air motion and having a high heat or ventilation load necessitating a high rate of air change.

OUTLET LOCATIONS

In selecting the location of outlets, consideration must be given to the factors of physical construction, physical appearance, location of heating or cooling loads, and outlet performance. Final outlet location will be a compromise between these factors.

1. The *physical construction* of a building, particularly of old buildings, immediately places limitations on the type of distribution system which can be employed. Therefore, the first factor in the selection of outlet locations is a consideration of the possible location of the supply duct, that is, whether it is above the ceiling, within the walls, through furred spaces above corridors, or in the conditioned space, etc. A particular method of distribution may be highly desirable but its execution, due to the location of beams and masonry walls, may be an impossibility.

2. The *physical appearance* of the outlets should conform to the esthetic appearance of the room. In factories, warehouses, etc., the esthetic demand may not be high; however, in department stores, clubs, theaters, etc., the location of the grilles may be entirely dictated by such demands. In carefully decorated rooms, it may even be necessary to completely conceal the method of distribution by the use of slots located in recesses in the walls or ceilings.

3. The location of *heating or cooling loads* in a room dictate to a great extent the general location of the outlets. The outlets should be located to neutralize any undesirable cold drafts or radiation effects set up by a concentration of the heating or cooling load. The problem can be divided into natural loads due to outside weather and internal heat loads.

Natural Heating or Cooling Loads

Winter. In winter the primary heating load is from exposed walls, windows and skylights. Heat is lost primarily through convection to these exposed surfaces. The convection currents or cold drafts drop down the exposed surfaces and seriously impair the comfort conditions in the room, and particularly at the floor level near the exposed surfaces. The outlet should be located to counteract these down drafts. Two methods may be employed:

1. Direct counteraction of convection currents can be obtained by locating the outlets to blow upward from beneath windows or exposed walls or to blow across the exposed wall. This method is desirable in small offices or bedrooms, or any location where people are seated or working near exposed surfaces. In northern climates, where the outside temperature may be constantly below 40 F, and the construction consists of uninsulated walls and single glass this method of distribution is particularly useful for the maintenance of comfort requirements.

2. High induction by ceiling or wall outlets may be employed to nullify the convection currents from exposed surfaces. If outside temperatures are consistently below 40 F, and the exposed surfaces are poorly insulated, the induction effect required for neutralization of the down drafts is so great that the air motion in the room will exceed comfort limits. Therefore, in northern latitudes, this method can only be recommended for use in factories, warehouses, etc., where comfort conditions are not critical. The wide use of ceiling suspended heat diffusers in cold climates and in factory spaces illustrates the results to be obtained by such distribution. If the exposed walls are well insulated or the windows are equipped with double glass, ceiling distribution may prove reasonably satisfactory even in the coldest of climates. In mild climates, where the outside temperature seldom drops below 40 F, offices and bedrooms may be satisfactorily heated by ceiling or wall outlets.

Summer. The primary discomforting effect to be experienced in summer is the radiation from sun exposed walls or windows. Radiation from a wall is a function of the surface temperature of the wall which is related to the amount of wall insulation. Radiation from well insulated walls is negligible compared to that from uninsulated walls. Radiation

may be countered by blowing the air supply from an inner wall towards the exposed surface, or by discharging the air vertically upward along the exposed surface. If vertical distribution is employed, the outlet air should be fanned out at an angle of 15 to 20 deg with the vertical and in a plane parallel to the wall. Directing the air parallel to the wall minimizes the formation of a cold spot directly in front of the outlet when low velocities are used.

Internal Heat Load

If a concentrated source of heat is located at the occupancy level of the room, the heating or radiation effect may be countered by blowing the supply air toward the heat source or by locating an exhaust or return grille adjacent to the heat source. The latter method will prove more economical, as heat will be withdrawn at its source rather than be dissipated into the conditioned space. Where a lighting load is particularly heavy (five watts per square foot), and located high in a conditioned

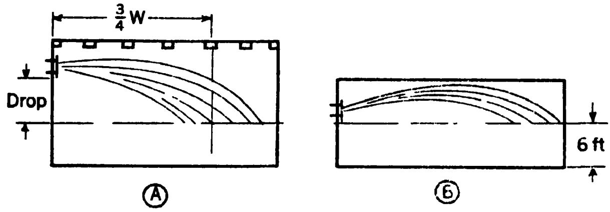


FIG. 6. BLOW OF WALL OUTLETS

space, it may be economically desirable to locate the outlets below the lighting load. Warm air from the lights will stratify near the ceiling and can be removed by an exhaust fan.

Outlet Performance

The factors of outlet performance, throw, drop, capacity, noise, dirt and room air motion place considerable limitations on the design of a satisfactory distribution system.

1. *Blow.* The blow of wall or ceiling outlets should be selected to cover three-quarters of the distance toward an exposed wall or window as shown in *A* of Fig. 6. Overblowing is considerably more serious than underblowing, as an overblow will create objectionable down drafts from any surface it strikes; although underblowing in the case of heated air may be serious in that the warm air may rise too rapidly and thus cause stratification in the occupied zone. In spaces with beamed ceilings, the outlets should be located below the bottom of the lowest beam level, and preferably low enough so that an upward or arched blow may be employed. The blow should be arched sufficiently to miss the beams and, at the same time, in such a manner as to prevent the primary or induced air stream from striking furniture and obstacles producing objectionable drafts. If an outlet is adjusted downward to avoid a beam, cold air may enter the zone of occupancy long before the desired induction has taken place, thus causing serious complaints.

2. *Drop.* The outlets should be located so that the air stream at the termination of the blow is not less than 5 or 6 ft above the floor level. As illustrated in *B* of Fig. 6 the maximum permissible blow for a given ceiling height may be obtained by locating the outlet low on the wall, arching the blow, and sweeping the air across the flat ceiling. The air, as it traverses the room, will adhere to the ceiling. The objection to this method is the possible streaking of the ceiling with dirt.

3. *Capacity.* The *Catalog Data Section* or manufacturers' rating sheets may be consulted for selection of the proper number of outlets for a given air quantity. Due to their high induction ratio, ceiling outlets will in general handle more air per outlet than either comparable sidewall or floor outlets.

4. *Noise.* The noise of an outlet is primarily the function of the outlet velocity and size, and secondarily of the outlet construction. The maximum acceptable noise level in a space may completely dictate the permissible outlet velocities that may be employed.

Table 2. Recommended Return Intake Face Velocities

Intake Location	Velocity Over Gross Area Fpm
Above occupied zone	800 up
Within occupied zone, not near seats	600–800
Within occupied zone, near seats	400–600
Door or wall louvers	500–700
Undercutting of doors (through undercut area)	600

(See Chapter 32 for discussion of permissible room noise levels and noise generated by outlets.)

5. *Room Air Motion.* The factors leading to high air motion are excessive velocity, high air volume per square foot of outlet wall area, overblow, striking of beams causing a spilling of the air into the zone of occupancy, and heating in severe climates by means of ceiling outlets which are directed downward.

6. *Dirt.* Although the primary air may be carefully filtered, dirt from the conditioned space may be deposited on the walls or ceiling wherever there is considerable secondary air motion. With ceiling outlets, dirt streaking may be minimized by carefully streamlining the discharge of the outlets. With wall outlets, dirt streaking may be minimized by not directly impinging the air on any ceiling or room surface. Floor outlets may offer objection as dirt collectors.

RETURN AND EXHAUST INTAKES

Where the air supply causes a relatively large induction effect, only three factors govern the selection and application of return and exhaust intakes: (1) velocity in occupied zone adjacent to intake, (2) permissible pressure drop through intake, and (3) noise.

Velocity

Air handled by an exhaust or return intake is drawn from all directions, the velocity dropping off rapidly in every direction. The only locality where drafts may prove objectionable is adjacent to the intake. To prevent excessive air motion in the occupied space due to the return system, it is advisable to compute the total air motion toward the exhaust opening as outlined in Equation 5 where A is the exhaust wall area in square feet. Recommended return intake face velocities are given in Table 2.

The withdrawal of air from a space through a return intake is a minor factor in control of the room air motion. The control of the room air motion for the maintenance of comfort conditions depends on the proper selection of the supply outlets. Thus the location of the return intake is not critical, nor the use of an elaborate return system necessary, provided the air motion in the occupied zone adjacent to the intake does not exceed comfort limits. A single return intake or a few large intakes will prove satisfactory provided no local high velocity zones are created.

The permissible pressure drop will depend on the choice of the designer. Table 3 gives pressure drop through plain lattice intakes as a function of free area and face velocity.

Noise

The problem of noise generated by return intakes is the same as that generated by supply outlets. In computing resultant room noise levels from the operation of an air conditioning system, the return intake must be included as a part of the total grille area. The only difference between the supply outlets and return intakes is in their frequent installation at

TABLE 3. APPROXIMATE PRESSURE DROPS FOR LATTICE RETURN INTAKES

Inches Water Gage—Standard Air

PER CENT FREE AREA	FACE VELOCITY, FPM						
	400	500	600	700	800	900	1000
50	0.06	0.09	0.13	0.17	0.22	0.28	0.35
60	0.04	0.06	0.09	0.12	0.16	0.20	0.24
70	0.03	0.05	0.07	0.09	0.12	0.15	0.18
80	0.02	0.03	0.05	0.07	0.09	0.11	0.14

the ear level. When located at ear level, it is recommended that the return intake velocity be 75 per cent of the maximum permissible outlet velocity.

Ceiling locations are recommended for bars, kitchens, lavatories, dining rooms, club rooms, etc., where warm air will gravitate to the ceiling level. Ceiling returns are less desirable in spaces with severe winter exposure, and where stratification of cold air may take place at the floor level. During the heating season the air will tend to short circuit between the supply and the ceiling exhaust or return intakes.

Floor Location

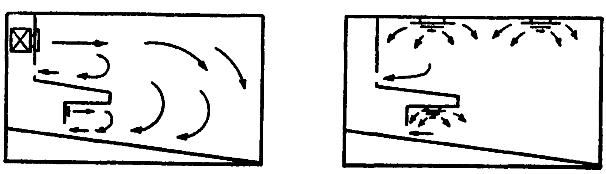
Where ceiling or high sidewall distribution is used for winter heating, floor returns along the exposed wall will tend to improve the heating performance of the system. In general floor locations are collectors of dirt and refuse.

Wall and Door Locations

Depending on their elevation, wall returns have the characteristics of either floor or ceiling returns. In large buildings with many small rooms, the return air may be brought through door grilles or door undercuts into the corridors and then to a common return or exhaust. The pressure drop through door returns should not be excessive (50 per cent of supply outlet pressure); otherwise the air distribution to the room may be seriously unbalanced with the opening or closing of the doors. Outward leakage through doors or windows cannot be counted upon for dependable results. In many cases, particularly in buildings with double glass or hollow glass block walls, forced return and relief systems are essential.

SPECIFIC APPLICATIONS

The two methods shown in Fig. 7 are suitable for application to theaters, churches, and auditoriums. In small or medium size theaters, it is sometimes practical to use sidewall or front wall distribution. For the satisfactory operation of such a system during the winter heating



Rear wall distribution Ceiling distribution

FIG. 7. AIR DISTRIBUTION METHODS FOR THEATERS, CHURCHES, AND AUDITORIUMS

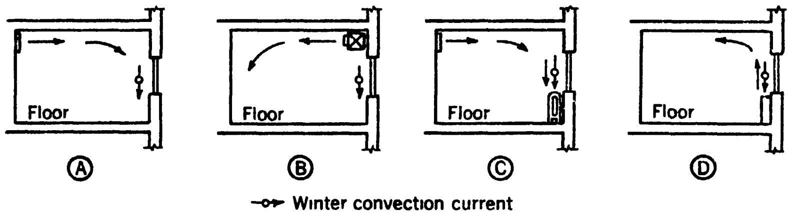


FIG. 8. DISTRIBUTION METHODS FOR SMALL ROOMS

A Satisfactory for cooling Unsatisfactory for heating in severe climates where the outside temperature is consistently below 40 F, and single glass and uninsulated walls are prevalent.

B. Performance slightly poorer than A, unless supplemented by circular diffusers in bottom of the duct.

C. Satisfactory for cooling. Satisfactory for heating if direct radiation is properly controlled.

D. Satisfactory for both cooling and heating. The air should be discharged in a vertical plane parallel to the wall, and should be fanned out in this plane at an angle of 10 to 20 deg with the vertical

period, the returns should be preferably located at the floor level and near the front of the theater to prevent cold spots which may result from exposed wall convection or infiltration from exits. Return intakes may be located higher where the exits and stage have separate means of heating.

Diagrams shown in Fig. 8 illustrate distribution methods for small rooms with exposed walls, such as for offices, hospital rooms, hotel rooms, apartments, etc. The cooling performance of various distribution methods as applied to a small store is shown in Fig. 9.

For specific requirements in connection with air distribution in marine applications see Chapter 38.

BALANCING SYSTEM

In designing an air conditioning system, it should be the aim of the engineer to so proportion the duct system that proper distribution of air to every supply opening will be obtained. Since this is almost impossible to accomplish in practice, it becomes necessary to have means of balancing

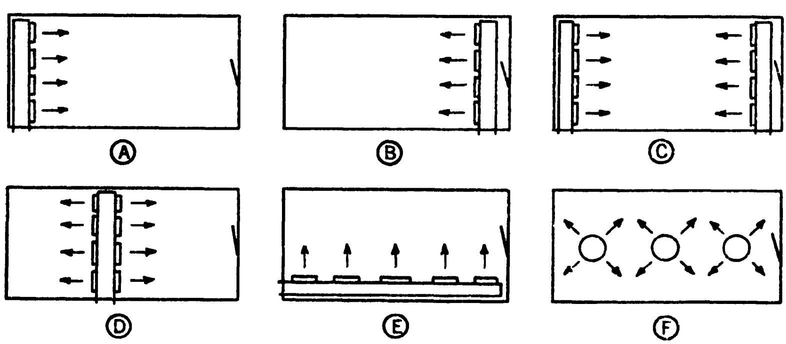


FIG. 9. SMALL STORE COOLING DISTRIBUTION

A. *Rear Wall.* High outlet velocity, possibility of excessive air motion and drafts.

B. *Front Wall.* High outlet velocity, possibility of excessive air motion and drafts. May cause excessive infiltration of outside air.

C. *Front and Rear Walls.* Moderate room air motion, outlet blows should not impinge giving rise to down drafts in center of store.

D. *Center.* Moderate air motion, no impingement of air streams. Good results.

E. *One Side.* Moderate room air motion, should blow toward exposed wall. Good results.

F. *Ceiling* Low room air motion. Good results.

the system to secure the desired amount of air in each space. There are a number of ways in which this may be accomplished, some of which are:

1. Dampers on the supply and return faces.
2. Dampers in the supply and return ducts.
3. Reducing the effective area of some supply openings by blank-offs.
4. Combinations of dampers in both supply and return air.

Dampers on the supply faces themselves are objectionable unless of special design (see Fig. 5), because of their effect on the air stream and noise. Dampers on the return faces are frequently objectionable because of noise. A damper in the supply duct some distance back of the supply opening forms a very satisfactory means of regulating the flow without disturbing distribution across the supply opening face. A damper in the return air duct has the advantage over one immediately behind the face in that it does not tend to create high localized velocities through the face as the latter might do if nearly closed. Blank-offs consisting of pieces of sheet metal covering a portion of the supply opening face can frequently be used satisfactorily, although determination of just what is required is a matter of experiment, and the balancing of the system is not nearly so conveniently accomplished as with dampers. Blank-offs have the further objection that as the area is reduced, pressure builds up and air motion tends to remain constant. Dampers in both supply and return air form the most flexible means of controlling the supply to the room and the static pressure within the room. When feasible, these dampers, particularly those in the supply ducts, should be a substantial distance from the supply opening, and ahead of the acoustic duct lining if used. Due consideration should also be given to the use of the several volume control and uniform distribution devices now available. See *Catalog Data Section.*

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CHAPTER 31

Air Duct Design

Pressure Losses, Friction Losses, Friction Loss Chart, Elbow Friction Losses, Proportioning the Losses, Duct Sizes, Procedure for Duct Design, Velocities, Main Trunk Ducts, Proportioning the Size for Friction, Velocity Method, Equal Friction Method, Duct Construction Details, Duct Heat Loss and Insulation

THE resistance of an air handling system can be computed from the methods and data given in this chapter. The actual resistance for any given installation, however, may vary considerably from the calculated resistance because of variation in the smoothness of materials, the type of joints used and the ability of the mechanics to fabricate in accordance with the design. It is best to select fans and motors of sufficient size to allow a factor of safety. Volume dampers should be installed in each branch outlet to balance the system. It is improbable that the required quantities of air will be delivered at each outlet without adjustment of the dampers, which usually results in a total pressure exceeding that of the design, unless a liberal factor of safety is allowed.

The flow of air due to large pressure differences is most accurately stated by thermodynamic formulae for air discharge under conditions of adiabatic flow, but such formulae are complicated, and the error occasioned by the assumption that the gas density remains constant throughout the flow may be considered negligible when only such pressure differences are involved as occur in ordinary heating and ventilating practice.

In the development of the formulae, diagrams, and tables for the flow of air, use is made of the following basic equation for the flow of fluids:

If H_V be the velocity head in feet of a fluid, and the velocity, V, be expressed in feet per minute, the fundamental equation is

$$V = 60 \sqrt{2g\, H_V}$$

The factor g is the acceleration due to gravity, or 32.17 fps per second.

It is usual to express the head in inches of water for ventilating work and, since the heads are inversely proportional to the densities of the fluids,

$$\frac{H_V}{\frac{h_V}{12}} = \frac{62.4}{d}$$

or

$$H_V = 5.2\,\frac{h_V}{d}$$

therefore,

$$V = 1096.5 \sqrt{\frac{h_V}{d}} \qquad (1)$$

where

V = velocity, feet per minute.

h_V = velocity head or pressure, inches of water.

d = weight of air, pounds per cubic foot.

For dry air (70 F and 29.921 in. Hg barometer) d = 0.075 lb per cubic foot[1]. Substituting this value in Equation 1:

---

[1]See Chapter 47 for definition of standard air.

$$V = 1096.5 \sqrt{\frac{h_v}{0.075}} = 4005 \sqrt{h_v} \tag{2}$$

The relation of air velocity and velocity head expressed in Equation 2 is shown diagrammatically in Fig. 1 for air at 70 F and 29.92 in. Hg barometer.

The drop in pressure in air distributing systems is due to the *dynamic* losses and the *friction* losses. The friction losses for turbulent flow (which occur in all practical air flow problems) are due to the friction of air against the sides of the duct and to internal friction between air molecules. The dynamic losses are those due to the change in the direction or in the velocity of air flow.

Dynamic losses occur principally at the entrance to the piping, in the elbows, and wherever a change in velocity occurs. The entrance loss is the difference between the actual pressure required to produce flow and the pressure corresponding to the flow produced; it may vary from 0.1 to 0.5 times the velocity head. The pressure loss in elbows must also be allowed for in the design.

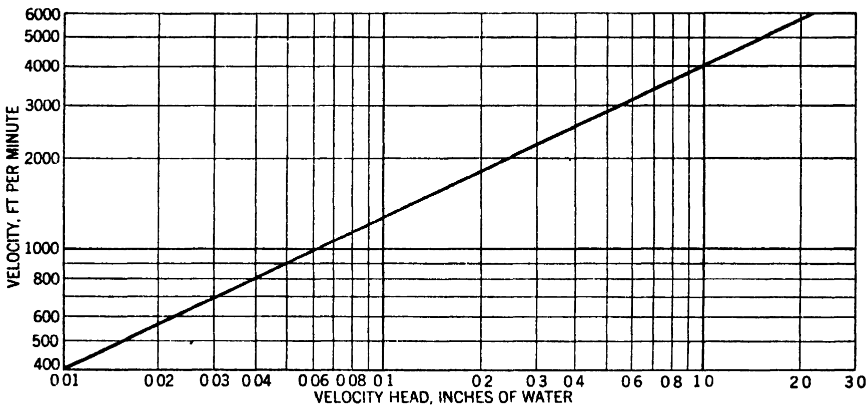

FIG. 1. RELATION BETWEEN VELOCITY AND VELOCITY HEAD FOR DRY AIR

FRICTION LOSSES

A study of the frictional resistance to the flow of air in ducts was begun by the A.S.H.V.E. Research Laboratory in 1938. This study resulted in modifications of the Fanning friction loss formula, for 100 ft of round galvanized iron duct and for air at standard conditions[2]:

For round duct with no joints,

$$H_s = \frac{1.157}{D^{1.212}} \left(\frac{V}{4000}\right)^{1.83} \tag{3}$$

For round duct with 40 joints per 100 ft,

$$H_s = \frac{1.48}{D^{1.287}} \left(\frac{V}{4000}\right)^{1.83} \tag{4}$$

[2]A.S.H.V.E. REPORT No. 1105—Frictional Resistance to the Flow of Air in Straight Ducts, by F. C. Houghten, J. B. Schmieler, J A Zalovick and N Ivanovic (A S H V E TRANSACTIONS, Vol 45, 1939, p. 35). A.S.H.V.E. REPORT No. 1154—Analysis of Factors Affecting Duct Friction, by J. B. Schmieler, F. C. Houghten and H. T. Olson (A.S.H.V.E. TRANSACTIONS, Vol 46, 1940, p. 193).

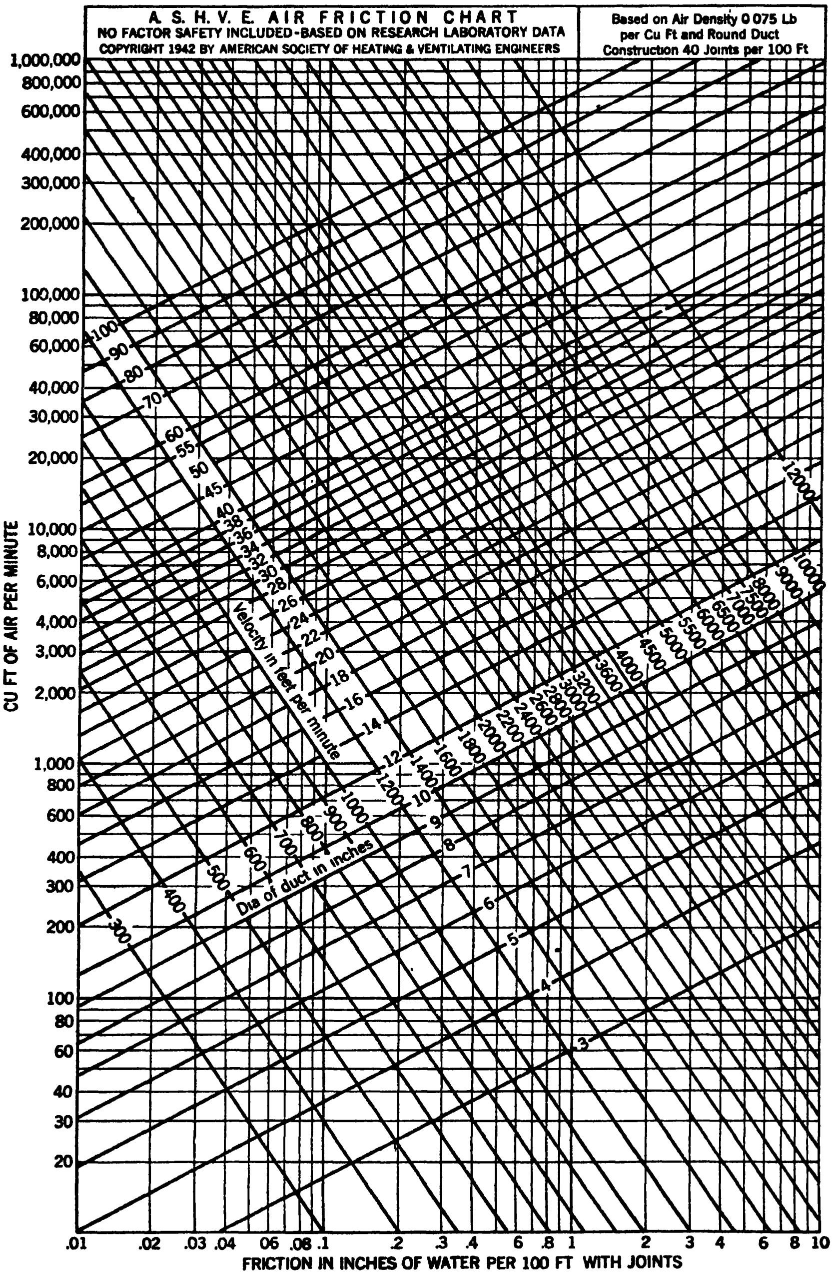

Fig. 2. Friction of Air in Pipes

Note As no safety factor is included in chart due allowance must be made. For field application multiply pressure drop from chart by not less than 1.1.

where

H_s = friction loss, inches of water at standard conditions.
V = velocity of air, feet per minute.
D = diameter of duct, feet.

The chart shown in Fig. 2 was constructed from Equation 4 and therefore applies only for round galvanized iron duct of good construction with 40 joints per 100 ft, and for air at standard conditions. *No factor of safety has been applied.* In view of the many variations that may occur in duct construction and application, *it is recommended that a factor of safety be used* which, in the judgment of the engineer, will make due allowance for these variations. In the Laboratory tests the variation found in pressure loss between the best joints and the worst joints was approximately 10 per cent. This would suggest a *minimum factor of safety of 10 per cent.*

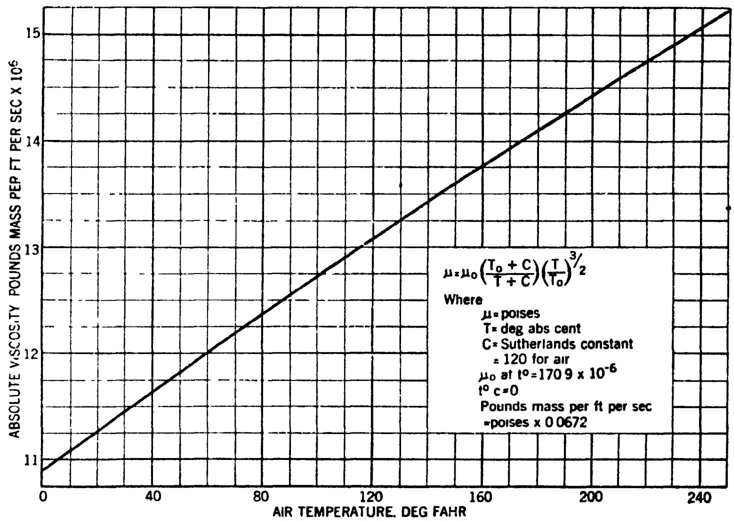

FIG. 3. TEMPERATURE—VISCOSITY CURVE FOR DRY AIR

The friction loss varies with the surface characteristics of a duct and in case a *rough conduit* of tile, brick or concrete is used, a factor of *about 35 per cent* should be added to the friction values obtained from Fig. 2.

Since the friction chart applies only for standard conditions, it is necessary to apply correction factors for other than standard conditions. These corrections are:

$$H_A = H_s \times S \left(\frac{\gamma_A}{\gamma_s}\right)^{0.17} \tag{5}$$

where

H_A = friction loss, inches of water at actual conditions
S = ratio of density of air at actual conditions to density of air at standard conditions.
γ_A = kinematic viscosity at actual conditions.
γ_s = kinematic viscosity at standard conditions.

$$\text{Kinematic Viscosity} = \frac{\mu}{\rho}.$$

where

μ = absolute viscosity, pounds per foot second (see Fig. 3)
ρ = density, pounds per cubic foot.

The absolute viscosity of dry air at various temperatures is given in Fig. 3. It is assumed that the viscosity is not appreciably affected by the moisture content.

For temperatures ordinarily used in heating, ventilating and air conditioning work, the correction for viscosity may be neglected without serious error. The correction equation would then be simplified to:

$$H_A = H_s \times S \tag{6}$$

Example 1. Assume that it is desired to circulate 10,000 cfm of air through 75 ft of 24 in. diameter pipe. Find 10,000 cfm on the left scale of Fig. 2 and move horizontally right to the diagonal line marked 24 in. The other intersecting diagonal shows that the velocity in the pipe is 3200 fpm. Directly below the intersection it is found that the friction per 100 ft is 0.41 in.; then for 75 ft the friction will be 0.75 × 0.41 = 0.31 in. In a like manner any two variables may be determined by the intersection of the lines representing the other two variables.

Circular Equivalents of Rectangular Ducts

Where rectangular ducts are used it is frequently desirable to know the equivalent diameter of round pipe to carry the same capacity and have the same friction per foot of length. Table 1 gives directly the circular equivalents of rectangular ducts for equal friction and capacity, which are based on values determined from Equation 7:

$$d = 1.265 \sqrt[5]{\frac{(a\,b)^3}{a + b}} \tag{7}$$

where

a = one side of rectangular pipe, feet or inches.
b = other side of rectangular pipe, feet or inches.
d = equivalent diameter of round pipe for equal friction per foot of length to carry the same capacity, feet or inches.

Rectangular equivalents of round ducts are also given in the curves of Fig. 4 which are plotted from data based on Equation 7. To use the chart, locate the diagonal curve giving the diameter of the round duct. The width and height of an equivalent rectangular or square duct may then be read as the abscissa and the ordinate of any point of the curve.

Multiplying or dividing the length of each side of a pipe by a constant is the same as multiplying or dividing the equivalent round size by the same constant. Thus, if the circular equivalent of an 80 x 24-in. duct is

Table 1. Circular Equivalents of Rectangular Ducts for Equal Friction

Side Rectangular Duct	8	8.5	9	9.5	10	10.5	11	11.5	12	12.5	13	13.5	14	14.5	15	15.5	16
3	5.2	5.4	5.5	5.7	5.8	5.9	6.0	6.2	6.3	6.4	6.5	6.6	6.7	6.8	6.9	7.0	7.1
3.5	5.7	5.9	6.0	6.2	6.3	6.5	6.6	6.7	6.9	7.0	7.1	7.3	7.4	7.5	7.6	7.7	7.8
4	6.1	6.3	6.5	6.7	6.8	7.0	7.1	7.2	7.4	7.5	7.7	7.8	7.9	8.1	8.2	8.3	8.4
4.5	6.5	6.7	6.9	7.1	7.2	7.4	7.6	7.7	7.9	8.0	8.2	8.4	8.5	8.6	8.7	8.9	9.0
5	6.9	7.1	7.3	7.5	7.7	7.8	8.0	8.2	8.3	8.5	8.7	8.8	8.9	9.1	9.2	9.4	9.5
5.5	7.3	7.5	7.7	7.8	8.1	8.3	8.5	8.6	8.8	9.0	9.2	9.4	9.5	9.6	9.8	9.9	10.1

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION[a]—(Continued)

Side Rectangular Duct	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	24
8	6.1	6.9	7.6	8.2	8.8															
9	6.5	7.3	8.0	8.7	9.3	9.9														
10	6.8	7.7	8.4	9.2	9.8	10.4	11.0													
11	7.1	8.0	8.8	9.6	10.2	10.9	11.5	12.1												
12	7.4	8.3	9.2	10.0	10.7	11.4	12.0	12.6	13.2											
13	7.6	8.7	9.6	10.4	11.1	11.8	12.5	13.1	13.7	14.3										
14	7.9	8.9	9.9	10.8	11.5	12.3	12.9	13.6	14.3	14.9	15.4									
15	8.2	9.2	10.2	11.1	11.9	12.7	13.4	14.1	14.7	15.3	16.0	16.5								
16	8.4	9.5	10.5	11.4	12.3	13.1	13.8	14.5	15.2	15.8	16.5	17.1	17.6							
17	8.6	9.8	10.8	11.8	12.6	13.5	14.2	15.0	15.7	16.3	17.0	17.6	18.2	18.7						
18	8.9	10.0	11.1	12.1	13.0	13.8	14.6	15.4	16.1	16.8	17.4	18.1	18.7	19.2	19.8					
19	9.1	10.3	11.4	12.4	13.3	14.2	15.0	15.8	16.5	17.2	17.9	18.6	19.2	19.8	20.4	20.9				
20	9.3	10.5	11.6	12.7	13.6	14.5	15.4	16.2	17.0	17.6	18.4	19.0	19.7	20.3	20.9	21.5	22.0			
22	9.7	11.0	12.1	13.2	14.2	15.2	16.1	16.9	17.8	18.5	19.2	19.9	20.6	21.3	21.9	22.5	23.1	23.6	24.2	
24	10.0	11.4	12.6	13.8	14.8	15.8	16.8	17.6	18.5	19.3	20.0	20.8	21.5	22.2	22.8	23.5	24.0	24.7	25.2	26.4
26	10.4	11.8	13.1	14.3	15.4	16.4	17.3	18.3	19.2	20.0	20.8	21.6	22.3	23.0	23.8	24.4	25.1	25.7	26.3	27.5
28	10.8	12.2	13.5	14.8	15.9	17.0	18.0	19.0	19.8	20.7	21.5	22.4	23.1	23.9	24.6	25.3	26.0	26.6	27.3	28.5
30	11.0	12.6	13.9	15.2	16.4	17.5	18.5	19.5	20.5	21.4	22.2	23.1	23.9	24.7	25.4	26.2	26.8	27.5	28.2	29.5
32	11.3	12.9	14.3	15.6	16.9	18.0	19.1	20.1	21.1	22.0	22.9	23.8	24.6	25.4	26.2	27.0	27.7	28.4	29.1	30.5
34	11.6	13.2	14.7	16.1	17.3	18.5	19.6	20.7	21.6	22.6	23.5	24.4	25.3	26.2	26.9	27.7	28.5	29.2	30.0	31.3
36	11.9	13.6	15.1	16.4	17.7	19.0	20.1	21.2	22.2	23.2	24.2	25.1	26.0	26.8	27.7	28.5	29.3	30.0	30.8	32.2
38	12.2	13.9	15.4	16.8	18.2	19.4	20.6	21.7	22.8	23.8	24.8	25.8	26.7	27.5	28.4	29.2	30.0	30.8	31.5	33.1
40	12.5	14.3	15.7	17.2	18.6	19.8	21.1	22.2	23.3	24.4	25.4	26.4	27.3	28.2	29.1	29.9	30.8	31.6	32.4	33.9
42	12.7	14.5	16.1	17.6	19.0	20.3	21.6	22.7	23.8	24.9	25.9	26.9	27.9	28.8	29.8	30.7	31.4	32.2	33.0	34.5
44	13.0	14.8	16.4	18.0	19.4	20.7	22.0	23.1	24.3	25.4	26.5	27.5	28.5	29.5	30.3	31.2	32.1	32.9	33.7	35.3
46	13.3	15.1	16.7	18.4	19.8	21.1	22.4	23.6	24.8	25.9	27.0	28.1	29.1	30.1	31.0	31.9	32.8	33.8	34.6	36.2
48	13.5	15.4	17.0	18.7	20.1	21.5	22.8	24.1	25.2	26.4	27.5	28.6	29.6	30.5	31.6	32.5	33.4	34.3	35.2	37.0
50	13.7	15.7	17.3	19.0	20.4	21.9	23.2	24.5	25.7	26.9	28.0	29.2	30.3	31.3	32.2	33.1	34.1	35.0	35.9	37.6
52	13.9	15.9	17.6	19.2	20.8	22.2	23.6	24.9	26.2	27.4	28.5	29.6	30.7	31.8	32.9	33.8	34.7	35.6	36.5	38.3
54	14.1	16.1	17.9	19.6	21.1	22.6	24.0	25.3	26.6	27.8	29.0	30.1	31.2	32.3	33.4	34.4	35.3	36.3	37.2	38.9
56	14.3	16.3	18.2	19.9	21.5	22.9	24.4	25.7	27.0	28.3	29.5	30.6	31.7	32.8	33.9	34.9	35.9	36.9	37.8	39.6
58	14.6	16.6	18.4	20.2	21.8	23.3	24.7	26.1	27.4	28.7	30.0	31.1	32.2	33.3	34.4	35.4	36.4	37.4	38.4	40.3
60	14.7	16.8	18.7	20.4	22.1	23.6	25.1	26.5	27.8	29.1	30.5	31.6	32.7	33.8	34.9	36.1	37.1	38.1	39.1	40.9
62	15.0	17.0	19.0	20.7	22.4	24.0	25.5	26.9	28.2	29.5	30.9	32.1	33.2	34.3	35.4	36.6	37.7	38.7	39.6	41.6
64	15.1	17.3	19.2	21.0	22.7	24.3	25.9	27.3	28.6	29.9	31.3	32.6	33.7	34.8	35.9	37.1	38.2	39.2	40.2	42.2
66	15.3	17.5	19.5	21.2	23.0	24.6	26.2	27.7	29.0	30.3	31.7	33.0	34.2	35.3	36.4	37.6	38.7	39.8	40.8	42.8

[a]Additional sizes: 4 × 5 = 4.9; 4 × 6 = 5.4; 4 × 7 = 5.8; 5 × 5 = 5.5; 5 × 6 = 6.3; 5 × 7 = 6.5.

Table 1. Circular Equivalents of Rectangular Ducts for Equal Friction—(Concluded)

Side Rectangular Duct	26	28	30	32	34	36	38	40	42	44	46	48
26	28.6											
28	29.7	30.8										
30	30.7	31.9	33.0									
32	31.7	32.9	34.1	35.2								
34	32.7	33.9	35.1	36.3	37 4							
36	33.7	34.9	36.1	37.3	38.5	39.6						
38	34.6	35.9	37.1	38.4	39.5	40 7	41.8					
40	35.3	36.7	38.0	39.3	40.5	41.7	42 9	44.0				
42	36.0	37.6	39.0	40.3	41.5	42.7	44.0	45.1	46 2			
44	36.9	38.5	39.9	41.2	42.5	43.7	44.9	46.1	47.2	48.4		
46	37.8	39.3	40.8	42.2	43.5	44.8	46.0	47.2	48.4	49.5	50.6	
48	38.5	40.0	41.5	43.0	44.4	45.6	46.9	48 1	49 3	50.5	51.6	52.8
50	39.2	40.8	42.3	43.8	45.2	46.5	47.9	49.1	50.4	51.6	52.9	54.0
52	40.0	41.6	43.1	44.7	46.1	47.5	48.9	50.1	51.3	52.5	53.8	55.0
54	40.7	42.4	44.0	45.5	47.0	48.4	49.9	51.1	52.3	53 5	54 8	56.0
56	41.3	43.0	44.6	46 2	47.7	49.1	50 6	52 0	53 3	54 6	55.9	57.0
58	42.1	43.8	45.4	47.0	48 5	50.0	51 5	52.9	54.2	55.5	56 8	58.0
60	42.7	44.5	46.1	47.8	49 3	50 9	52.3	53.8	55.0	56.4	57.7	58.9
62	43.4	45.1	46.8	48.4	50 0	51.7	53.0	54.5	55.9	57.2	58.5	59.7
64	44.0	45.8	47.5	49.2	50 9	52.4	53.9	55 4	56.8	58.1	59 4	60.6
66	44.7	46.5	48.2	30.0	51.6	53.1	54.7	56 2	57.6	59 1	60.4	61.6
68	45.3	47.2	48.9	50 7	52 2	53 8	55 5	56 9	58.4	59.9	61 3	62 6
70	46.0	47.8	49.5	51.3	52 9	54 5	56 2	57.7	59 1	60.6	62 1	63.5
72	46.5	48 4	50.1	51.9	53 7	55.4	57 0	58.7	60.0	61.3	63.0	64.5

Side Rectangular Duct	50	54	60	66	72	78	84	88
50	55.0							
52	56.1							
54	57.2	59.4						
56	58.3	60.5						
58	59.3	61.6						
60	60.3	62.7	66.0					
62	61.3	63.7	67.1					
64	62.2	64.7	68.2					
66	63.2	65.7	69.3	72.6				
68	64.1	66.6	70 3	73.7				
70	65.0	67.6	71.3	74.8				
72	65.9	68.5	72.3	75.9	79.2			
74	66.8	69.4	73.3	76.9	80.3			
76	67.6	70.3	74.2	77.9	81.4			
78	68.4	71.2	75.2	78.9	82.5	85.8		
80	69.2	72.1	76.1	79.9	83.6	86.9		
82	70 1	73.0	77.1	80.9	84.6	88.0		
84	70.9	73.8	78.0	81.9	85.6	89.1	92.4	
86	71.7	74.6	78.9	82.9	86.6	90.2	93.5	
88	72.5	75.5	79.8	83.9	87.5	91.2	94.6	96.8
90	73.3	76.3	80.6	84.7	88.5	92.2	95.7	97.9
92	74.1	77.1	81.4	85.6	89.5	93.2	96.7	99.0
94	74.8	77.8	82.2	86.5	90.4	94.2	97.8	100.1
96	75.5	78.7	83.0	87.4	91.3	95.2	98.8	101.2

required, it will be twice that of a 40 x 12-in. duct, or $2 \times 23.3 = 46.6$ in.

Elbow Friction Losses

It is customary to express the dynamic and friction losses in elbows as equal to a number of diameters of round pipe, or a number of widths of

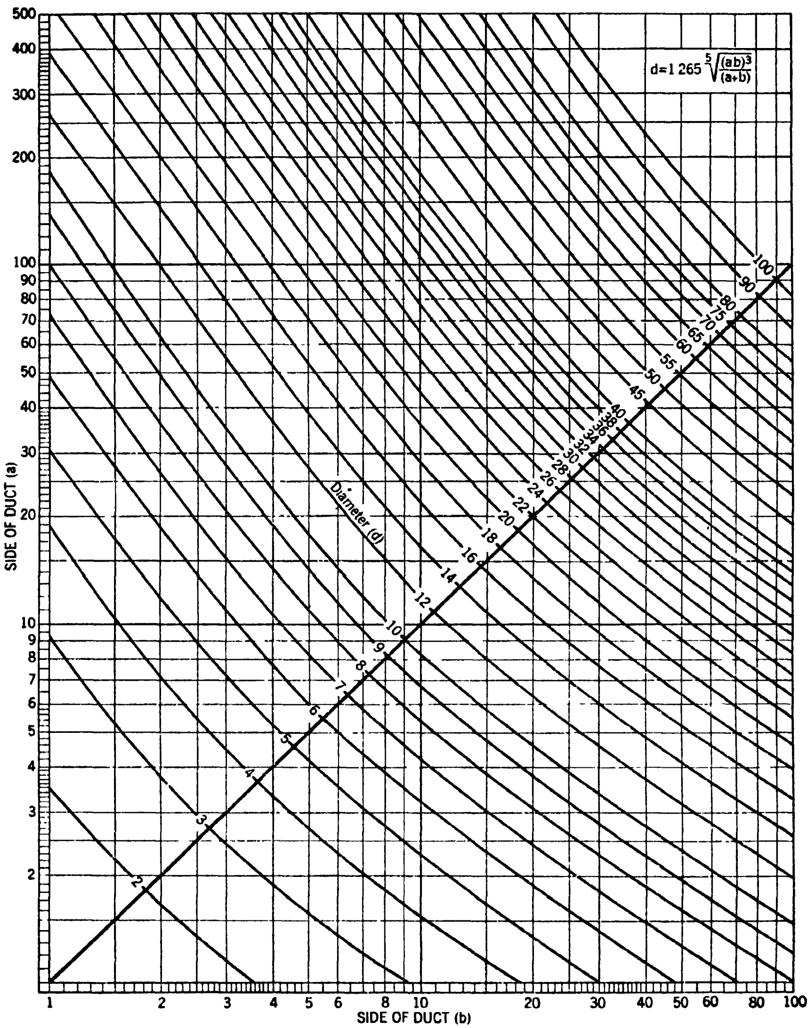

FIG. 4. RECTANGULAR EQUIVALENTS OF ROUND DUCTS

rectangular pipe, or equivalent length of duct[3]. The curves in Fig. 5 are arranged to read the number of diameters or widths for determining the lineal feet of pipe having a frictional resistance equivalent to the pressure

[3]A.S.H.V.E. RESEARCH REPORT No. 1211—Pressure Loss Caused by Elbows in 8-Inch Round Ventilating Duct, by M. C Stuart, C. F. Warner and W. C. Roberts (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 335).

drop in the elbows. Curves *B* and *C* are based on tests of round and square elbows[4] of ordinary good sheet metal construction.

Values obtained from Curve *A* should be used when there is any doubt as to quality of duct construction. It is suggested that this curve be used for rectangular elbows and five-piece elbows as it will thus allow an additional factor of safety without seriously affecting the design.

As indicated on the chart, long radius elbows will offer much less resistance to the flow of air than short radius elbows. Experience has shown that good results may be expected when the radius to the center of the elbow is 1.5 times the pipe diameter or duct width parallel to the radius. Examination of the curve will indicate that little advantage is to be gained by selecting elbows having a centerline radius of more than two diameters[5]. Elbows having a radius of more than three diameters show a

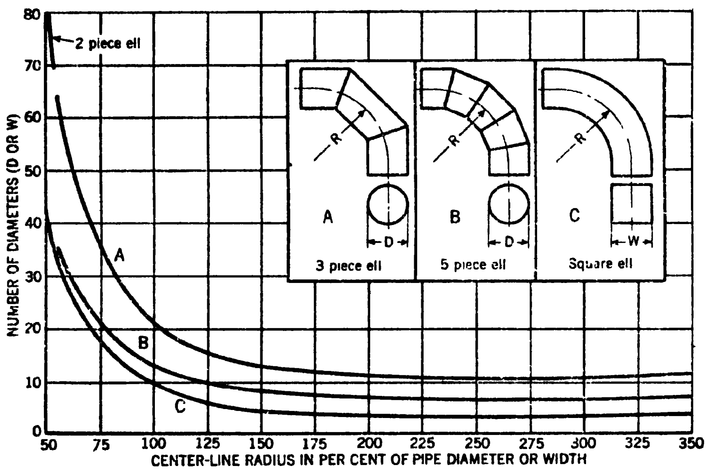

Fig. 5. Loss of Pressure in Elbows

slightly increased resistance due to the increased length of pipe but, when used, they reduce the over-all resistance of the system and therefore should not be avoided.

Where space conditions necessitate the use of short radius or miter elbows in square or rectangular duct work, turning vanes should be used to reduce the pressure losses. Rough or raw edges on the vanes should be avoided to prevent objectionable noise. Typical types of vanes are shown in Table 2 with the total equivalent length of duct that may be used in estimating the resistance of each type.

The pressure loss through elbows of less than 90 deg may be assumed to be directly proportional to the ratio of the angle through which the turn is made. The resistance will vary widely for the large degree turns depending upon the aspect ratio and the length of straight pipe between the elbows, but for practical purposes, it may be assumed that the ratio

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[4]Loss of Pressure Due to Elbows in the Transmission of Air Through Pipes or Ducts, by F. L. Busey (A.S.H.V.E. Transactions. Vol 19, 1913, p. 366).

[5]Pressure Losses in Rectangular Elbows, by R D Madison and J R. Parker (*Heating, Piping and Air Conditioning*, July, p. 365, August, p. 427, September, p. 483, 1936)

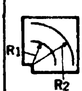
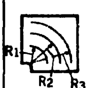
remains proportional to the angle through which the turn is made. Reverse 90 deg elbow turns should be avoided wherever possible but, where used, the friction of the elbows indicated in Fig. 5 should be doubled for the second elbow.

PROPORTIONING THE LOSSES

The entrance loss through the outside air intake louvers will vary with the design of the louvers and method of connection to the system. The louvers and connecting duct will have a friction resistance of from 0.25 to 1.00 times the velocity pressure. Therefore, the total entrance loss will

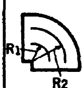
TABLE 2. EFFECT OF VANES ON PRESSURE LOSS OF 7-INCH SQUARE VENTILATING DUCT[a]
Expressed in feet of total equivalent length of duct (ELD)

SQUARE MITER ELBOW

	Radius Ratio R_1/W	0	2	4	6	8	10
	ELD, ft	41.1	30 5	27 5	30 1	37 7	38.5
	Radius Ratio R_1/W	0	.2	.3			
	Radius Ratio R_2/W	0	.4	.5			
	ELD, ft	41 1	23 5	23.3			
	Radius Ratio R_1/W	0	2	.3			
	Radius Ratio R_2/W	0	4	.5			
	Radius Ratio R_3/W	0	.6	7			
	ELD, ft	41 1	20 7	22 2			

VANE	A	B	C	D
				
ELD, ft	21 8	17 0	17 9	17.8

STANDARD ELBOWS WITH VARIOUS RADII

	Radius Ratio R_1/W	0	.2	4	.6	.8	1.0
	ELD, ft	39 7	23.3	22 0	25 7	28 9	39 7
	Radius Ratio R_1/W	0	.2	.3	.4	.5	.6
	Radius Ratio R_2/W	0	4	.5	.6	.7	.8
	ELD, ft	39 7	20 0	22.0	23 0	23.8	25.7
 1½" I. R.	Radius Ratio R_1/W	0	.4	.6	.8	1.0	1 2
	ELD, ft	25 3	17 7	16 5	18.7	23.5	25 6
 4" I R	Radius Ratio R_1/W	0	.7	.8	.9	1.0	1.2
	ELD, ft	14.2	13.3	13.0	12 7	12 5	12 7

[a]For more complete data see A S H V E RESEARCH REPORT No 1216—Effect of Vanes in Reducing Pressure Loss in Elbows in 7-Inch Square Ventilating Duct, by M. C. Stuart, C. F. Warner and W. C. Roberts (A S H V E TRANSACTIONS, Vol 48, 1942, p 409).

Note A: Vane *A* made up of a large number of small splitters; *B* made up of a small number of large splitters bent on a large radius; *C* hollow vanes having different outside and inside curvature; and *D* four splitters with R/W = 0 4. Elbow same as *D* except 2 in trailing edge on the end of each splitter, ELD in feet = 17 0

Note B: The air velocity has no effect on the loss of elbows when the loss is expressed as equivalent length of duct

vary from 1.25 to 2.00 h_v. Common practice is to use 1.5 h_v for a 75 per cent free area louver with connecting duct having 15 deg tapered sides. Wherever air passes through a plenum space having a negligible velocity, allowance must be made for the loss in velocity head. This may be taken as the velocity head corresponding to the difference in velocities in the plenum and the duct. Where the ducts are very smooth with long transformation fittings, a regain in static pressure is sometimes allowed, but generally ordinary construction does not warrant a consideration of this factor, and it is customary to neglect it. When it is allowed, the regain is estimated at one-half the difference between the velocity pressure at the fan outlet and at the last run of pipe.

Other losses of pressure occur through the heating units, at the air washer and at air filters. In ordinary practice in ventilation work it is

usual to keep the sum of the duct losses one-third to one-half and the loss through the other units at less than one-half of the static pressure. The remainder is then available for producing velocity. In the design of an ideal duct system, all factors should be taken into consideration and the air velocities proportioned so that the resistance will be practically equal in all ducts regardless of length.

DUCT SIZES

Ducts and flues for gravity circulation must be sized so that the friction loss will not exceed 50 per cent of the available aspirating effect due to the temperature and height of the column of heated air. Duct systems for mechanical circulation may be sized so as to have much higher pressure losses than gravity systems. The total pressure of these systems is limited to the available pressure from the fan used.

The general rules to be followed in the design of a duct system are enumerated herewith:

1. The air should be conveyed as directly as possible at reasonable velocities to obtain the results desired with greatest economy of power, material and space.
2. Sharp elbows and bends should be avoided unless turning vanes are used.
3. Transformation pieces should be made as long as possible. The angle between the sides and axis of the duct should never exceed 30 deg and, where possible, 15 deg should be made the maximum.
4. Especial care should be taken to maintain a true cross-section and not to restrict the air flow either in transformation pieces or in elbows
5. Rectangular ducts or flues should be made as nearly square as possible. Good practice limits the ratio between the long side and the short side to 3 to 1 In no case should this ratio exceed 10 to 1.
6. Wherever possible, ducts should be constructed of smooth material such as sheet metal. Where masonry ducts are used, proper allowance for the surface coefficient should be made.
7. The use of furred spaces, spaces between joists, etc., should be avoided unless lined with sheet metal.

Procedure for Duct Design

The general procedure for designing a duct system is outlined in the several items listed herewith:

1. Study the plan of the building and draw in roughly the most convenient system of ducts, taking cognizance of the building construction, avoiding all obstructions in steel work and equipment, and at the same time maintaining a simple design
2. Arrange the positions of duct outlets to insure the proper distribution of air.
3. Divide the building into zones and proportion the volume of air necessary for each zone.
4. Determine the size of each outlet, based on the volume as obtained in the preceding paragraph, for the proper outlet velocity and throw.
5. Calculate the sizes of all main and branch ducts by either of the following two methods:
   a. *Velocity Method.* Arbitrarily fix the velocity in the various sections, reducing the velocity from the point of leaving the fan to the point of discharge to the room. In this case the pressure loss of each section of the duct is calculated separately and the total loss found by adding together the losses of the various sections of the continuous run.
   b. *Friction Pressure Loss Method.* Proportion the duct for equal friction pressure loss per foot of length.
6. Calculate the friction for the duct offering the greatest resistance to the flow of air, which resistance represents the static pressure which must be maintained in the fan outlet or in the plenum space to insure distribution of air in the duct system. The duct having the greatest resistance will usually be that having the longest run, although not necessarily so.

Air Velocities

The air velocities given in Table 3 have been found to give satisfactory results in engineering practice. Where the higher velocities are used, the ducts should be cross-braced to prevent breathing, buckling or vibration. High velocities at one point in the system offset the effect of proper design in all other parts of the system; hence the importance of air velocities, elbow design, location of dampers, fan connections, grille and register approach connections, and similar attention to details. For industrial buildings, noise is seldom given much consideration, and main duct velocities as high as 2800 or 3000 fpm are sometimes used but, when these velocities are used, due consideration should be given to duct design, resistance pressure, fan efficiencies and motor horsepower. For department stores and similar buildings, 2000 to 2200 fpm are sometimes

TABLE 3. RECOMMENDED AND MAXIMUM DUCT VELOCITIES

DESIGNATION	RECOMMENDED VELOCITIES, FPM			MAXIMUM VELOCITIES, FPM		
	Residences	Schools, Theaters, Public Buildings	Industrial Buildings	Residences	Schools, Theaters, Public Buildings	Industrial Buildings
Outside Air Intakes[a]	700	800	1000	800	900	1200
Filters[a]	250	300	350	300	350	350
Heating Coils[a]	450	500	600	500	600	700
Air Washers	500	500	500	500	500	500
Suction Connections	700	800	1000	900	1000	1400
Fan Outlets	1000–1600	1300–2000	1600–2400	1700	1500–2200	1700–2800
Main Ducts	700–900	1000–1300	1200–1800	800–1000	1100–1400	1300–2000
Branch Ducts	600	600–900	800–1000	700	800–1000	1000–1200
Branch Risers	500	600–700	800	650	800–900	1000

[a]These velocities are for total face area, not the net free area.

used in main ducts where noise is not objectionable and space conditions warrant it. Wherever velocities higher than those shown in Table 3 are used, it is essential that the ducts should be of heavier gages, have additional bracing and be carefully constructed for a minimum resistance.

Where the high velocity diffusing outlets are used, the duct velocity should not be less than the throat velocity of the diffusers, as dynamic losses occur wherever velocities are stepped up or down. One recent trend in grille design is toward the use of much higher grille and branch duct velocities. Some installations have been made with velocities as high as 1600 fpm in branches and through the net area of grilles, but many of these have proven unsatisfactory because of noise and drafts.

Grille manufacturers publish selection tables which size the grilles for volume of air, temperature differential, and distance of throw. In following these tables, maximums should be avoided and the manner in which the duct connects to the grille should be given careful consideration. Most of the selection tables are based on straight approach to the grille. Elbow connections to supply grilles should be provided with turning vanes to equalize the face velocity. See Chapter 30 for a discussion of grilles.

Fan outlet velocities are discussed in Chapter 29 and will not be dealt with here except to indicate that fan noises should be given proper consideration.

Main Trunk Ducts

Main trunk ducts with branches are commonly used to convey the air from the fan to the grille or register outlets in preference to individual ducts from the fan to these outlets. The velocities in these ducts and branches vary according to the nature of the installation and the degree of quietness desired. The recommended velocities in Table 3, with good construction, should give satisfactory results. The maximum velocities indicated should not be used except where noise is not a deciding factor.

Velocity Method

The velocity method of designing a duct system involves arbitrarily selecting velocities at various sections of the duct system with the highest velocities generally chosen at the fan and progressive lower velocities toward the duct openings to the room. To find the total static pressure against which the fan must operate, the static pressure loss of each section must be calculated separately and the total loss found by adding the individual losses of the various sections of the run having the highest resistance. Usually this is the longest run but in some cases a shorter run may have more elbows, transformations, booster heaters, etc., which will cause it to have a higher resistance pressure. This method requires judgment and experience in choosing the proper velocities to approach equal friction for all lengths of run but many engineers believe that the velocity method is handier to use than other methods and will give satisfactory results for most practical applications. The air velocities given earlier in this chapter are helpful in choosing proper velocities. Adjustable dampers or splitters are used to regulate air quantities delivered.

Equal Friction Method

The equal friction method of design is sometimes preferred because it does not require nearly so much judgment and experience in selecting the proper velocities in the various sections of a system. The usual procedure in this method of design is to select the main duct velocity to be consistent with good practice from a standpoint of noise for a particular type of building. This velocity should be less than the fan outlet velocity. All main ducts and branch ducts are sized for equal friction by the use of Fig. 2 and Table 1 or Fig. 4.

In cases where the fan or factory assembled air conditioning unit has a limited external resistance, it is necessary to divide the available resistance by the total equivalent length of the longest or most complicated run of duct to determine the resistance per 100 ft and then to size all ducts at this resistance value, which will automatically determine the duct velocities and give the desired total duct resistance. A further refinement which is sometimes used in large systems is to size each branch duct so that it has a resistance equal to the resistance of the main system at the point of juncture. Even when this refinement is added, regulating dampers are recommended in each branch.

After the duct system is designed the frictional resistance is calculated and tabulated together with the resistance of all component parts. The fan is then selected for the required volume of air, static pressure and outlet velocity.

Example 2. Fig. 6 shows a typical layout of an air distribution system which is applicable for ventilation of hotel dining rooms and offices. The volume of air in cubic feet per minute for the room is determined on the basis of the number of air changes per hour required. In the example shown, the room ventilated is a hotel dining room 135 ft x 85 ft x 15 ft. A 7½-min air change (8 air changes per hour) is assumed for proper ventilation, giving 22,935 cfm as the air required.

The free area of the outdoor air inlet is based on a velocity of 1000 fpm or 22,935 ÷ 1000 = 22.94 sq ft. The main duct velocity selected from Table 3 is 1250 fpm which gives a main duct area of 22,935 ÷ 1250 = 18.354 sq ft (60 × 44 in.). From Table 1 a 60 × 44 in. duct is approximately equivalent to 56 in. diameter.

Referring to Fig. 2, a volume of 22,935 cfm through a 56 in. diameter duct gives a resistance of 0.028 in. per 100 ft. The amount of air to be handled by each section of pipe is shown in Fig. 6, and by locating each of these values on the 0.028 in. friction line,

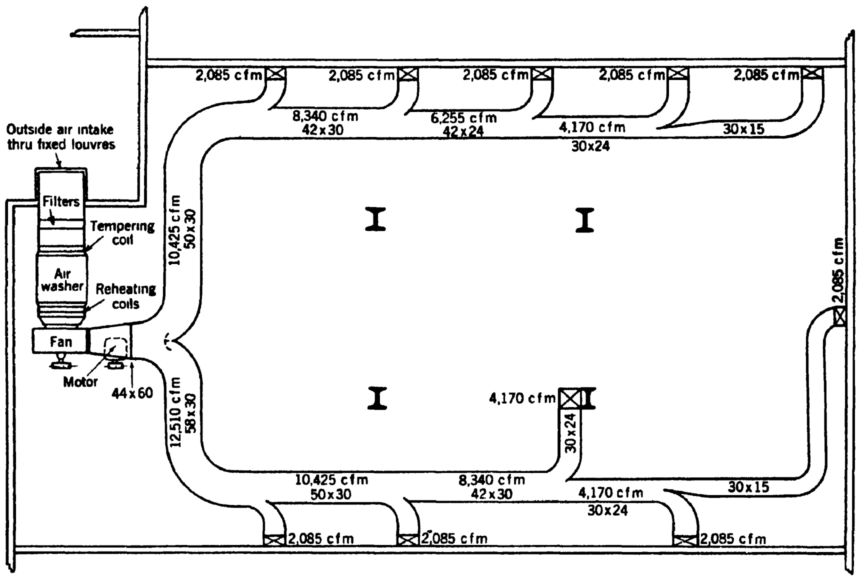


FIG. 6. TYPICAL LAYOUT OF AIR DISTRIBUTION SYSTEM

the round pipe sizes are obtained and then, referring to Table 1, the equivalent rectangular sizes are selected as shown in Table 4.

The pressure at the outlets nearest the fan will be greater than at the pipes farther along the run so that the former will tend to deliver more than the calculated amount of air. To remedy this condition, volume regulating dampers should be located at the base of each riser, or in each branch duct, and adjusted for proper distribution. At points where branches leave the main it may be advisable, depending upon the nature of the installation, to install adjustable splitters similar to that shown in Fig. 6 where the main duct divides into the 58 x 30 in. and 50 x 30 in. branches.

Resistance Losses for the System

(1) Outdoor air intake, 1000 fpm velocity (1.5 heads × 0.0625) .... 0.094 in.

(2) Filters (from manufacturer's tables) .... 0.250 in.

(3) Tempering coil loss (from manufacturer's tables) .... 0.074 in.

(4) Air washer loss (from manufacturer's tables) .... 0.250 in.

(5) Reheating coil loss (from manufacturer's tables) .... 0.083 in.

(6) Duct resistance:

The longest run is .... = 150 ft

Two, 58 x 30 in. elbows (150% ratio) $\frac{2 \times 13 \times 58}{12}$ .... = 126 ft

Two, 30 x 15 in. elbows (150% ratio) $\frac{2 \times 13 \times 30}{12}$ .... = 65 ft

Three, 15 x 30 in. elbows (75% ratio) $\frac{3 \times 35 \times 15}{12}$ .... = 131 ft

Total equivalent run .... 472 ft

472 ft at 0.028 in. per 100 ft .... 0.132 in.

(7) Allowance for damper adjustment, 25% of 0.132 .... 0 033 in.

(8) Supply grille resistance (from manufacturer's tables) .... 0.036 in.

Total static pressure loss of system .... 0.952 in.

The fan is selected from the manufacturer's ratings to deliver 22,935 cfm at a static pressure of 0.952 in. as outlined in Chapter 29.

Example 3. If the rooms and offices of the hotel building of Example 2 are to be

TABLE 4. PIPE SIZES FOR EXAMPLE 2[a]

VOLUME OF AIR (CFM)	DIAMETER OF PIPE (INCHES)	EQUIVALENT SIZE OF RECTANGULAR DUCT (INCHES)
22,935	56	60 x 44
12,510	45	58 x 30
10,425	42	50 x 30
8,340	39	42 x 30
6,255	35	42 x 24
4,170	29.5	30 x 24
2,085	23	30 x 15

[a]Velocity through grilles (not shown) to be approximately 300 fpm

served from a manufactured unit with a capacity of 22,935 cfm against an external resistance of 0.35 in., the known resistances are calculated as:

(1) Outdoor air inlet .... 0.094 in.

(2) Allowance for damper adjustment .... 0.033 in.

(3) Supply grille resistance (from manufacturer's tables) .... 0.036 in.

Total known resistance .... 0.163 in.

Subtracting this from the total available resistance: 0.35 in. − 0 163 in. = 0.187 in. available for duct resistance.

Known length of run .... 150 ft

The duct width is then estimated for the following elbow calculations:

Four 150% ratio elbows, 4 x 13 x 3.5 ft .... 182 ft

Three 75% ratio elbows, 3 x 35 x 1.5 ft .... 158 ft

Total estimated length .... 490 ft

The duct friction per 100 ft is then 0.187 ÷ 4.90 = 0.0382 in. and the mains and branches are sized from the 0.038 in. friction line in Fig. 2.

If it is desired to size each branch for equal resistance, the total resistance back to the point of juncture is calculated and the branch is then sized in a manner similar to that outlined in Example 3.

DUCT CONSTRUCTION DETAILS

Straight sections of round duct are usually formed by rolling the sheets to the proper radius and grooving the longitudinal seam. Rectangular ducts are generally constructed by breaking the corners and grooving the longitudinal seam, although some fabricators still use the standing seam due to lack of equipment. Elbows and transformation sections are generally formed with *Pittsburgh* corner seams because this seam is easier to

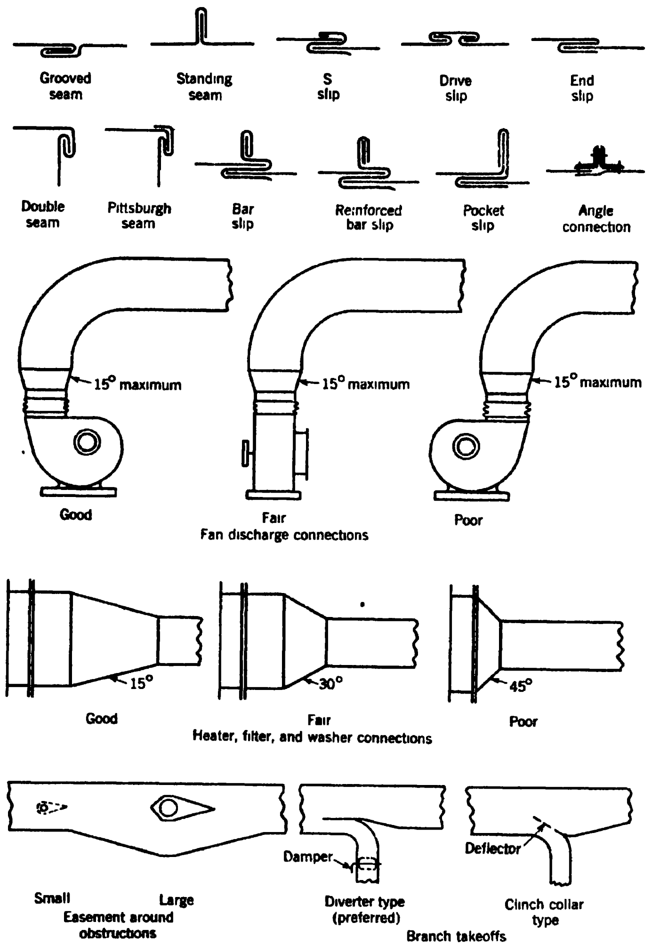


FIG. 7. SHEET METAL DUCT AND ARRANGEMENT DETAILS

lock in place than the double seam, but complicated fittings such as double compounded elbows are usually constructed with double seam corners. The construction of these various seams as well as the types of girth connections are shown in Fig. 7. The application of the various slips and connections is outlined in Table 5. The end slip may be used wherever *S* slips are recommended. Where drive slips are used the end slip may be applied on the narrow side of the duct and only the drive slips on the

maximum side. Ducts 25 to 30 in. in size should be reinforced between the joints, but not necessarily at the joint. Ducts 31 in. and up should be reinforced at the joint and between the joints; if drive slips are used the angles are usually riveted to the duct about 2 in. from the slips. It is good practice to cross-break or kink all flat surfaces to prevent vibration or buckling due to the air flow and accompanying variations in internal pressure. Round ducts are sometimes swedged 1.5 in. from the ends so that the larger end will butt against the swedge and are held in place with

Table 5. Recommended Sheet Metal Gages for Rectangular Duct Construction[a]

U. S. Std. Gage	Maximum Side, Inches	Type of Transverse Joint Connections[b]	Bracing
26	Up to 12	S, Drive, Pocket or Bar Slips, on 7 ft 10 in. centers	None
24	13 to 24	S, Drive, Pocket or Bar Slips, on 7 ft 10 in. centers	None
	25 to 30	S, Drive, 1 in. Pocket or 1 in. Bar Slips, on 7 ft 10 in. centers[c]	1 x 1 x ⅛ in. angles 4 ft from joint
22	31 to 40	Drive, 1 in. Pocket or 1 in. Bar Slips, on 7 ft 10 in. centers[c]	1 x 1 x ⅛ in. angles 4 ft from joint
	41 to 60	1½ in. Angle Connections, or 1½ in. Pocket or 1½ in. Bar Slips with 1⅜ in. x ⅛ in. bar reinforcing on 7 ft 10 in. centers[c]	1½ x 1½ x ⅛ in. angles 4 ft from joint
20	61 to 90	1½ in. Angle Connections, or 1½ in. Pocket or 1½ in. Bar Slips 3 ft 9 in. maximum centers with 1⅜ x ⅛ in. bar reinforcing	1½ x 1½ x ⅛ in diagonal angles, or 1½ x 1½ x ⅛ in. angles 2 ft from joint
18	91 and up	2 in. Angle Connections or 1½ in. Pocket or 1½ in. Bar Slips 3 ft 9 in. maximum centers with 1⅜ x ⅛ in. bar reinforcing[d]	1½ x 1½ x ⅛ in. diagonal angles, or 1½ x 1½ x ⅛ in. angles 2 ft from joint

[a]For normal pressures and velocities (see Table 3) utilized in typical ventilating and air conditioning systems. Where special rigidity or stiffness is required, ducts should be constructed of metal two gages heavier. All uninsulated ducts 18 in. and larger should be cross-broken. Cross-breaking may be omitted on uninsulated ducts if two gages of heavier metal are used.

[b]Other joint connections of equivalent mechanical strength and air tightness may be used

[c]Duct sections of 3 ft 9 in may be used with bracing angles omitted, instead of 7 ft 10 in. lengths with joints indicated.

[d]Ducts 91 in. and larger require special field study for hanging and supporting methods

sheet metal screws. Where swedges are not used it is general practice to paste the joint with asbestos paper to insure a tight joint.

The construction of elbows and changes of shape cannot be definitely outlined because of the varied conditions encountered in the field, but in general long radius elbows and gradual changes in shape tend to maintain uniform velocities accompanied by decreased turbulence, lower resistance and a minimum of noise.

Heavy canvas connections are recommended on both the inlet and outlet to all fans. The fan discharge connections shown in Fig. 7 are marked *good*, *fair*, and *poor* in the order of the amount of turbulence produced. An inspection of the heater connections shown in Fig. 7 will readily show that uniform velocity through the heater cannot be expected

in the diagram noted *poor*. When obstructions cannot be avoided, the duct area should never be decreased more than 10 per cent and then a *streamlined* collar should be used. Larger obstructions require an increase in the duct size in order to maintain as nearly uniform velocity as possible. Branch take-offs should always be arranged to cut or slice into the air stream in order to reduce as far as possible the losses in velocity head.

The recommended gages for sheet metal duct construction are given in

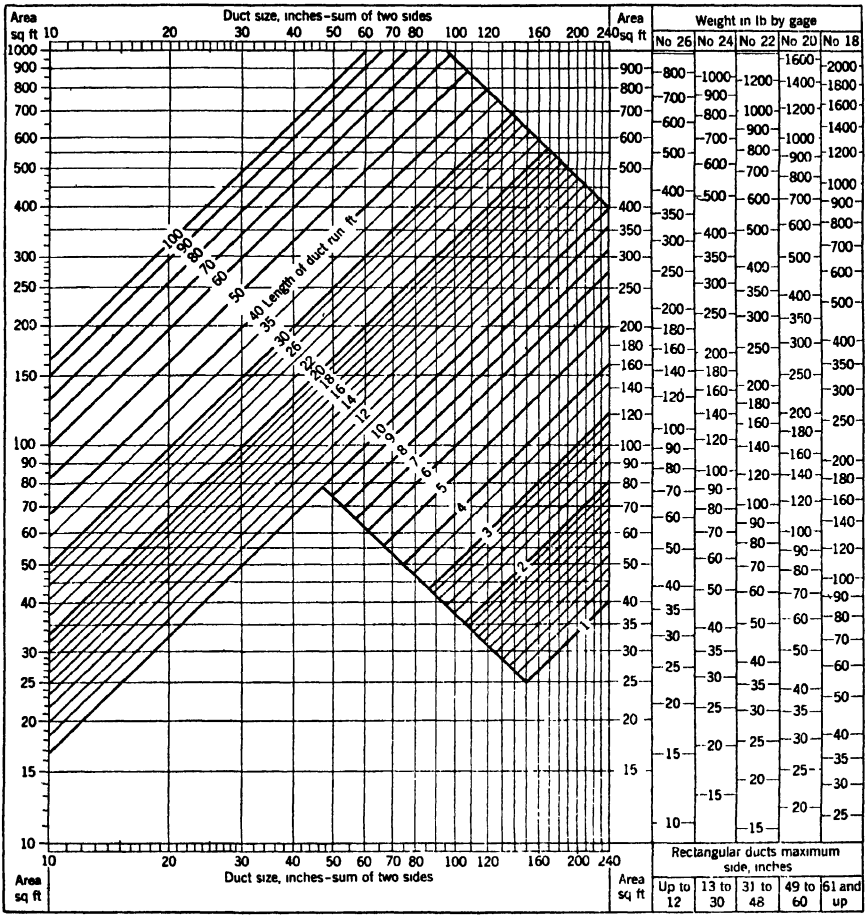


FIG. 8. AREA AND WEIGHT OF RECTANGULAR SHEET METAL DUCTS

Table 5. Weights of sheet metal per square foot of surface for different gages are given in Table 6. The weights of various gages and the areas for any length of run of rectangular sheet metal ducts may also be determined from Fig. 8. The bottom scale represents the sum of the two sides of the duct and the oblique lines give the length of run in feet. Proceeding horizontally to the right from the intersection of vertical and oblique lines on the chart, the area of the duct may be determined in the first vertical scale.' The scales to the right give the weights of the duct

Table 6. Weights of Sheet Metal Used for Duct Construction

U. S. Std. Gage	Black Sheets				Galvanized Sheets[a]			
	Approximate Thickness, In.		Weight Per Square Foot		Approximate Thickness, In.		Weight Per Square Foot	
	Steel	Iron	Ounces	Pounds	Steel	Iron	Ounces	Pounds
30	0.0123	0.0125	8	0.500	0.0163	0.0165	10.5	0.656
28	0.0153	0.0156	10	0.625	0.0193	0.0196	12.5	0.781
26	0.0184	0.0188	12	0.750	0.0224	0.0228	14.5	0.906
24	0.0245	0.0250	16	1.000	0.0285	0.0290	18.5	1.156
22	0.0306	0.0313	20	1.250	0.0346	0 0353	22.5	1.406
20	0.0368	0.0375	24	1.500	0.0408	0.0415	26.5	1.656
18	0.0490	0.0500	32	2.000	0.0530	0.0540	34.5	2.156
16	0.0613	0.0625	40	2.500	0.0653	0.0665	42.5	2.656
14	0.0766	0.0781	50	3.125	0.0806	0.0821	52.5	3.281
12	0.1072	0.1094	70	4.375	0.1112	0.1134	72.5	4.531
11	0.1225	0.1250	80	5.000	0.1265	0.1290	82.5	5.156
10	0.1379	0.1406	90	5.625	0.1419	0.1446	92.5	5.781

[a]Galvanized sheets are gaged before galvanizing and are therefore approximately 0.004 in. thicker.

run for different gages of metal. In calculating the weights of duct, it is considered good practice to allow 20 per cent additional for weights of joints and bracings. Various weights and thicknesses of standard copper sheets will be found in Table 7.

HEAT LOSSES FROM DUCTS

The thermal transmission coefficient U for an uninsulated metal duct can be obtained from the equation:

$$U = \frac{1}{\frac{1}{f_1} + \frac{1}{f_0}} \qquad (8)$$

In the case of non-metallic ducts the formula in Equation 8 will become:

$$U = \frac{1}{\frac{1}{f_1} + \frac{x}{k} + \frac{1}{f_0}} \qquad (9)$$

where U, f_1, f_0, x and k are as given on pages 81 and 82, Chapter 4.

Table 7. Weights and Thicknesses of Standard Copper Sheets[a]
Rolled to Weight

Weight per Square Foot		Thickness, Inches		Nearest Gage No.		
Ounces	Pounds	Decimal Equivalent	Nearest Fraction	B. & S.	Stubs	U. S. Std.
10	0.625	0.0135	1/64	27	29	29
12	0.750	0.0162	1/64	26	27	28
14	0.875	0.0189	1/64	25	26	26
16	1.000	0.0216	1/32	23	24	25
18	1.125	0.0243	1/32	22	23	24
20	1.250	0.0270	1/32	21	22	23
24	1.500	0.0324	1/32	20	21	22
28	1.750	0.0378	1/32	19	20	20
32	2.000	0.0432	3/64	17	19	19
36	2.250	0.0486	3/64	16	18	18
40	2.500	0.0540	3/64	15	17	17
44	2.750	0.0594	1/16	15	17	17
48	3.000	0.0648	1/16	14	16	16
56	3.500	0.0756	5/64	13	15	14
64	4.000	0.0864	5/64	11	14	13

[a]Variations from these weights must be expected in practice.

Where x is small and k is large, however, this factor $\frac{x}{k}$ is of little importance and may be neglected.

Film conductance f_i for air flowing in ducts apparently depends only on the velocity of the air and the diameter of the duct. A fairly reliable inside coefficient can be calculated from Schultz's modified equation:

$$f_i = \frac{0.32\, V_o^{0.8}}{D^{0.25}} \tag{10}$$

where

V_o = velocity of air in duct, feet per second.
D = diameter of duct, feet.

Film conductance f_o depends on a number of variables including temperature, diameter, and emissivity of the outer surface and can readily be calculated from data in Chapter 3. From this explanation, it is seen that it is unwise to recommend a given value of U for all uninsulated metal ducts.

The heat loss from a given length of duct can be expressed by:

$$Q = UPL\left[\left(\frac{t_1 + t_2}{2}\right) - t_3\right] \tag{11}$$

The heat given up by the air in the duct is:

$$Q = 0.24\, M\,(t_1 - t_2) = 14.4\, A V \rho\,(t_1 - t_2) \tag{12}$$

Equating 11 and 12 enables the determination of the temperature drop in the duct:

$$\frac{t_1 + t_2 - 2t_3}{t_1 - t_2} = \frac{28.8\, A V\rho}{UPL}$$

Let $y = \frac{28.8\, A V\rho}{UPL}$ for rectangular ducts, $= \frac{7.2\, D V\rho}{UL}$ for round ducts, solving for t_1 and t_2:

$$t_1 = \frac{t_2\,(y + 1) - 2t_3}{(y - 1)} \tag{13}$$

$$t_2 = \frac{t_1\,(y - 1) + 2t_3}{(y + 1)} \tag{14}$$

For low velocities and long ducts of small cross-section, a somewhat more accurate formula may be used as follows:

$$t_2 = \frac{t_1 - t_3}{e^{\left(\frac{UPL}{14.4\, A\rho V}\right)}} + t_3 \tag{15}$$

In these equations

Q = heat loss through duct walls, Btu per hour.
U = thermal transmission coefficient, Btu per square foot per hour per degree Fahrenheit.
P = perimeter of duct, feet.
L = length of duct, feet.
t_1 = temperature of air entering duct, degree Fahrenheit.
t_2 = temperature of air leaving duct, degree Fahrenheit.
t_3 = temperature of air surrounding duct, degree Fahrenheit.
M = weight of air per hour, through the duct, pounds.

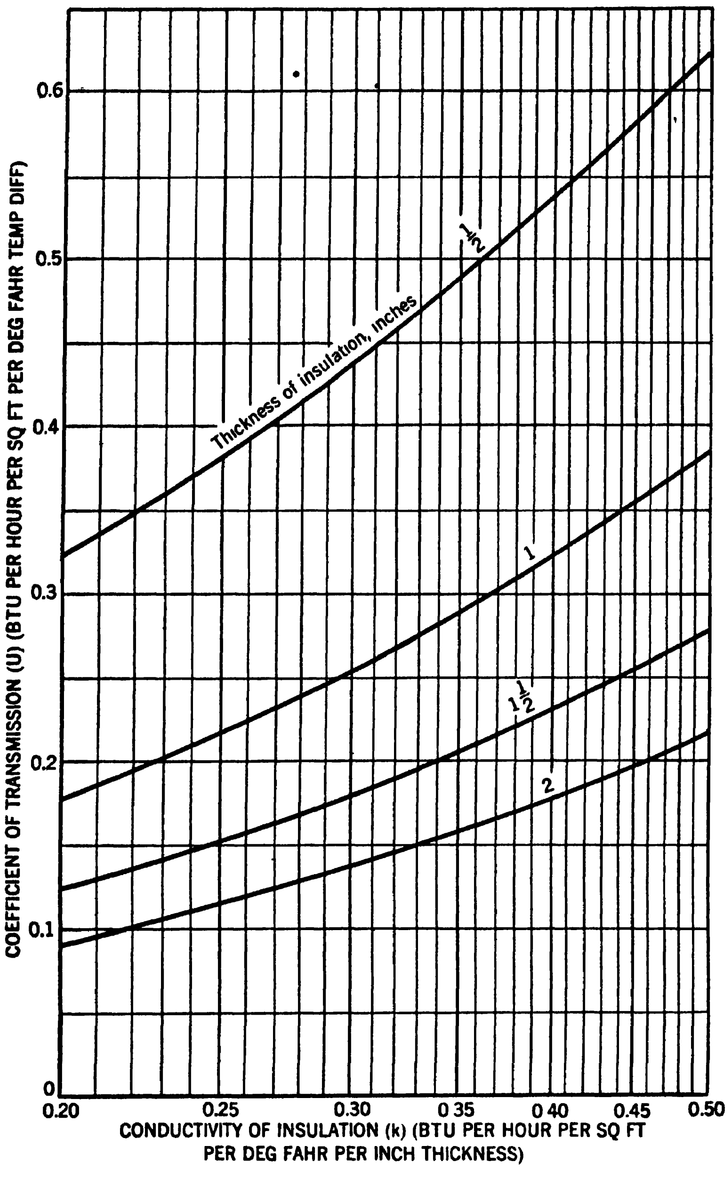


Fig. 9. Heat Loss Coefficients for Insulated Ducts[a]

[a]For round ducts less than 30 in. diameter, increase heat transmission values by the percentages shown.

Thickness of Insulation (Inches)	½	1	1½	2
21 to 30 in. Duct Diameter	1%	2%	3%	4%
12 to 21 in. Duct Diameter	3%	5%	7%	9%

A = cross-sectional area of duct, feet.
D = diameter of round duct, feet.
V = velocity of air in the duct, feet per minute, at specified temperature.
ρ = density of air, pounds per cubic foot, at the specified temperature at which V is measured.
e = Naperian base of logarithms = 2.718.

In using Equations 13, 14 and 15, one of the duct air temperatures will be unknown and will be solved for by substitution of the other known or assumed values.

Heat loss coefficients for insulated ducts with various conductivities are given in Fig. 9. The conductivities of various materials, which are based on mean temperatures, ranging from about 70 to 90 F, will be found in Table 2 of Chapter 4. For cases where the mean temperature is other than that on which the test was conducted, a correction should be made. However, in most cases the effect of this factor will be small and may be neglected.

Example 4. Determine the entering air temperature and heat loss for a duct 24 × 36 in. cross-section and 70 ft in length, insulated with ½ in. of a material having a conductivity of 0.35 Btu at 86 F mean temperature, carrying air at a velocity of 1200 fpm, measured at 70 F, to deliver air at 120 F with air surrounding the duct at 40 F.

Solution. Referring to Fig. 9, the over-all heat transmission coefficient is found to be 0.49 Btu. From Table 6, Chapter 1 the density of air at 70 F and 29.92 in. Hg is found to be 0.0749 lb per cubic foot. Substituting these and the other given values in Equation 13:

$$y = \frac{28.8 \times 6 \times 1200 \times 0.0749}{0.49 \times 10 \times 70} = 45.3$$

$$t_1 = \frac{120\ (45.3 + 1) - 80}{45.3 - 1} = 123.7$$

Substituting in Equation 11:

$$Q = 0.49 \times 10 \times 70 \left[\left(\frac{123.7 + 120}{2} \right) - 40 \right]$$

$$Q = 28{,}100 \text{ Btu per hour.}$$

For special considerations which apply to insulation of ducts in marine installations see Chapter 38.

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CHAPTER 32

Sound Control

Unit of Noise Measurement, Apparatus for Measuring Noise, General Problem, Kinds of Noise, Noise Transmitted Through Ducts, Design Room Noise Level, Noise Generated by Fan, Natural Attenuation of Duct System, Duct Sound Absorbers, Air Supply Noises, Grille Selection, Cross Transmission Between Rooms, Controlling Vibration from Machine Mountings

IN ventilating and air conditioning a building or a room, the effect of the mechanical system employed must be considered on the acoustics of the space conditioned. It is important to consider also that the use of air conditioning often permits keeping the windows closed, thus giving relief from certain external noises, but at the same time increasing the necessity of providing adequate sound control.

It is assumed that in a given space the architect and acoustical engineer have produced a room or rooms which are satisfactory for speech, music, or other uses. The ventilating engineer's sole function is to ventilate and air condition these rooms properly so that they will be physically comfortable without adding any acoustical hazards.

UNIT OF NOISE MEASUREMENT

According to an international standard, two terms are used for noise measurement. The *decibel* (db) is the physical unit for expressing intensity or pressure levels. The *phon* is the unit of loudness level. The loudness level, in phons, of any sound is by definition equal to the intensity level in decibels of a thousand cycle tone which sounds equally loud. The decibel is defined by the relation $N = 10 \log_{10} \frac{I_1}{I_0}$, where N is the number of decibels by which the intensity flux I_1 exceeds the intensity flux I_0. The intensity flux is the measure of the intensity of a sound wave and is defined in terms of watts per square centimeter passing through a unit area of wave front in a freely traveling plane wave. It is usually more convenient to select an arbitrary reference intensity for I_0 and express all other intensities in terms of decibels above that level. For this purpose a reference intensity of 10^{-16} watts per square centimeter has been selected. This intensity is slightly less than the threshold of audibility for the average ear at a frequency of 1,000 cycles per second. This reference level also corresponds to a pressure of 0.0002 dynes per square centimeter for sound in air at usual room temperatures.

A stated sound level in decibels, unless otherwise defined, will thus be related to a threshold of 10^{-16} watts. For example, a level of 60 db above this reference threshold is 10^{-10} watts. In a similar manner, when sound measurements are given in actual intensity or energy units, they can be converted to decibels by this relation.

Since the decibel is based on a ratio, it can only be employed when related to a reference threshold level as given. Noise levels, which vary with frequency as well as intensity, must not only be related to this reference threshold level, but also to a reference frequency, which is taken as 1000 cycles. These terms and procedures may be found in Standards[1] published by the *American Standards Association.*

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[1]American Standards for Noise Measurement, Z24.2-1942, *American Standards Association*

APPARATUS FOR MEASURING NOISE

Since the relative loudness to the ear, rather than the actual physical intensity, is the quantity in which engineers are usually interested, it has been found necessary to allow for the varying response of the ear at different frequencies in designing noise measuring equipment. The most satisfactory method of measuring noise is by means of a sound level meter which usually consists of a microphone, a high gain audio-amplifier, and a rectifying milliammeter which will read directly in decibels. This meter is calibrated to give readings above the standard reference level and usually contains a weighting network to make it less sensitive at those frequencies where the ear is less sensitive. For complete specifications relative to the approved type of sound level meters refer to the information[2] published by the *American Standards Association.*

GENERAL PROBLEM OF SOUND CONTROL

As previously stated, the problem confronting the air conditioning engineer is that of designing a system which will operate without increasing the noise level in the conditioned space. To be sure that this is accomplished, it is necessary:

1. To determine the noise level existing without the equipment
2. To ascertain the noise level which would exist if the equipment were installed without sound control.
3. To provide as a part of the installation sufficient sound control appliances to reduce the noise level substantially to that found in Item 1.

To accomplish this the engineer should have information of three kinds:

1. A knowledge of the noise levels currently considered acceptable in various rooms in order that he may have a basis on which to proceed.
2. A knowledge of the nature and intensity of the noise created by the various parts of the equipment.
3. A knowledge of how, when necessary, to vary and control the noise level between the equipment and the conditioned space

In addition, the engineer should have information to analyze noises which may be transmitted by the duct system from one conditioned space to another, or from an outside space to the conditioned space.

Information concerning the noise levels created by ventilating and air conditioning equipment such as fans, motors, air washers, and similar items is not yet on a basis which permits tabular presentation, although certain manufacturers are prepared to offer such data and do state the noise producing properties of their products. A sound test code for fans has been developed by the *National Association of Fan Manufacturers* which uses the flat response network of the sound level meter. However, when determining the noise generated by an air distribution system, it is customary to use the noise level of the fan as determined by the weighting network most nearly approaching the noise level of the space. In most cases this will be the noise level of the fan as determined on the 40 decibel weighting network. Refer to Table 1 for typical noise levels.

KINDS OF NOISE

To solve a sound problem of this type it is desirable to consider separately the several means by which noise reaches the room. This avoids to some extent the necessity of knowing the noise level at the source, and

[2]American Standards for Sound Level Meters for Measurement of Noise and Other Sounds, Z24 3-1944, *American Standards Association.*

instead, places the emphasis on ascertaining the level at the point where the sound enters the room.

The noise introduced into a room or building by ventilating or air conditioning equipment may be divided into two general kinds depending on how it reaches the room with various sub-divisions:

1. Noise transmitted through the ducts.
   a. From equipment such as sprays, fans, etc.
   b. From outside, and transmitted through duct walls into air stream.
   c. From air current, including eddying noises.
   d. Cross talk and cross noises between rooms connected by the same duct system.
   e. Noise produced by the grilles.
2. Noise transmitted through the building construction.
   a. From machine mountings as vibration.
   b. From equipment through room wall surfaces.

The next step in the solution of this problem is to present data and discuss methods whereby solutions to the noise problem can be obtained when the allowable room noise level and the path through which the noise reaches the room are known.

NOISE TRANSMITTED THROUGH DUCTS

Operation of an air distribution system results in the generation of noise which may be transmitted through the ducts to the ventilated or conditioned room. The transmission of this noise may be controlled by the proper application of sound absorptive material within the ducts. The application of the absorptive material is a problem in balancing the room noise level requirements against the intensity of the noise generated. The four steps in the problem are:

1. Determination of acceptable room noise level resulting from the operation of the equipment.

2. Determination of noise level generated by the equipment.

The difference between items 1 and 2 in decibels is the over-all noise reduction required between the equipment and the room. In the discussion which follows reduction of noise will be referred to as *attenuation* of noise.

3. Determination of the natural attenuation of the duct system.

4. Selection of the proper sound treatment for the duct system.

The difference in decibels between the over-all attenuation required and the natural attenuation (3) is the additional sound attenuation to be provided by absorptive materials installed in the duct system.

DESIGN ROOM NOISE LEVEL

Measurements of noise levels have been observed by several investigators in various rooms and locations and are listed in Table 1. The values given were determined with the air conditioning or ventilation equipment not in operation, and with all windows and doors closed simulating the conditions of an actual installation.

This is an important consideration, for in offices or stores adjacent to busy thoroughfares the difference between the typical noise level in the space with the windows and doors open and closed may be as high as 10 db. Minimum, representative, and maximum levels are given for each type of space. The values are intended to give the variation with respect to location and not to time, and may be roughly classified by the following:

Minimum loudness refers to: Spaces of expensive construction, typified

by double windows, carpeted floors, heavy upholstered furniture, or acoustically treated walls and ceilings.

Representative loudness refers to: Spaces of average construction and furnishings which are exposed to external noises typical of the locality in which the space is usually found.

Maximum loudness refers to: (1) Any space of inexpensive construction, and bare furnishings where noise is not an important factor. (2) Spaces in close proximity to very intense street traffic or to intense factory noise.

TABLE 1. TYPICAL NOISE LEVELS[a]

Rooms	Noise Level in Decibels to be Anticipated		
	Min.	Representative	Max.
Sound Film Studios	10	14	20
Radio Broadcasting Studios	10	14	20
Planetarium	15	20	25
Residence, Apartments, etc.	33	40	48
Theaters, Legitimate	25	30	35
Theaters, Motion Picture	30	35	40
Auditoriums, Concert Halls, etc.	25	30	40
Churches	25	30	35
Executive Offices, Acoustically Treated Private Offices	30	38	45
Private Offices, Acoustically Untreated	35	43	50
General Offices	50	60	70
Hospitals	25	40	55
Class Rooms	30	35	45
Libraries, Museums, Art Galleries	30	40	45
Public Buildings, Post Offices, etc.	45	55	60
Court Rooms	30	35	45
Small Stores	40	50	60
Upper Floors Department Stores	40	50	55
Stores, General, Including Main Floor Dept. Stores	50	60	70
Hotel Dining Rooms	40	50	60
Restaurants and Cafeterias	50	60	70
Banking Rooms	50	55	60
Factories	65	77	90
Office Machine Rooms	60	70	80
Vehicles			
Railroad Coach	60[b]	70	80
Pullman Car	55[b]	65	75
Automobile	50	65	80
Vehicular Tunnel	75	85	95
Airplane	75	80	90

[a]These values are only tentative and approximate and, in general, are for acoustically treated spaces. Recent measurements by D F. Seacord, Bell Telephone Laboratories (*Journal Acoustical Society of America*, Vol 12, pp 183-187, 1940) give average values and standard deviations of room noise in residences, offices, stores, factories, etc, in large American cities.

[b]For train standing in station a level of about 45 db is the maximum which can ordinarily be tolerated.

In general, if the noise level in the space resulting only from the operation of the air conditioning equipment is equivalent to or less than the typical level given in Table 1, the installation will prove satisfactory. If the typical level and the equipment level are equal and heard together the resultant level will be 3 db higher than either of them alone.

In some cases it is desirable to keep the equipment noise level in the ventilated or conditioned room at such a value that it actually will not increase the noise level in the room to any measureable degree. This can be accomplished if the equipment noise at the room can be kept 10 db below the noise levels shown in Table 1.

TABLE 2. ATTENUATION IN STRAIGHT SHEET METAL DUCT RUNS

DUCT	SIZE, IN.	ATTENUATION PER FT, db
Small	6 x 6	0.10
Medium	24 x 24	0.05
Large	72 x 72	0.01

NOISE GENERATED BY FANS

Noise generated by fan wheels may be divided into two classifications, rotational noise and vortex noise. In ventilation and air conditioning work, where the maximum ratio between the fan tip speed and the velocity of sound is not greater than 0.12, vortex noise is by far the most important. The rotational noise may be described as that due to the thrust and torque applied to the air. Vortex noise is that due to the shedding of vortices from the blade and is dependent on the angle of attack, velocity, air turbulence, and blade shape. Vortex noise is due to pressure variations on the blade as a result of variations of air circulation. Given the noise level at the outlet or inlet of one type of fan construction under specific conditions of size, tip speed, and total pressure, noise levels at other values of tip speed, total pressure, and size may be approximated by the relationships:

1. For constant size and point of rating, the noise level of a fan will increase with increasing speed.

$$db \text{ (change)} = 55 \log_{10} \left(\frac{RPM_2}{RPM_1}\right) \quad (1)$$

2. For constant pressure and tip speed, the noise level of a given type of fan will increase with increasing fan size.

$$db \text{ (change)} = 20 \log_{10} \left(\frac{\text{Size}_2}{\text{Size}_1}\right) \quad (2)$$

Fan size refers to wheel diameter, housing height or some dimension that is directly proportional to lineal units. Fan sizes based on arbitrary systems or systems of preferred numbers have no significance.

The noise of a given fan is not constant at constant speed if the air delivery changes due to change of resistance. In general, a backward curved blade fan is lowest in noise at or near the point of maximum efficiency; a forward curved blade fan at or between the point of maximum efficiency and shut-off; an axial flow fan at or between the point of maximum efficiency and free delivery. The noise level of a double width fan

TABLE 3. ATTENUATION OF ELBOWS[a]

ELBOW	SIZE, IN[b]	ATTENUATION PER ELBOW, db
Very small	2 wide	3
Small	3 to 15	2
Medium	15 to 36	1.5
Large	36 plus	1

[a]The attenuation in vaned elbows should be considered the same as in elbows having the same dimensions as the radius of curvature of the vanes. If the vanes are lined for the purpose of damping any vibrations in them, one third may be added to the attenuation values listed.

[b]These attenuation values are based on elbows having a center line radius 1.5 to 2 times the diameter or width of the duct. The attenuation will be greater if the ratio is less than 1.5 and less when the ratio is greater than 2.

may be taken as 3 db higher than a similar single width fan operating under the same conditions of speed and pressure.

NATURAL ATTENUATION OF DUCT SYSTEM

Straight Sheet Metal Ducts. The attenuation of sound in straight sheet metal ducts is a function of the length, shape, and size of the duct[8]. Attenuation values are given in Table 2. In general, this attenuation is so negligible except for long runs that it may be disregarded for all practical purposes.

Elbows and Transformations. Due to reflective interference, attenuation will take place at elbows and transformations. The magnitude of the attenuation will depend on the size and abruptness of the elbow or transformation as shown in Table 3.

When the area of a duct increases abruptly, an attenuation of noise level takes place in the duct. In duct design practice the total area of the branch ducts is greater than the supply duct. Similarly with outlets,

TABLE 4. ATTENUATION AT DUCT BRANCHES OR OUTLETS

Ratio $\frac{\text{Branch Duct} + \text{Outlet Area}}{\text{Supply Duct Area}}$ or $\frac{\text{Sum of Branch Areas}}{\text{Supply Duct Area}}$	Attenuation per Transformation, db
1.00	0.0
1.20	0.8
1.35	1.3
1.50	1.8
1.75	2.5
2.00	3.0

the area of the outlet plus the area of the duct after the outlet is greater than the duct area before the outlet. Therefore in an outlet run, attenuation occurs in the duct as it passes each outlet. Table 4 gives the *db* reduction for various ratios of total branch duct and outlet area to supply duct area.

Grilles to Room. The large abrupt change in area between the grilles and the surfaces within a room results in an appreciable noise attenuation. This attenuation is a function of the total grille area (supply and return) and the total sound absorption of the room in *sabines*. (The sound absorption of a room in *sabines* is the summation of the products of each surface of the room measured in square feet multiplied by its corresponding absorption coefficient). The attenuation is given in Equation 3 as:

$$db\left(\begin{matrix}\text{Attenuation between}\\ \text{grilles and room}\end{matrix}\right) = 10\log_{10}\frac{\text{Total Room Absorption in Sabines}}{\text{Total Grille Area}} \quad (3)$$

Values in Table 5 approximate the attenuation for various rates of air change, and general types of room surfaces.

DUCT SOUND ABSORBERS

The difference between the required sound attenuation and the natural attenuation is that which must be supplied by the proper sound treatment of the ducts.

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[8]A.S.H V.E. RESEARCH REPORT No. 1205—Determining Sound Attenuation in Air Conditioning Systems, by D A Wilbur and R F. Simons (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 267).

Selection of the Absorptive Material

When a sound wave impinges on the surface of a porous material, a vibrating motion is set up within the small pores of the material by the alternating sound waves. As the ratio of the cross sectional area of the pores to their interior surface is small, the resistance to the movement of air in the pores is large. This viscous resistance within the pores of the material converts a portion of the sound energy into heat. The decimal fraction representing the absorbed portion of the incident sound wave is called the absorption coefficient. Considerable absorption may also result, particularly in the low frequency range, from the flexural vibrations of the duct. In the selection and application of the absorptive material, several points should be considered.

TABLE 5. APPROXIMATE ATTENUATION BETWEEN GRILLES AND ROOM

OUTLET VELOCITY FPM	AIR CHANGE MIN.	LIVE ROOM[b] $a^{a} = 0.05$ db	MEDIUM ROOM[c] $a = 0.15$ db	DEAD ROOM[d] $a = 0.25$ db
500	5	11	16	18
	10	14	19	21
	15	16	21	23
	20	17	22	24
750	5	13	18	20
	10	16	21	23
	15	18	23	25
	20	19	24	26
1000	5	14	19	21
	10	17	22	24
	15	19	24	26
	20	20	25	28
1250	5	15	20	22
	10	18	23	25
	15	20	25	27
	20	21	26	28

[a]Average absorption coefficient for the room.

[b]Live room average absorption coefficient 0.05 Bare wood or concrete floor—hard plaster walls and ceiling—minimum of furniture.

[c]Medium room average absorption coefficient 0.15. Carpeted floor, upholstered furniture, hard plaster walls and ceiling or bare room with acoustically treated ceiling.

[d]Dead room average absorption coefficient 0.25. Heavy carpeted floor. Walls and ceiling acoustically treated. Upholstered furniture.

1. For the absorption of the low frequencies the material should be at least 1 to 2 in. thick. Thin materials, particularly when mounted on hard solid surfaces, will absorb the high frequencies and reflect the low.

2. In order to take advantage of low frequency noise absorption by panel vibration, it is advisable to fasten the absorptive sheets to stripping so that the panels themselves may vibrate. However, the exact resonance characteristics of the panels and thus their absorption are so unpredictable that panel resonance cannot be relied upon for a specific value of attenuation.

Requirements for a good sound absorption material are: (1) high absorption at low frequencies[4], (2) adequate strength to avoid breakage, (3) fire resistance and compliance with national and local code requirements, (4) low moisture absorption, (5) freedom from attack by bacteria and algae, (6) low surface coefficient of friction, (7) particles should not

[4]For coefficients of commercial sound absorbent materials see Bulletin *Acoustical Materials Association*, 919 No. Michigan Ave., Chicago, Ill.

fray off at the higher design velocities, and (8) freedom from odor when either dry or wet.

With every application the use of sound absorptive material should be considered in the dual function of insulation and sound absorption. It has been shown theoretically[5] that the reduction, in decibels per linear foot, of sound transmitted through a duct lined with sound absorbing material is related in a rather complicated manner to the size and shape of the duct, to the frequency of the sound, and to the sound absorbing characteristics of the lining. Experimental evidence likewise indicates that there is no simple formula involving the variables which will apply accurately to all cases. However, it may be stated generally that the attenuation is directly proportional to the length of lined duct. It decreases as the cross-sectional area increases, and increases as the aspect ratio is increased.

The noise reduction varies to a considerable extent with the frequency of the sound. In calculating noise reduction, consideration should be

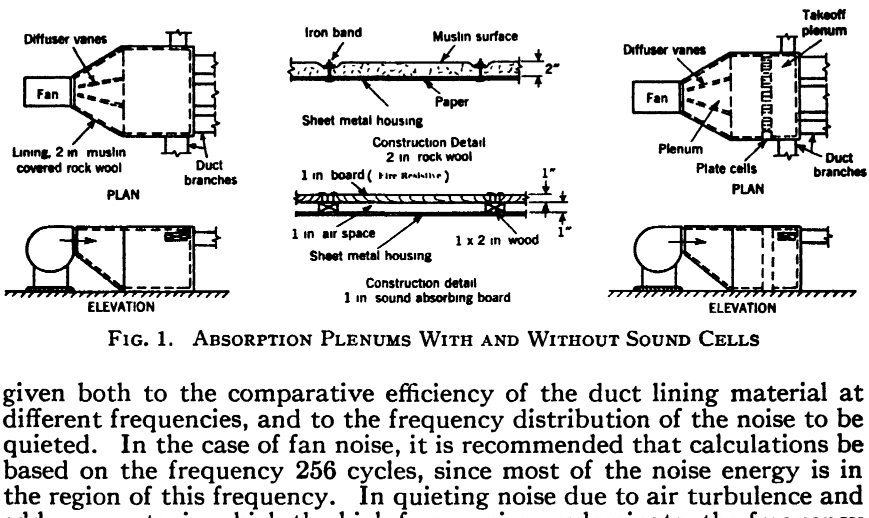

FIG. 1. ABSORPTION PLENUMS WITH AND WITHOUT SOUND CELLS

given both to the comparative efficiency of the duct lining material at different frequencies, and to the frequency distribution of the noise to be quieted. In the case of fan noise, it is recommended that calculations be based on the frequency 256 cycles, since most of the noise energy is in the region of this frequency. In quieting noise due to air turbulence and eddy currents, in which the high frequencies predominate, the frequency 1024 cycles should be used.

Plenum Absorption

In systems, where individual ducts are directed to a number of rooms and sound treatment is required in every duct, a sound absorption plenum on the fan discharge as shown in Fig. 1, will often prove the most economical arrangement. The absorption in the plenum may be approximated by Equation 4.

$$db \text{ (Attenuation)} = 10 \log_{10} \frac{\text{Plenum Absorption in Sabines}}{\text{Area Fan Discharge}} \tag{4}$$

The area of the plenum should be at least ten times as great as the fan discharge area. The plenum should be lined with 2 in. of muslin covered rock wool blanket or 1 in. sound absorbing board preferably nailed to wood strips on the inside of the plenum. With such a lining the plenum

[5]Sound Propagation in Ducts Lined with Absorbing Materials, by L. J. Sivian (*Journal Acoustical Society of America*, Vol. 9, 1937-38, pp. 135-140).

TABLE 6. END REFLECTION OF PLATE ABSORBERS

PERCENTAGE FREE AREA OF ABSORBER	ATTENUATION, db
50	1
40	2
30	4
25	5
20	6

is particularly effective in reducing low frequency fan noise. The absorption of the plenum in sabines is the sum of the products of each interior area of the plenum measured in square feet multiplied by its corresponding absorption coefficient.

Plate Cells

One of the most economical methods of applying sound absorbent material from the standpoint of both labor and material is the plate cell. The plate cell consists of ½ or 1 in. sound absorbent board, spaced on 2, 3 or 4 in. centers. The attenuation due to the plate cell may be divided into two parts. There is reflection at each end due to the change in area and the absorption at the ends. Values for this attenuation are given in Table 6 which depend on the spacing. There is also attenuation due to absorption of sound within the passages of the cell, which depends on the length and the spacing. The attenuation within the cell for 1 in. board neglecting the end effect is given approximately by Equation 5.

$$R = \frac{10La^{1.4}}{S} \tag{5}$$

where

R = attenuation, decibels.
L = linear length of duct, feet.
S = spacing in inches between plates up to 3 in.
a = absorption coefficient for the full thickness of the cell material. For typical value of a see Table 7.

An important objection to the plate cell is the increase in duct cross-sectional area required. Often on the fan discharge, particularly with unitary equipment, where a number of branch ducts take off, the plate cell may be installed with little or no difficulty.

Outlet Sound Absorbers

Outlet sound absorbers are rectangular or plate cells installed directly behind an outlet or they may be the lining of a pan or plaque outlet. They are particularly effective in the elimination of high frequency whistles which are generated by air flow in the ducts. They are also employed in large systems with long runs where only a few outlets near

TABLE 7. ATTENUATION FORMULAE FOR 1 IN. THICK TYPICAL DUCT LINING BOARD

FREQUENCY	ABSORPTION COEFFICIENT	ATTENUATION REDUCTION, db
256	0.37	3.0 L P/A
512	0.69	7.5 L P/A
1024	0.78	9.5 L P/A
2048	0.78	9.5 L P/A

the fan require treatment. Frequently outlet cells are the only means of correcting existing noisy installations, as the duct sections directly behind the outlets may be the only sections accessible for treatment. (See Fig. 2.)

Duct Lining or Rectangular Cells

One series of experiments[6] made on a commonly used type of duct lining material (1 in. rock wool sheet) has shown that, subject to certain restrictions, the attenuation of single-frequency sounds may be expressed by the approximate Equation 6. This equation is accurate within plus or minus 10 per cent for duct sizes ranging from 9 x 9 in. to 18 x 18 in., for cross-sectional dimension ratios of 1:1 to 2:1, for frequencies between 256 and 2048 cycles, and for absorption coefficients between 0.20 and 0.80.

$$R = 12.6\, L \frac{P}{A}\, a^{1.4} \tag{6}$$

where

R = attenuation, decibels.
L = length of lined duct, feet.
P = perimeter of duct, inches.
A = cross-sectional area of duct, square inches.
a = absorption coefficient of lining.

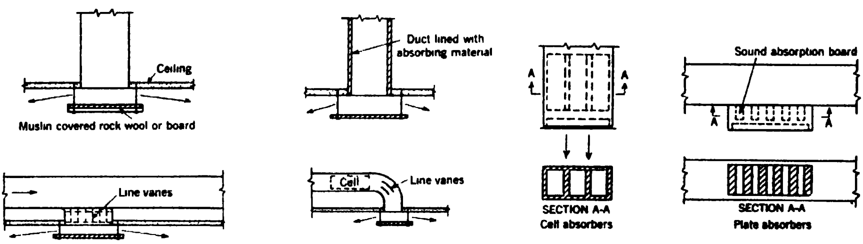

FIG. 2. OUTLET CELLS FOR FAN OUTLETS OR GRILLES

In Table 7, the absorption coefficients at different frequencies of a material of the previously mentioned type are listed, together with the corresponding values for Equation 6.

Results of other experiments indicate, however, that Equation 6 may be in error when applied to other types of duct lining material and to duct sizes and shapes outside of the range specified. An empirically derived chart[7] representing the *average* experimental data on a number of different types of materials including the rock wool sheet mentioned as applicable to Equation 6 is shown in Fig. 3. Since individual materials vary, the curves in Fig. 3 are given only as representing the best available averages for duct sizes of square cross-sections from 6 x 6 in. to 48 x 48 in. As an illustration, the dotted lines in the chart show values calculated from Equation 6 which indicate that the slope for this particular material is somewhat different than from the average curves. The curves in Fig. 3, as well as Equation 6, show that the attenuation in decibels is directly proportional to the length of duct lined, and that the larger the duct the

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[6]The Absorption of Noise in Ventilating Ducts, by Hale J. Sabine (*Journal Acoustical Society of America*, Vol. 12, p. 53, 1940).

[7]The Prediction of Noise Levels from Mechanical Equipment, by J. S Parkinson (*Heating and Ventilating*, March, 1939, pp. 23-26). Methods of Rating the Noise from Air Conditioning Equipment, by J. S Parkinson (A S.H.V E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, July, 1940, p. 447)

greater will be the length which must be lined in order to obtain a given noise reduction.

If the length of duct from the main duct to a grille is shorter than the length of lining indicated by the calculations, this duct may be subdivided into smaller ducts, as shown in Fig. 4. The increase in noise reduction thus obtained may be calculated from Equation 7, provided the splitters are installed parallel to the long side of the duct:

$$R_s = R_0 \frac{a + bn}{a + b} \tag{7}$$

where

R_s = reduction with splitters, decibels.
R_0 = reduction in same length of duct, without splitters, decibels.
a = dimension of short side of duct, inches or feet.
b = dimension of long side of duct, inches or feet.
n = number of channels formed by splitters.

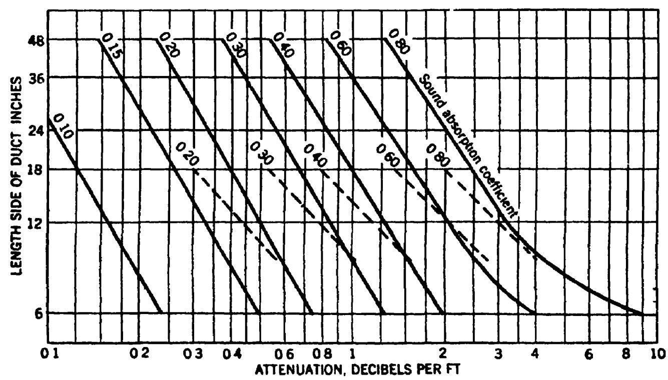

FIG. 3. SOUND ATTENUATION FOR VARIOUS ABSORBING DUCT LINERS

Example 1. An air conditioning installation is to be installed in a small theater. Determine the necessary sound treatment for the air distribution system to provide a satisfactory noise level in the theater utilizing these conditions:

Fan tip speed 4000 fpm, total pressure 1.25 in.	77 db
Acceptable room noise level (Table 1)	40 db
Required attenuation	37 db

Solution: Natural attenuation of supply duct.

Sheet metal duct 50 ft long 48 in. x 36 in. (Table 2) 50 x 0.01	0.5 db
Elbows, two size 48 in. x 36 in. (Table 3) 2 x 1	2.0 db
Attenuation grilles to theater air change 10 min (Table 5) outlet velocity 1000 fpm	22.0 db
Total natural attenuation	24.5 db

Difference between required and natural attenuation, 37 minus 24.5, is 12.5 db. This attenuation must be supplied by sound treatment in the duct, either in the form of duct lining or plate cells.

A similar analysis of the return duct system, shows that 15 db attenuation are to be furnished by absorptive material. An inspection of the installation shows that the lining of the plenum on the suction side of the fan would prove the most economical, where it would secure the dual function of heat insulation and sound absorption.

Example 2. A 10 x 20 in. duct is connected to a private office space in a quiet location. Determine the length of lining necessary to attenuate average fan noise satisfactorily,

using a lining material of a type to which Equation 6 applies, and having an absorption coefficient of 0.40 at 256 cycles. Assume that the duct is only 12 ft long as shown in Fig. 4, and that a 30 db reduction is required in this length.

Solution:

Case 1. (No splitters), From Equation 6,

$$R_0 = 12.6 \times 12 \times \frac{60}{200} \times 0.40^{1.4} = 12.6 \text{ db}$$

Case 2. (Two splitters, three channels), From Equation 7,

$$R_s = 12.6 \times \frac{10 + (20 \times 3)}{10 + 20} = 29.6 \text{ db}$$

AIR SUPPLY OPENING NOISES

When air is introduced into a room through a grille or register at a constant velocity, sound energy is being introduced into the enclosure at a constant rate[8]. Due to partial reflection at the boundaries of the en-

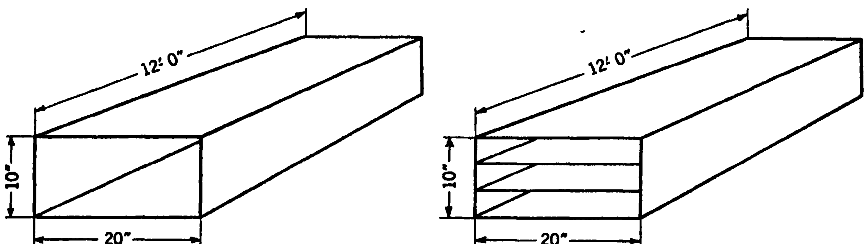

Fig. 4. Diagram of Branch Duct Treatment Where Length is Insufficient for Adequate Absorption

closure, the intensity of sound at any point in the space builds up to some maximum value. In a large room at a point remote from the source of sound (the supply opening) the intensity can be shown to be substantially proportional to the rate at which sound energy is generated and inversely proportional to the number of sound absorption units (sabines) in the room. It would thus appear that doubling the sound absorption of the room would halve the intensity and result in a noise level decrease of 3 db.

Grille noise is similar in character to fan vortex noise. Knowing the noise level at the face of a grille for a given grille blade setting the noise will vary as given in Equation 8 where V is the velocity of the air through the grille.

$$db \text{ (change)} = 55 \log_{10} \left(\frac{V_2}{V_1}\right) \quad (8)$$

For a change in blade setting Equation 9 applies and in this case the total pressure is measured directly behind the face of the grille. For a typical air conditioning grille the noise level at the grille face may be approximately 48 db with a total pressure behind the grille of 0.1 in.

$$db \text{ (change)} = 27.5 \log_{10} \left[\frac{\text{(Total Pressure)}_2}{\text{(Total Pressure)}_1}\right] \quad (9)$$

[8]The Noise Characteristics of Air Supply Outlets, by D. J. Stewart and G. F. Drake (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 81).

The resultant room noise level can be approximated by Equation 10.

$$\text{Room Level} = \left[\begin{matrix}\text{Noise Level at Face} \\ \text{of Grille}\end{matrix}\right] - 10 \log_{10} \frac{\text{Total Room Absorption in Sabines}}{\text{Total Grille Area}} \quad (10)$$

Grille Selection

In practice the allowable total sound and the required air flow are usually known, and it is desired to determine the maximum allowable velocity. In comparing sound ratings of various grilles several factors must be known if the information is to be properly applied:

1. The threshold intensity on which the decibel ratings are based.
2. The distance from the grille at which data were taken.
3. If stated as loudness level versus velocity for a given grille, the *core area* (not nominal area) must be known.
4. The sound absorbing characteristics of the test room.
5. Whether or not corrected for test room loudness level: if not, the room level (without grille noise) must be known.
6. Methods used for recording data. (Characteristics of sound meter).

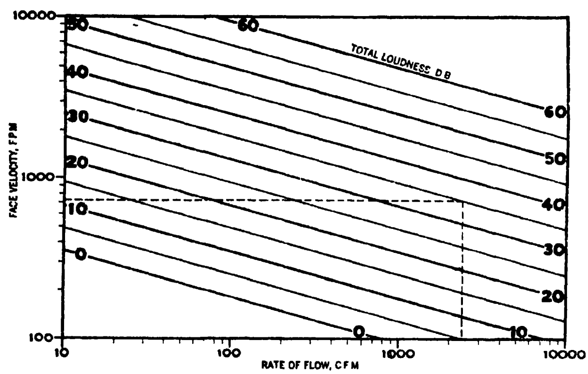

Fig. 5. Air Flow and Loudness Chart

Since total loudness and air flow are both functions of velocity and area, the solution of the problem implies a trial and error method. It has been found possible to present these data with sufficient practical accuracy as a family of uniform curves, as illustrated in Fig. 5, which are based on these assumptions:

1. Threshold intensity = 10^{-16} watts per square centimeter[9].
2. Microphone location 5 ft from lower edge of supply opening on a line downward at 45 deg and in a plane bisecting the supply opening perpendicularly.
3. Where data are given as loudness level versus velocity, the rating is per square foot of core area.
4. The room is assumed to have 100 sabines absorption.
5. Plotted data are loudness levels of *supply openings only*, correction having been made for test room level.
6. Data taken with a direct reading sound-level meter with frequency weighing network intended to approximate the response of the human ear.

If the published ratings are in terms of decibels per square foot, cor-

[9]Loc. Cit. Note 1.

rection must be made for area to secure the total sound level of supply openings of more or less than one square foot area from Equation 11.

$$\text{Decibel Addition} = 10 \log_{10} A \qquad (11)$$

where

A = core area, square feet.

With Fig. 5 it is possible to find directly the velocity in feet per minute which will give a predetermined total loudness at a predetermined rate of flow expressed in cubic feet per minute. The values used are arbitrarily chosen for the purpose of discussion and do not necessarily represent data

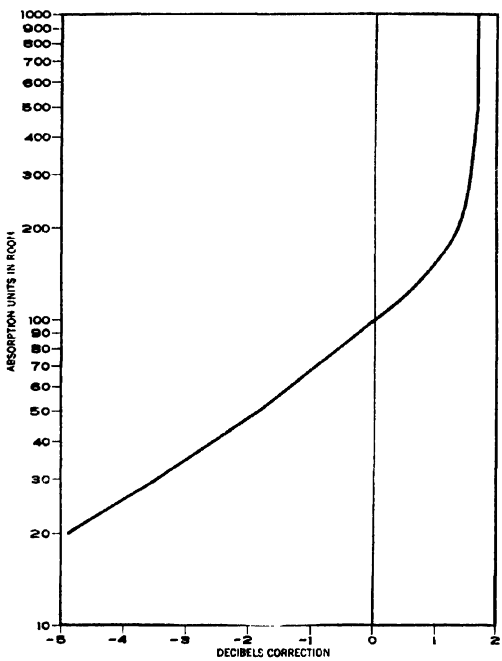

FIG. 6. ROOM ABSORPTION CORRECTION CHART

referring to any particular design of air supply opening. A correction chart is shown in Fig. 6 for a room having a sound absorption other than 100 sabines.

Example 3. Determine the core area (see Chapter 30) of an air supply grille which will maintain a noise level of not more than 40 db in a room having 100 sabines of sound absorption, if an air volume of 2400 cfm is required to maintain the proper air conditioning.

Solution. Assuming a grille noise rating of at least 5 db below the noise level of the room, Fig. 5 shows that the limiting grille velocity for a total loudness of 35 db is about 725 fpm and the core area becomes fixed at 2400 ÷ 725 or 3.31 sq ft.

If the room absorption had been greater, the previously selected velocity of 725 fpm would be safe, since the loudness reduces. If the room absorption had been 200 sabines a correction of plus 1.3 should be made by reference to Fig. 6, and the permissible velocity becomes that corresponding to a total loudness of 36.3 or approximately 800 fpm.

If the room had been highly reflective with an absorption of less than 100, the correction would be much more important. For instance, for a room of 35 sabines a correction of minus 3 db should be made and the maximum velocity corresponding to the 32 db total loudness would be approximately 600 fpm.

Where more than one supply opening must be considered, the problem is more complicated. If a similar supply opening is added in a far corner of a highly absorbent room, the change in noise level at the 5 ft station at the first supply opening is small; however, if the room is small, or highly reverberant or both, the intensity at the 5 ft station may be almost doubled and the noise level increased nearly 3 db thereby. The simplest method of handling this problem is to treat the room as though all the air were being supplied by one supply opening. Thus, if two outlets, each supplying 1000 cfm are used, the value 2000 cfm should be used with Fig. 5. Although this method may place an unwarranted limit on velocity. when used in a large room, it is seldom that such a room has a noise level low enough to justify a more complicated though more exact procedure.

In general, return grilles are selected for velocities about half the supply velocity, and when this is done, they may be neglected in sound computations. However, if supply and return grilles are the same size, resulting in the same face velocity, they must be treated as two supply openings. That is, if 1000 cfm are supplied and exhausted through grilles of the same area, 2000 cfm must be used in the solution with Fig. 5.

CROSS TRANSMISSION BETWEEN ROOMS

Ducts serving more than one room permit cross talk between the rooms and should be lined with acoustical material. Where the rooms are close together and the ducts short, the ducts should be sub-divided to provide ample acoustical treatment. Lagging material similar in character to acoustical board, when placed on the outside of ducts serves to prevent noise originating outside the ducts being carried inside the ducts and into the air stream.

A case where outside lagging is desirable occurs when ducts originate at the fan in the equipment room and pass through this room on the way to the room being conditioned or ventilated. Unless the ducts are lined some of the mechanical noise from the equipment room air may be transmitted through the wall of the duct, thus reaching the air stream and be carried into the room. In such cases, that portion of the duct which is exposed to the sounds in the equipment room should be lagged with material such as cork, pipe covering or other sound damping material to prevent the sound from entering the duct at this point. Numerical data are not available to permit a simple and practical calculating procedure to determine thickness of covering which should be used for this purpose.

Laboratory measurements have shown that the loss through a sheet of No. 22 gage metal is 24 db. When a sheet of rock wool insulation 1 in. thick and weighing 1.4 lb per square foot is added to this, the insulation value is increased to 29 db. In general, however, adding a layer of insulation or pipe covering does not materially increase the sound insulation value unless the material is dense, or unless it is surfaced with another sound impervious layer such as metal or board. Standard reference books should be consulted for sound insulating properties of various materials. Inside lining material used in the case previously mentioned would serve as an absorber of the sound transmitted through the duct walls, and thus act as a means of preventing the transfer of noise into the air stream. Inside lining may also be used in ducts to absorb noise which reaches the air stream from equipment such as fans, sprays and coils; noise due to eddying currents set up by elbows, dampers and similar obstructions; and noise transmitted from room to room where there is a common duct system.

CONTROLLING VIBRATION FROM MACHINE MOUNTINGS

It is impossible to select equipment which will operate without producing some mechanical noise, and since the equipment must be mounted in a building, it is probable that a part of this noise will be transmitted to the building to such a degree as to make noisy conditions in the rooms which are to be air conditioned.

Much of this noise may be transmitted by the duct if it is rigidly connected to the fan outlet. It is common practice to make the connection between the fan and the duct with a canvas sleeve which effectively restricts noise at this point. Noise may also enter the building through the mounting of the motor and the fan. Flexible mountings should be provided in all installations but these mountings must be carefully designed so that they will actually reduce the energy transmitted between the machinery and the supporting floor. If a flexible material is used, it is desirable to investigate the installation so that it is not short-circuited by through bolts which are improperly insulated and by electrical conduit which is not properly broken and is attached both to the equipment and to the building. The flexible mounting, if improperly engineered, may actually increase the energy transmitted between the equipment and the floor upon which it is supported.

In the proper isolation of vibration, which is usually in the lower range of frequencies and does not include the *air borne* vibrations known as sound, there is one basic law which is important in the solution of the problem. That is the law of transmissibility as governed by the equation:

$$T = \frac{1}{1 - \left(\frac{w}{w_n}\right)^2} \tag{12}$$

where

T = transmissibility of the support.
w = frequency of the vibratory force.
w_n = natural frequency of the machine unit on its support (Damping = 0).

Equation 12 shows that the transmissibility approaches unity for disturbing frequencies considerably lower than the natural frequency of the mounting. As the disturbing frequency is increased the transmissibility is also increased until at the resonant frequency, where $w = w_n$ the transmissibility becomes infinite. This is not true in practice because all materials have some internal damping effect. However, operating at or very close to the resonant frequency is always serious as forces and stresses may be multiplied 10 to 100 times. As the disturbing frequency becomes greater than the natural frequency the transmissibility becomes a smaller quantity and at the value of $w/w_n = \sqrt{2}$ it again has the value of unity. Beyond this point true isolation is first accomplished. At a ratio of 3 to 1 for w to w_n the isolation is effective enough for practical application, and experience and economical design have shown that a ratio of 5 to 1 is good. For high speeds, higher ratios for w to w_n are easily attained and give better results for effective vibration control but for the lower speeds as experienced with compressor work the higher ratios become uneconomical.

For a given installation the speed of the compressor is fixed by the specifications, therefore the value of w is fixed. That leaves only w_n to be determined and that is accomplished by the choice of mounting material and design for the support of the machine. It is well to keep in mind that

when trying to isolate vibration, no attempt should be made to isolate the driving and driven piece of equipment separately. The two should be mounted on a rigid frame and then the entire assembly isolated according to the rules presented in this chapter.

The value of w_n can be controlled by the flexibility of the machine support, and when the deflection of the machine support is proportional to the load applied (such as with springs or nearly so with rubber in shear) the value of w_n can be determined by Equation 13.

$$w_n = \sqrt{\frac{g}{d}} \qquad (13)$$

where

g = gravitational constant.
d = static deflection of supporting material.
w = radians per second and may be converted to frequency (f) expressed in cycles per second by Equation 14.

$$f = \frac{w}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{g}{d}} \qquad (14)$$

By the use of Equation 13 a set of curves may be plotted as shown in Fig. 7. The first line *AB*, plotted as the critical frequencies for the various static deflections, is a curve showing the worst possible conditions or resonant conditions.

Plotting another curve *CD*, which is $\sqrt{2}$ times curve *AB*, shows the area *MCDN* in which the resilient material or mounting does more harm than good. Plotting two more curves *EF*, 3 times curve *AB*, and *GH*, 5 times curve *AB*, shows area *EGHF* which represents efficient and economical isolation. Area *GPOH* is excellent isolation but for all except the highest speeds becomes rather uneconomical because of the large deflections required.

Example 4. An electric motor driven compressor unit is to be isolated. The compressor is partially balanced and operates at a speed of 360 rpm. The speed of the motor is 1160 rpm and is belt connected to the compressor. Total weight of the compressor and motor is 4500 lb.

Solution: The minimum disturbing frequency to be isolated is 360 cycles per minute. Assume that the desired ratio of forced to natural frequency is 3 as a minimum and that 5 is desired. The desired natural frequency of the mounting is 360 ÷ 5 = 72 cycles per minute.

From Fig. 7 a deflection of 7 in. is required to attain a natural frequency of 72 cycles per minute. This value may be obtained from critical curve *AB* for 72 cycles or from curve *GH* (5 times critical) for 360 cycles. For the minimum ratio of 3 the deflection would be 2.5 in.

The next step is to determine the total weight to be supported by the springs. For low speed partially balanced compressors, it has been found necessary to add a foundation weighing 2 to 3 times the weight of the motor and compressor, in order to maintain the machine movement below 0.03 in.

Compressor and motor	4,500 lb
Concrete foundation	9,000 lb
Total	13,500 lb

Practical application dictates the number of springs to be used, which is based on the design of the machine foundation and the supporting floor structure. However, it is desirable to design for at least 8 springs and one or two spares for cases of unknown weights. As many as 50 springs have been used on one installation. The distribution of the springs must be balanced against the masses to be supported, otherwise the foundation design and supporting structure determine the location of the springs.

The choice of the material used in the design of the resilient mounting is also important. For the slow-speed type compressor a common speed found in practice is 360 rpm. For speeds below this, isolation should not be attempted except under careful supervision. Referring to Fig. 7, it is found that for 360 rpm the static deflection required for a ratio of w/w_n of 3 to 1 (line *EF*) is 2.5 in. and for a ratio of 5 to 1 (line *GH*) it is 7 in. For these values of deflection the only choice of material is the coil spring. This is also true for speeds up to about 700 rpm. In consideration of the transverse spring constant (so as to maintain good ratios among the various degrees of freedom) experience has shown that the spring should

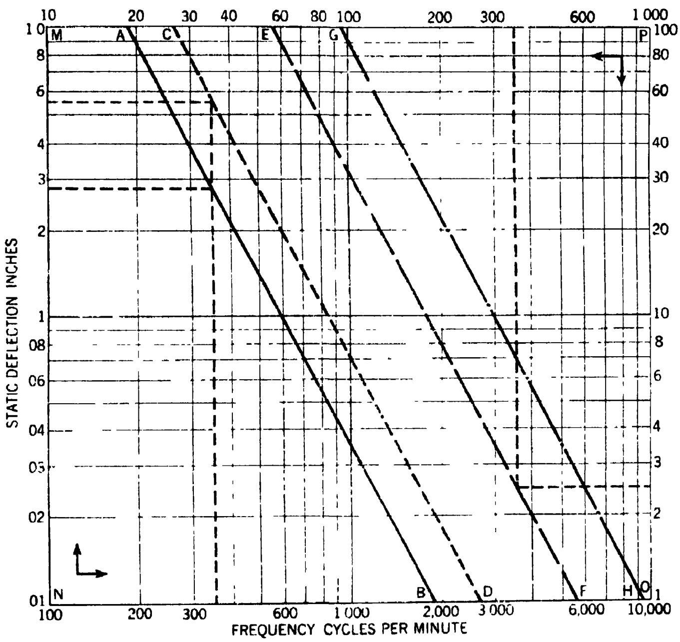

FIG. 7. STATIC DEFLECTION FOR VARIOUS FREQUENCIES

be designed with a *working height* equal of 1.0 to 1.5 times the outside diameter. A long spring of small outside diameter has very low transverse rigidity and therefore requires some additional means of preventing side drift of the unit and on very sensitive applications this may tend to destroy the isolation efficiency. For speeds of 700 to 1200 rpm the required deflections range from 0.22 in. to 1.75 in. For these conditions rubber in shear serves as a rather satisfactory material if protected from oil. For speeds higher than 1200 rpm cork specially made for vibration damping can be applied with good results. These limitations are by no means absolute because with careful and well engineered installations, especially, in consideration of all six degrees of freedom, certain liberties may be taken and still good results accomplished.

When a machine unit is properly isolated it will have a definite amount of movement which is determined by the ratio of the unbalanced forces to the total mass of the machine. If this resultant machine movement is too great for the necessary connections or the satisfaction of the customer

it can be reduced only in two ways without destroying the quality of the isolation; first, adding mass or dead weight to the machine (such as concrete) common in the application of low speed, partially balanced machinery; second, accurately balancing (both statically and dynamically) all moving parts so as to eliminate the vibration at the source. This latter method is the best engineering practice and is the modern trend. However, even with well balanced machinery, installed in the vicinity of quiet offices it is usually necessary to properly isolate the equipment to prevent the transmission of vibration likely to cause complaints.

Where limitation of machine movement is desired during the starting and stopping periods, the application of friction or hydraulic damping will serve without seriously interfering with the efficiency of the isolation.

CHAPTER 33

Automatic Control

Basic Types of Control, Types and Functions of Controllers, Designation of Temperature Control, Automatic Control Terminology, Residential Control Systems, Control of Automatic Fuel Appliances, Zone Control, Control of Unit Systems, Control of Refrigeration Equipment, Application of Control Devices to a Typical System

THE function of *automatic control*, as applied to the heating, ventilating and air conditioning industry, may be broadly subdivided into the maintenance of *temperature, humidity, pressure*, and *liquid levels* within predetermined ranges. It automatically coordinates the operation of the various controlled mechanisms in proper sequence to produce the desired result. A thermostat, for instance, is a controller responsive to temperature which initiates a force that repositions valves, dampers, etc., as required to maintain selected temperatures. Similarly, a hygrostat or humidistat is a controller responsive to relative humidity which initiates a force that repositions the controlled sources as required to maintain a selected relative humidity.

BASIC TYPES OF CONTROL

There are two basic types of controllers, (1) the *two-position*, *on-off*, or *positive acting controller*, which functions to move a valve, damper or electric switch through its full travel (not necessarily 100 per cent) from open to closed position and vice versa, and (2) the *throttling*, *proportioning*, or *intermediate acting controller* which functions to reposition the valve, damper, or other controlled device by small increments of its total travel as the controller senses a slight change in the controlled condition. The available equipment may be broadly sub-divided into three main groups, (*a*) *self-contained equipment*, (*b*) *electrically operated equipment*, and (*c*) *pneumatically operated equipment*.

Self-Contained Equipment

In *self-contained controllers* the primary source of power is the vapor pressure of an enclosed volatile liquid which varies with the temperature changes of the controller bulb. Such pressure changes are transmitted directly into the bellows or diaphragm mechanism of the controlled valve. Devices of this type produce gradual operation and are used to regulate the flow of heating and cooling media to coils, radiators, and liquid tanks. Each instrument is designed to operate within a predetermined selected temperature range and for a given condition. Care must be exercised to select a type which has the ability to withstand any possible over-run of temperature at the control bulb without damage to the instrument. Instruments of this type are available either with a rigid bulb or with a flexible tubing which connects the control bulb to the head of the instrument. This tubing is usually protected by a flexible metal covering or armor and is available in varying lengths.

Electrically Operated Equipment

Electrically operated equipment for either two position control or throttling control is available in either low voltage or high voltage type. Low voltage equipment usually is designed to operate on 20 to 24 volt current. High voltage equipment may be used in connection with any of the

the removal of the controller bulb without the necessity of draining the vessel to the bulb level.

A *differential thermostat* has two flexible capillary tubes fitted with plain bulbs, or with bulbs having flanges or having threaded connections. One sensitive element is subjected to the controlled temperature and the second element is exposed to the temperature which is to determine the controlled temperature. The instrument functions to maintain a definite relation between the two temperatures.

A *remote bulb or extended tube controller* may be of liquid, vapor, or gas filled type. A bulb of this type of instrument is connected to the instrument head by means of a flexible capillary tube of the desired length. The external diameter, bore, and material used to form the capillary vary with the type of control system.

An *indicator controller* is a controller fitted with a pointer, thermometer, or gage which indicates the controlled condition.

A *recorder controller* is a controller, combined with a clock mechanism and chart, which both controls and records the controlled condition.

A *time cycle controller* is a controller equipped with a clock mechanism which automatically raises and lowers the control point of the instrument to meet a required time schedule.

A *hygrostat or humidistat* is an instrument for controlling relative humidity. It may be of a room, insertion, indicating or recording type, and may be direct or reverse acting.

Many forms of hygroscopic materials, such as human hair, wood, bi-wood and membrane of selected type, are used as the sensing medium for these instruments. Where the controlled condition ranges below 20 per cent or above 70 per cent, or the dry-bulb temperature is above 100 F, careful selection of the hygroscopic element is essential.

A *static pressure controller* is frequently used to provide a constant pressure in a duct system in which the volume of air supply is varied to maintain temperature or some other condition, and usually it operates to reposition dampers installed in the fan vortex or other selected location. It usually functions to maintain a pressure differential between the atmosphere and the pressure or vacuum of the controlled condition. It is frequently necessary to pipe the neutral chamber to a selected location to obtain a true basic condition.

A *velocity regulator* is a controller actuated by the velocity of the medium which it controls.

A *solenoid air valve or electric pneumatic switch* is an electrically actuated three-way air valve which may be either direct or reverse acting. It is frequently used in connection with fan ventilating systems to close outdoor intake dampers and heater coil valves during the period when the fan motor is stopped.

A *pneumatic relay or cumulator* is a device fitted with a readjusting diaphragm to which a separate pilot connection is made. It is installed in the branch between the controller and the controlled valves or dampers, and functions to accelerate, magnify or reverse the action of the controller. Direct acting relays increase the branch pressure on an increasing pilot pressure. Reverse acting relays decrease the branch pressure on increase of pilot pressure.

A *duplex cumulator or relay* is a device having two pilot connections either of which will vary the branch pressure on a changing pilot pressure.

An *averaging cumulator or relay* is a device arranged to vary the branch pressure as dictated by the average of the pressure of two or more pilot connections.

A *multiple cumulator or relay* is a device having more than two pilot connections any one of which will vary the branch pressure on a changing pilot pressure.

A *control valve* may be considered as a variable orifice designed to control the flow of liquids, gases and other fluids.

The use of the flat disc type of valve is usually restricted to on-off control requirements. Throttling valves are generally equipped with selected forms of plug to produce the flow characteristics required for the special problem. Special plugs can be shaped to produce almost any desired flow characteristics. The three types of plugs most widely used are the V-port skirt type, the throttle plug, and the ratio plug.

A *diaphragm valve* is a valve which is operated by a diaphragm attached to the valve stem. Increasing air pressure on the diaphragm moves the valve against the pressure of opposing springs which return the valve to its normal position on a decreasing diaphragm pressure. The pressure range required to fully reposition a valve has four limiting factors, viz., the size of the diaphragm top, the tension of the selected spring, the packing gland friction and, for all single seated valves, the pressure of the controlled media against which the valve must hold.

A *direct acting or normally open diaphragm valve* will assume an *open* position due to spring action when all operating power is removed.

A *reverse acting or normally closed diaphragm valve* will assume a *closed* position due to spring action when all operating power is removed.

A *double seated balanced or semi-balanced valve* is designed for use with high pressures and is practically balanced regardless of the pressure differential across the plug. The diaphragm tops for valves of this type need only have ample power to close the valve against the tension of the opening spring plus the friction of the valve packing gland. While plugs and seats for this type of valve can be arranged to provide for tight seating for special purposes, such a requirement is not ordinarily expected.

A *three-way mixing valve* having two inlets and one outlet, is fitted with a double faced disc, operating between two inlet ports, which functions to close one port as the other is opened, and is used for mixing service.

A *three-way diverting or by-pass valve* has one inlet and two outlets with two separate discs operating on the outside of the respective valve seats to close one as the other is opened; and is used for diverting service.

A *pilot positioner or valve positioner* is an auxiliary control device repositioned by the controller pressure and having a direct connection to the valve stem. It permits the full available air pressure to be used on the diaphragm top to overcome hysteresis, packing gland friction and pressure variation of the controlled media, to position the valve precisely in accordance with the controller pressure.

A *pneumatic damper motor* is a diaphragm or bellows assembly operating against resisting springs to reposition an arm or lever assembly. They may be attached directly to the damper frame, either inside or outside of same, or fitted with brackets for wall or floor mounting. The springs used to return the motors to their normal positions are available in several tensions to permit their operation in selected sequence. Pilot positioners may be applied to damper motors in the same manner as described for diaphragm valves.

Pneumatic dampers in either the single blade or multi-louver type should be substantially built in heavy frames, properly braced to hold true against any sag in the supporting structure, in order to prevent distortion of the louvers and binding of their supporting trunnions. When large dampers are used for open and shut control, the regular multi-louver type of damper serves the purpose. Where accurate volume control is desired with gradual positioning of the damper blades, the adjacent louvers should be arranged to move in opposite directions in order to properly proportion the available free area through the damper from its fully open to its fully closed position. Multi-louver dampers with louvers which all move in the same direction provide full free area through all but their top and bottom louvers when the louvers reach 45 deg open position. Where fairly tight closing is desired, the damper frames should be fitted with solid stops against which the louvers can close and where additional precautions are necessary, rubber or felt stripping may be glued and riveted to the louver edges. The metal frames of the smaller dampers are usually made of ¼-in. stock. Solid stops are usually ½ in. deep. The free area of the duct enclosing the damper is thereby reduced 1½ in. in each dimension through the damper. Where channel frames are used, the restriction is increased accordingly. Damper louvers should be suspended on non-corrosive trunnions. Thrust bearings are necessary to properly support large vertical louvers. Where ball or roller bearings are required for special conditions, they should be of the non-corrosive type.

DESIGNATION OF TEMPERATURE CONTROL

Temperature may be controlled from three basic conditions:

Dry-bulb Control: The controller element senses the dry-bulb temperature.

Wet-Bulb Control: The controller element senses a true wet-bulb temperature. Wicking of selected material suspended from the controller element to a water trough directly below the element is one of the mediums frequently used to obtain wet-bulb temperature. True wet-bulb temperatures can only be maintained with clean wicking and air motion of not less than 600 fpm. The use of distilled water in the wet-bulb reservoir is desirable. Since the wet-bulb temperature indicates the heat content, this form of control while requiring additional and intelligent care to maintain true wet-bulb conditions, has many definite uses.

Dew-Point Control: A saturated condition at a predetermined temperature is maintained.

AUTOMATIC CONTROL TERMINOLOGY

The following is a generally accepted terminology referring to the operations of controls in a central air conditioning system.

Control Point: The desired constant condition of the controlled media to be main-

tained. Some change of the controlled condition must occur to alter the prevailing controller action.

Thermometer Lag: The instantaneous temperature differential existing at any time between the true temperature of the medium and the temperature indicated on the thermometer.

Controller Lag: The time delay in the controller's ability to reposition the controlled sources to properly compensate for a change in the controlled condition.

Process Lag: (Process Time Lag) The time which elapses between the instant the control valve is positioned by the controller and the instant the controller element senses the effect of the change. This depends upon the rate of heat transfer, the specific heat, and the velocity of the fluid movement. Generally a decrease in process time lag simplifies the control problem.

Rate of Load Change: Time element of change in the magnitude of load.

Magnitude of Load Change: The per cent load at a specified time. It directly affects the size of the controlling valve. Where the inlet temperature varies constantly, the control problem becomes more difficult as the thermal difference between the inlet temperature and the leaving temperature of the controlled media becomes greater.

Drift: The deviation from the control point due to change in load conditions from 0 to 100 per cent.

Drift increases as the sensitivity adjustment of the controller is lowered, thereby tending to decrease hunting. Drift decreases as the sensitivity adjustment of the controller is raised, thereby increasing hunting.

Hunting or Cycling: The temperature variation between the high and low points of the control cycle due to lag. This balances out at the point where each successive wave of temperature reaching the controller is as large as its predecessor. As the sensitivity is lowered, hunting decreases but drift increases. Oversized control valves increase hunting.

Throttling Range or Sensitivity Adjustment: The total rise in temperature at the controller bulb required to reposition the control valve from a fully open to a fully closed position or to operate several devices controlled in sequence throughout the full sequence range. In connection with chart type controllers, throttling range is usually expressed in per cent of the total range of the chart. For other types of instruments, sensitivity adjustment is expressed in pounds variation of branch pressure per degree change in temperature, or per other unit of change of the controlled media.

Low Sensitivity: A small change in branch pressure for a comparatively large temperature or pressure change in the controlled medium. This results in a greater deviation from the control point. Low sensitivity tends to reduce hunting but the lower the sensitivity, the greater will be the drift.

High Sensitivity: A large change in branch pressure for a comparatively small temperature or pressure change of the controlled medium. The extreme of high sensitivity is the on-off or two-position control. As the sensitivity is raised, drift decreases, but the tendency to hunt increases.

Automatic Reset: A mechanical addition to a controller which corrects for drift by constantly resetting the instrument to operate on the control point, while repositioning the controlled valve or damper from its fully open to its fully closed position. Reset action is superimposed on throttling range and both function simultaneously. The rate of reset is manually adjustable and must be set to meet the load requirements of the individual system. This supplementary device provides the accuracy of high sensitivity adjustment with a minimum of hunting.

Diaphragm hysteresis is the amount by which the stem travel of a given motor fails to assume the identical position on the upward and downward strokes for an identical air pressure in the diaphragm top. It prevails in varying degrees in connection with the operation of all pneumatic diaphragm motors, and is due to spring and diaphragm hysteresis and friction of guide bearing and stuffing box. Assume that a direct acting valve moving to a closed position will travel 50 per cent of its total with 8 lb air pressure on the diaphragm top. If the diaphragm pressure is increased to further close the valve and then reduced to 8 lb, the valve may be repositioned to be only 45 per cent open, indicating 5 per cent hysteresis. Hysteresis is usually expressed in per cent of the full stroke of valve stem.

RESIDENTIAL CONTROL SYSTEMS

The control installation in a residence may vary from the simple regulation of a coal-fired heating plant to the completely automatic all year air conditioning system. Residential installations with automatic fuel burning appliances, such as oil burners, gas burners or stokers, are

normally equipped with single room thermostat, limit and safety controls as outlined under Control of Automatic Fuel Appliances.

Coal-Fired Heating Plant

Control in the normal coal-fired domestic heating plant consists of regulating the combustion rate in accordance with requirements. This function is accomplished by a spring or electric-driven damper motor which, under the command of a room thermostat and through chain linkage, operates the draft and check dampers of a boiler or warm air furnace. Such installation should be protected against excessive temperature or pressure by means of a limit control serving to check the fire when conditions at the boiler or furnace reach a predetermined maximum.

All Year Domestic Hot Water Supply

Hot water or steam heating boilers with automatic fuel burning appliances can be used for all year heating of domestic water supply. The fuel burning appliance in this case is controlled from the temperature of water or pressure of steam in the boiler to maintain uniform boiler conditions and domestic hot water is heated by means of an indirect heater. The heating of the residence is normally governed by means of a thermostat which operates a control valve in the flow line of a gravity hot water or a steam system or which controls the circulating pump in a hot water system.

Air Conditioning Systems

Residential air conditioning systems normally include a heating source and a motor-driven fan for circulating air. In addition, such installations may involve spray-head equipment to supply humidity. Such installations distribute heated and humidified air during the heating cycle, and during the summer or cooling cycle may be used effectively as conditioners if equipped with refrigeration means.

Regulation of the humidity during the heating cycle is normally accomplished by opening and closing a solenoid water valve supplying water to the spray-heads, the solenoid valve being under control of a room type humidity control. In the average installation the fan is permitted to run only during such intervals as the thermostat is calling for heat or at the command of a limit control to prevent the overheating of the bonnet of a warm air furnace. The limit control should also prevent the operation of the fan at the command of the thermostat until the circulating air temperature has increased to a predetermined point.

For the cooling equipment provided in such installations, control during the cooling cycle will be an adaptation of the control principles described for central fan systems selected for the type of cooling equipment utilized.

CONTROL OF AUTOMATIC FUEL APPLIANCES

It is essential that automatic controls be used with oil burners, gas burners, and stokers in order to maintain even temperatures and provide safe and economical operation of the heating plant.

Oil Burner Controls

In the normal oil burner installation as encountered in residential and small commercial installations, the burner operation is frequently regulated by electric controls and primarily governed by a room thermostat. It is essential that a limiting control be incorporated in the control system to prevent the temperature of the heating medium from exceeding any

predetermined safe maximum. The type of limit control selected will depend on the type of the heating system. In a warm air furnace installation, a limit control would be used, reacting to the temperature of the heated air in the bonnet of the furnace; in a hot water system a control reacting to the temperature of the water in the boiler; and in a steam system a control reacting to the pressure of the steam in the boiler.

In addition to the normal control of the burner from the room thermostat and limit control, it is necessary that a combustion safety device be used to prevent operation of the burner under hazardous conditions. The oil fire is automatically ignited by means of gas, electric spark or incandescent element and the combustion safety control acting through a sequence device permits the burner operation only when the fire is properly established as the burner starts up. A further function of the combustion safety control is to react to any major disturbance in the flame during the running operation, shutting down the burner and preventing the discharge of unburned fuel if for any reason the flame is extinguished.

Gas Burner Controls

In the case of the domestic burner, full automatic operation is the normal requirement and the burner is started and stopped at the command of a room thermostat which, in turn, opens and closes a control valve in the gas supply line. Modulating controls and controls providing a high and low fire are also available for gas burners. For purposes of preventing abnormally high temperatures in the bonnet of gas-fired furnaces or in the temperature of the water in gas-fired hot water heating boilers or excessive pressures in gas-fired steam boilers, temperature and pressure limit controls are used. Ignition is normally secured through the use of a gas pilot flame and a safety device is provided, utilizing the heat of the pilot flame in such a manner that if the pilot light is extinguished for any reason, the main gas valve cannot be opened. For satisfactory and economical operation, all automatically-fired gas burners should be equipped with pressure regulators on the gas supply line.

Stoker Controls

Domestic stokers are normally placed under command of a room thermostat for primary operation subject also to the command of a limit control to prevent their operation when conditions in the boiler or furnace exceed predetermined safe maximums. Utilizing coal as fuel, automatic ignition is not provided and the stokers, once ignited, maintain their fire, merely changing the rate of combustion by changing the draft and the rate at which the coal is fed. Thus, at the command of the room thermostat the stoker motor is started, driving a forced draft fan and fuel feeding mechanism. The rate of combustion is thus increased and this operation continues until the thermostat has been satisfied when the motor is stopped and the fuel in the combustion chamber continues to burn at a slow rate with reduced draft.

Automatic controls may be used to operate the stoker sufficiently to maintain a fire in mild weather and also to prevent feeding of fuel if the fire is extinguished.

ZONE CONTROL

Zone control for winter heating of buildings in which a single room thermostat will not provide satisfactory conditions and where the refinement of control from individual room thermostats is not a requisite, may be secured in various ways. According to its size, use and exposure, the

building may be divided into zones where the general requirements will be relatively constant. The number of zones is determined by (1) size of building, (2) number and character of exposures, (3) variation in occupancy and other inside conditions, and (4) cost of additional zones. For large buildings it is desirable to have at least one zone for each exposure and for high buildings, each exposure may be sub-divided into upper and lower zones in order to properly provide for the stack effect. In buildings of this type it is advisable to provide a separate main with local thermostats for the street floor heat sources, especially those adjacent to entrances and exits.

For small buildings or where cost or other conditions limit the number of zones, a frequent compromise is to combine the North and West exposures in one zone and the South and East exposures in a second zone.

For steam heating systems, the radiator output may be proportionately reduced as the outdoor temperature rises by several methods. Some of those in common use are:

1. Throttling steam pressure to allow flow through orifices in proportion to the needs for heating.
2. Turning steam on and off on time intervals proportioned to the needs for heating.
3. Varying the absolute pressure of steam in the system.

The usual zone controlling device is arranged to sense the effect of sun, wind, or rain, as well as the outdoor local zone temperature, and in some instances is fitted with additional elements sensing the indoor zone or zone radiator temperatures. It functions to regulate the rate of flow or the flow impulse as required to maintain the desired indoor zone temperature condition. A switch panel containing manual switches arranged to raise or lower the operating point of each zone controller is frequently mounted at a selected location. These panels may contain time switches for automatic cycling of day-night or other predetermined control programs. In buildings where some zones or all zones have no night occupancy, provisions may be made to index the control system to maintain predetermined low economy temperatures during the unoccupied periods, a short morning warm-up period, and normal operation throughout the occupied period, by either manual operation or automatic cycling by time switch. For hot water heating systems, the zone controller functions to regulate the water temperature in accordance with the prevailing outdoor temperature.

For discussion of zoning in air conditioned buildings see section on Zoning in Chapter 20.

CONTROL OF UNIT SYSTEMS

Because of the usual segregated location of unit equipment throughout a building and its consequent lack of competent supervision, complete automatic control is essential to its satisfactory operation.

Unit Heaters

In its simplest form, unit heater control consists of a room thermostat to start the unit heater motor when heat is required and shut it off when the demand is satisfied. With this limited control, it is possible in some instances that, with no steam available at the heater, the operation of the fan would cause objectionable drafts. To avoid this, limit controls are available which will prevent the operation of the fan at the command of the room thermostat except when steam is available, as determined by

the temperature of the steam or return pipe or the pressure of the steam supply.

Where several unit heaters serve a limited area, they may be grouped for purposes of automatic control, and several heaters placed in operation at the command of one thermostat. By properly grouping the units which will operate together, the benefit of zone control can often be obtained with a minimum of control equipment. Where such group operation is utilized, the thermostat and limit control usually function through a relay, as the combined load of the several motors may exceed the current capacity of the thermostatic control device.

In some cases where cold drafts will not result, it is desirable to operate the unit heaters continuously for circulation of air. In such instances the room thermostat regulates the supply of steam to the unit through a control valve in the steam supply line and the unit heater motor operation is manually controlled.

Unit heaters equipped with dampers arranged for by-passing air around the heating coils are controlled by room thermostats operating modulating damper motors attached to these dampers so that as the temperatures rise, a decreasing amount of air is heated. When the by-pass is wide open the heating effect is so much reduced that control of the steam supplied to the coil is not generally important. If valve control is added, the throttling of the steam may be concurrent with, or subsequent to, the opening of the by-pass.

Cooling Units

The recommended form of temperature control for a cooling unit contemplates the continuous operation of the fan, with automatic regulation of the compressor or cooling coil, or both, as determined by a thermostat in the room, or in the return air to the cooling unit. Such operation insures continuous circulation of air in the room, and in addition to providing the cooling effect of moving air, overcomes the tendency of the air to stratify. As the temperature begins to rise, the controller opens the valve to a cold water cooling coil, or for direct expansion coils, opens a valve in the refrigerant line, closes a by-pass around the coil or starts a compressor.

Cooling units may also be controlled by arranging the room thermostats to start and stop the fan motors or by a combination of motor and refrigerant control.

Unit Ventilators

There are various types of unit ventilators available but in general all types are designed to draw air from the outside or to mix outside and recirculated air, heat it and introduce it into the room under control of a thermostat.

The design of unit ventilators has to an extent been based on the requirements for automatic temperature control and the cycles of control have been developed to include other heating devices in the rooms with unit ventilators. Unit ventilators are frequently used in schools and other types of buildings where states have laws or regulations governing the minimum amount of ventilation to be provided. The control of the amount of outdoor air is designed to conform to the various laws. Usually the device circulates a constant amount of air and the amount automatically taken in from outdoors is controlled in one of these ways:

1. Full recirculation until the room temperature reaches a certain point, generally two degrees, below the desired room temperature; then a minimum amount of outdoor

air for ventilation while the temperature is maintained by throttling steam; and if the room temperature rises with all steam shut off, a gradual increase in amount of outdoor air up to 100 per cent.

2. Full recirculation until the room reaches a set point below room temperature, after which all air is taken from outside.
3. Gravity recirculation while the fan motor is not running, with full outside air as soon as the fan starts, obtained by a relay in the motor circuit.
4. Full recirculation or all outdoor air as determined by a manual switch which can be operated at any time whether or not the fan is running. All the unit ventilators in a single building may be operated by one or many switches.

With arrangements 1 or 2, it is desirable to include a relay to prevent the intake dampers from opening while the fan is not running, regardless of room temperatures. With a dual system of control this is essential to prevent the thermostat keeping the outside damper open until the temperature falls to the reduced setting.

The intake and recirculated air quantities are determined by a single damper or by a pair of dampers working together, and operated by a damper motor. Although this affects the temperature of the air delivered, the main heat control comes from the throttling of the steam supplied to the heating coil, with or without by-pass damper control. To prevent air being delivered at too low a temperature, a low limit thermostat is commonly installed in the air stream and set at some point between 55 and 70 F. The lower settings may cause discomfort, the higher ones overheating, depending on circumstances. The air stream thermostat can be used to turn on steam, reduce the amount of outside air, or both.

Rooms with unit ventilators frequently have auxiliary heating devices, such as direct radiators, convectors or unit heaters, all under control of a single room thermostat. A common control cycle for such rooms is composed of the following functions, assuming that 72 F is desired:

1. Below 70 F the unit ventilator intake damper is in full recirculating position and all heat is turned on.
2. At 70 F the intake damper moves to a position that will admit a predetermined minimum amount of air from outdoors.
3. At 71 F the auxiliary heating devices are shut off.
4. From 71 to 72.5 F, the heating effect of the unit ventilator is throttled.
5. From 72.5 to 74 F, the intake damper is gradually moved to increase the amount of outside air from the set minimum to 100 per cent.
6. If the room thermostat calls for too much cooling, the air stream thermostat holds the delivery temperature at a proper minimum.

Other similar cycles may be used. One additional feature is the use of an air stream thermostat that has its control point reset by the room thermostat. Then as the room temperature rises, the delivery temperature is gradually reduced from a maximum to a minimum.

CONTROL OF REFRIGERATION EQUIPMENT

The most common means of providing cooling for air conditioning may be divided into four general classifications as follows:

Refrigeration compressors may furnish refrigerant to direct expansion cooling coils through which air is being passed, or to coils in cooling tanks through which water is passed which is then pumped to air washers or cooling coils through which the air is passed.

In either case the compressor motor may be started and stopped in order to meet the demand for refrigeration or a pressure controller may be used to regulate the low side or suction pressure of the compressor. When the latter method is used, the flow of refrigerant to cooling coils may be regulated by the opening and closing of a solenoid refrigerant valve at the command of a temperature controller or thermostat.

A high pressure cutout as an individual unit or in combination with either a temperature or pressure controller provides a safety feature against excessive pressures on the high side of the compressor.

Many compressors may be *unloaded* by instruments sensing room or duct conditions, or by refrigerant pressures, thus reducing the frequency of starting and stopping. If two or more compressors are used for a single cooling system, *step controllers* are used to start them in sequence at intervals of a few seconds to avoid the large momentary electric input that simultaneous starting would demand.

When condensers are water cooled, thermostatic control to vary the quantity of water is needed for economical operation. Mechanical air condensers may be started and stopped with temperature demands.

Chilled water may be stored in tanks at temperatures slightly lower

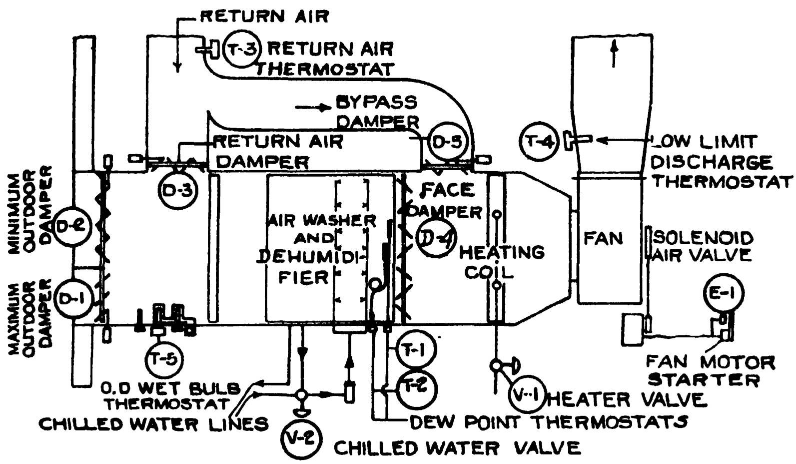

FIG. 1. LOCATION OF CONTROL DEVICES FOR YEAR 'ROUND AIR CONDITIONING SYSTEM

than required for air cooling coils. The control of temperature for the water distribution system is as described for Ice Cooling.

When ice is used for the cooling or dehumidification of air, it is usually placed in bunkers and water is sprayed over it. This water, after being cooled, may be used in air washers or surface cooling coils and is usually returned to the bunker for additional cooling after being used.

Control of the water temperature leaving the cold water tank may be maintained by a temperature controller, which measures the temperature of the water in the tank and modulates a control valve in a by-pass which permits a portion of the return water to return directly to the tank instead of passing through the sprays.

A vacuum refrigerating system consists of an evaporator, compressor, condenser and auxiliaries. The refrigerant used is water, and water vapor (steam) is the power medium.

Water which has been passed through an air washer or cooling coil is sprayed directly into the evaporator or water cooler where it is cooled by its own evaporation. A condenser is attached directly to the compressor

discharge and its function is to recondense the water vapor drawn from the evaporator, plus the steam which supplies the energy for compression.

The temperature of the cold water leaving the flash chamber should be measured by a temperature controller which will in turn operate a two-position or positive-control valve installed in the steam line to the jet so as to permit steam to flow only when cooling is required. If city water is

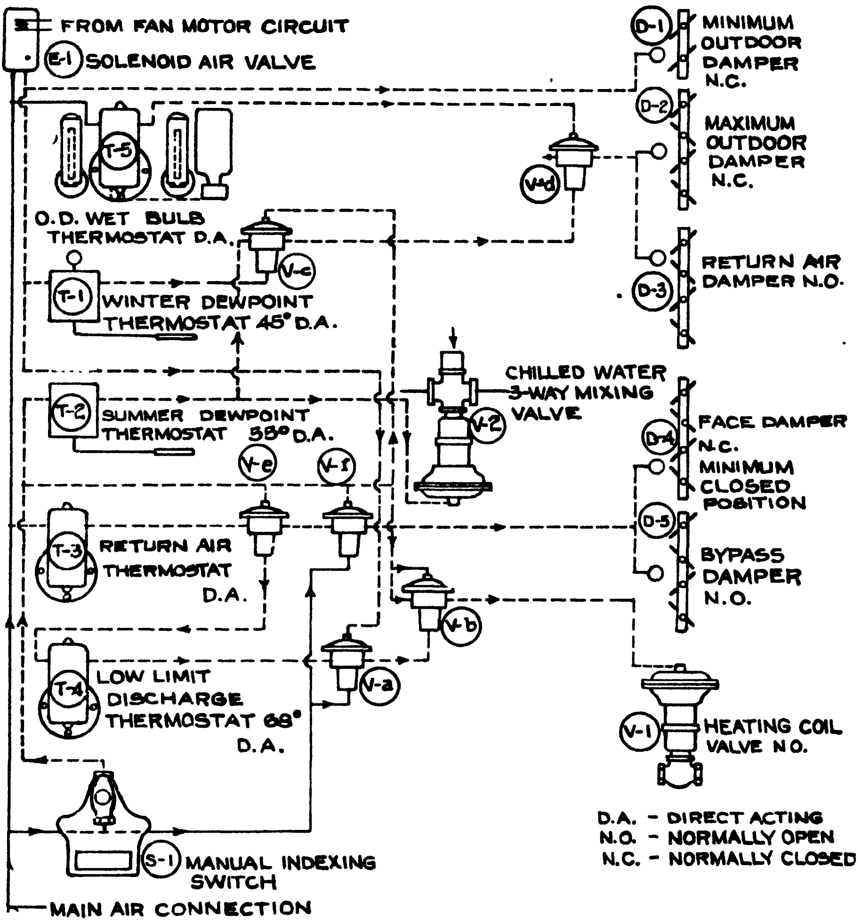

FIG. 2. CONTROL DIAGRAM FOR YEAR 'ROUND AIR CONDITIONING SYSTEM

used in the condenser, the amount of water should be modulated according to the demand as measured at the condenser outlet by means of a temperature controller and control valve.

Well water, if available in sufficient quantities at low temperatures during the cooling season, may be pumped directly to air washers or cooling coils. Control is usually effected through control valves on the water supply to the cooling unit actuated by temperature or humidity controllers, or both, located either at the outlet of the conditioner or in the conditioned space.

APPLICATION OF CONTROLLING DEVICES TO A TYPICAL SYSTEM

Fig. 1 shows the location of controlling devices for a year 'round air conditioning system such as shown in Chapter 20, Fig. 3. A control diagram for pneumatic control equipment is shown in Fig. 2 for convenience in explaining the function and sequence of operation of the control in Fig. 1, but obviously the individual controls may also be of self-contained or electrically operated type provided they obtain the same control of valves and dampers.

The auxiliary controlling devices indicated may all be mounted together on an instrument board which may also contain a framed copy of the control diagram and the description of the automatic control cycle. Air gages, identified by suitable inscription plates, may be installed in the branch connections to and from the auxiliary devices on the instrument board to indicate the functioning of the various devices.

A description of the automatic control cycle follows:

When the fan motor is stopped, solenoid air valve E-1, actuated from the fan motor circuit, is de-energized and exhausts its branch to close minimum outdoor damper D-1, reposition three-way air valve V-a to close heating coil valve V-1, and remove main air from remote bulb indicator dew-point thermostat T-1, thereby closing maximum outdoor damper D-2 and opening return air damper D-3.

When the fan motor is started, E-1 fills its branch, thereby opening minimum outdoor damper D-1, repositions V-a to permit T-3 and T-4 to control V-1 and supplies main air to T-1 to permit same to operate D-2 and D-3.

During the summer cooling season, manual indexing switch S-1 is positioned to fill its branch, whereby it supplies main air to dew-point thermostat T-2, positions three-way air valves V-e and V-f to permit T-3 to control face damper D-4 and by-pass damper D-5, positions three-way air valve V-b to close heating coil valve V-1, positions three-way air valveV-c to remove the control of maximum outdoor damper D-2 and return air damper D-3 from dew-point thermostat T-1 and place these dampers under the control of dew-point thermostat T-2.

When S-1 is positioned as noted, remote bulb dew-point thermostat T-2 functions on a rising temperature to first gradually open maximum outdoor damper D-2 while simultaneously closing return air damper D-3 and on a further slight temperature rise, gradually positions chilled water valve V-2 to pass chilled water to the dehumidifier. The reverse operating sequence occurs on a falling dew-point temperature. Should the outdoor wet-bulb temperature rise above the desired indoor wet-bulb temperature, positive acting outdoor wet-bulb thermostat T-5, positions three-way air valve V-d to remove dampers D-2 and D-3 from the control of T-2, thereby closing outdoor damper D-2 and opening return air damper D-3.

On a rising return air temperature T-3 functions to open face damper D-4, while simultaneously closing by-pass damper D-5 as required to maintain the desired return air temperature. The reverse operation occurs on a falling return air temperature

During all seasons except the summer cooling season, manual switch S-1 is positioned to exhaust its branch, thereby making dew-point thermostat T-2 inoperative and positions chilled water valve V-2 for continuous recirculation, positions three-way valve V-b to permit heating coil valve V-1 to be operated as required, positions three-way valve V-c to permit T-1 to control D-2 and D-3, positions three-way valve V-e to permit T-3 to operate through low limit discharge thermostat T-4 to control heater valve V-1 and positions three-way valve V-f to open normally closed face damper D-4 and close normally open by-pass damper D-5.

When switch S-1 is positioned as noted, remote bulb dew-point thermostat T-1 functions on a rising temperature to gradually open maximum outdoor damper D-2, while simultaneously closing return air damper D-3 as required to maintain the desired dew-point temperature. The reverse operation occurs on a falling dew-point temperature. When the outdoor wet-bulb temperature rises above the desired dew-point temperature, the dew-point temperature will rise accordingly until such time as the system is indexed for summer cooling and chilled water is made available to drop the dew-point temperature.

Return air thermostat T-3 functions on a rising temperature to pass air through low limit discharge thermostat T-4 to gradually close heating coil valve V-1. Should the discharge temperature fall below the operating point of T-4, this thermostat will release air from its branch to gradually open V-1 as required to maintain the desired low limit discharge temperature regardless of the operation of T-3.

CHAPTER 34

Instruments and Test Methods

Temperature Measurement, Pressure Measurement, Measurement of Air Movement, Air Change Measurements, Measurement of Relative Humidity, Dust Determination, Heat Transfer Through Building Materials, Measurement of Heat Exchange for Comfort Conditions, Combustion Analysis, Smoke Density Measurements, Carbon Monoxide Measurements

THIS chapter presents a description of many test instruments used for heating, ventilating and air conditioning tests and presents a discussion of their use.

TEMPERATURE MEASUREMENT

Changes in the intensity of heat may be determined by several methods* such as measuring the change in volume of a liquid, the change in internal pressure of a confined gas, the current set up between dissimilar metals joined in a circuit, or the change in resistance of an electrical circuit.

Thermometers

The most common method used is the change in volume of a liquid such as mercury or alcohol enclosed in glass. Mercurial thermometers may be used for measuring temperatures from -40 F to approximately 1000 F. The lower limit is set by the freezing point of mercury. Since the boiling point of mercury is only about 675 F, the space above the mercury in thermometers designed for higher temperatures must be filled with an inert gas under pressure. Alcohol thermometers may be used for temperatures from -94 F to $+248$ F.

The more accurate thermometers are individually calibrated and have divisions etched on the stem. The two most common reference points are the freezing and boiling points of water. On the Fahrenheit scale, which is most commonly used in engineering work, there are 180 divisions between these points. On the Centigrade scale which is used by chemists and physicists, there are 100 divisions in this range. The temperature in degrees Fahrenheit equals $^9/_5$ of the temperature in degrees Centigrade, plus 32.

For permanent installations, glass thermometers are often protected by metal jackets and equipped with metal scales. Due to the heat capacity and heat conductance of the jacket, it is more difficult to obtain the true temperature at a point with these than with the exposed etched stem type. The latter is usually preferred for test purposes. Where used to measure temperatures in a duct, it may be inserted through a cork or rubber plug. Care must be taken to locate the bulb at the point where the temperature is desired and in many cases several must be used to get a correct average.

The probable error in etched stem thermometers is plus or minus one scale division, which makes calibration after manufacture necessary for most test work. Mercury thermometers are usually calibrated for complete stem immersion.

When incompletely immersed, a stem correction should be made for the

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*For a comprehensive treatment of temperature measurement the reader is referred to Temperature, Its Measurement and Control in Science and Industry, a symposium sponsored by the *American Institute of Physics* and published by Reinhold Publishing Corp.

most accurate determination. At ordinary atmospheric temperatures the correction is negligibly small, but it usually is important when measuring high temperatures such as those of steam and flue gas. The emergent stem correction may be calculated by the equation:

$$K = 0.00009\, D\, (t_1 - t_2) \quad (1)$$

where

K = correction to be added, degrees Fahrenheit.
D = number of degrees on the thermometer scale which are not immersed.
t_1 = temperature indicated on the thermometer, degrees Fahrenheit.
t_2 = temperature of the non-immersed mercury column, degrees Fahrenheit.
0.00009 = difference in the coefficient of expansion of the mercury and glass.

Since the bulb has considerable area, radiant energy may affect temperature readings[1]. In measuring room temperatures, care must be taken to locate thermometers away from hot surfaces such as radiators or cold surfaces such as walls or windows. Where this is impracticable, shields should be used to screen the bulb from the radiant energy.

Thermocouple

When two dissimilar metals are joined at two points and a temperature difference exists between these junctions, an electromotive force will be developed. Its magnitude depends upon the metals used and the temperature difference of the two junctions. Often the cold junction is kept at 32 F by immersion in an ice bath. In other instances, a higher temperature such as that of the atmosphere is used for this junction. By proper selection of metals, any temperature up to 2900 F may be measured. Readings are obtained by means of a potentiometer or sensitive galvanometer which may be calibrated directly in degrees. A potentiometer balances the electromotive force against a known electromotive force with no current flowing, hence this method is independent of length and variations in resistance of leads. Calibration of thermocouples for high temperatures may be made against known melting points of metals. Radiation effects may be minimized by using the smallest size of wires consistent with mechanical strength. The use of small wires also makes the thermocouple sensitive to minute fluctuations in temperature.

Other advantages of thermocouples are: they are readable at remote points; they may be made recording; and an average temperature may be readily obtained by connecting several couples in parallel.

The temperature of a surface is at best difficult to obtain accurately.[2] In most cases the thermocouple is most adaptable for this purpose. In a metal surface, a common method is to peen the couple into a small drilled hole, bearing in mind that the temperature indicated is that existing at the last point of junction in the couple. Other methods involve fastening the couple to the surface with adhesive cellophane, or cementing the couple with litharge in a surface scratch, and grinding it flush with the surface. In any of these methods the leads should be of as fine wire as practicable, since conductance along the leads to the couple may be a source of considerable error.

Resistance thermometers depend for their operation upon the change of resistance of wire with change in temperature. Their use largely parallels

[1] Errors in the Measurement of the Temperature of Flue Gases, by P. Nicholls and W. E. Rice (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 473).

[2] Measurement of Surface Temperatures, by F. C. Houghten and H. T. Olson, In Temperature, Its Measurement and Control in Science and Industry (Reinhold Publishing Corp.).

that of thermocouples, although readings tend to be unstable above 950 F. Two-lead temperature elements are not recommended, since they do not permit correction for lead resistance. Three leads to each resistor are necessary to obtain consistent readings.

For measuring high temperatures, such as in furnaces, *pyrometers* are often used. Radiation pyrometers concentrate the radiant energy on a thermopile, and the reading is obtained on a galvanometer or potentiometer. Optical pyrometers match a narrow spectral band, usually red, emitted by the object with that from a standard electric lamp supplied with electric current.

Thermometer Wells

Where temperatures of fluids or gases in vessels or conduits must be read, it is often necessary to resort to thermometer wells. These are especially designed to contain thermometers and thermocouples. Since they separate the temperature sensitive element from the medium to be observed, large errors may result from their improper use.[3] These errors may be due both to poor heat transfer from the wall of the well to the sensitive element, and also to the tendency of heat to travel along the length of the well itself. To improve heat transfer in the case of thermometers, the void should be filled with a liquid of minimum practical viscosity. At the well mouth, the stem should be packed to check evaporation and heat transfer to the atmosphere. To reduce the temperature gradient over the length of the well, the wall of the conduit should be carefully insulated in the area surrounding the observation point.

PRESSURE MEASUREMENT

Barometer

The most accurate barometer for determining the atmospheric pressure is the mercurial type, consisting of a tube over 30 in. long closed at the top and standing in a mercury well. The barometric pressure is expressed as the height of the mercury column above the level of the mercury in the well. Such barometers are equipped with an adjustment to compensate for change in level of mercury in the well. The reading should be taken at the top of the meniscus and is obtained on a vernier scale.

Correction for variation of the density of the mercury column and for expansion of the brass scale, which are usually calibrated for 32 F mercury and 62 F scale temperature, should be made by subtracting from the observed height in inches the value of C determined by Equation 2.

$$C = \frac{h\,(t - 28.630)}{(1.1123\,t - 10978)} \qquad (2)$$

where

C = correction to be subtracted, inches of mercury.
h = observed height, inches of mercury.
t = observed temperature of the barometer, degrees Fahrenheit.

Standard atmospheric pressure at sea level is 29.921 in. Hg. Since normal atmospheric pressure decreases about 0.01 in. Hg for each 10 ft increase in elevation, it is important to make a correction if the elevation of the barometer is not that of the test apparatus. In many cases the barometric reading may be obtained from a nearby weather bureau station. Inquiry should be made as to whether the value is as observed or corrected to sea level.

[3] *A.S.M E.* Power Test Code, Instruments, and Apparatus, Part 3

Atmospheric pressure may also be measured by an aneroid barometer which is easily portable. In this type, variations in atmospheric pressure bend the thin surface of a box or tube which contains a reduced pressure. The aneroid type is not as accurate as the mercurial and needs frequent calibration against one of the latter type. Most of the pressure gages used in engineering work indicate the difference between the pressure being measured and the atmospheric pressure. Pressures as measured are called gage pressures. Absolute pressure may be obtained by adding barometric pressure and gage pressure algebraically.

Pressure Gages

The Bourdon type gage is a widely used device for measuring pressures. The Bourdon tube is elliptical in cross-section and circular in form, and is connected by suitable linkage to a hand which moves over a dial. An increase in pressure tends to straighten the tube and a decrease has the opposite effect. When used with high temperature steam, the tube must be protected by a water seal. When used with ammonia it must be made of steel or other material not attacked by this substance. When used for sub-atmospheric pressure, the gage is known as a vacuum gage, and is usually graduated in inches of mercury. For pressures above atmospheric, it is termed a pressure gage and is graduated in pounds per square inch. Some are made to read in both directions and are termed compound gages. Calibration is usually made with a dead weight tester, consisting of a platform and weights resting on a piston floating on oil. From the area of the piston and the total weight resting on the oil, the pressure at all points in the fluid is determined. Adjustments are provided in the gage linkage to make necessary corrections. A correction chart may also be made and used for accurate work.

For comparatively low gage pressures or differences in pressure between two points in a duct system, the vertical *U* tube is a simple and accurate gage and is often used for test work with various fluids such as mercury, water, kerosene, or alcohol. Readings may be in inches of any of these fluids.

For measuring pressure differences of a few inches of water, or less, *U* gages are often made sloping for greater magnification of scale. In commercial gages of this type, commonly termed *draft gages*, only one tube of small bore is used and the other leg is replaced by a reservoir. Although the scale is calibrated to read in inches of water, a fluid having the density and characteristics of kerosene is often used. It is important, of course, to use a fluid having the same gravity as that for which the gage was originally calibrated, or to use a correction chart with some other fluid. Such gages may be checked one against another to detect errors in gravity of fluid. For more accurate calibration the gage may be checked against a micromanometer or a calibrating device known as a hook gage[4]. The accuracy of a draft gage is dependent on the slope of the tubes and consequently the base of the gage must be leveled carefully. It is not desirable to use a slope of less than 1 in 10.

For measuring low pressure differences to within 0.001 in. of water very sensitive *micromanometers* are available, such as the Illinois or Wahlen, the Askania, and the Emswiler[5,6]. Calibration of these is impossible,

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[4]Standard Test Code for Centrifugal and Axial Fans, Edition of 1938. See also Standard Code for the Testing of Centrifugal and Disc Fans (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 407; Vol. 37, 1931, p. 363).

[5]Illinois Micromanometer (*University of Illinois, Engineering Experiment Station Bulletin* No 120, p 91).

[6]The Weathertightness of Rolled Steel Windows, by J. E. Emswiler and W. C. Randall (A.S H V E. TRANSACTIONS, Vol. 34, 1928, p. 527).

and readings are converted to pressure units by fundamental calculations involving the specific gravity of the fluids used and the design principles involved.

The type of *pressure tip* used and its location, in cases of fluid flow, may be fully as important as the accuracy of the gage to which it is connected. In the case of low pressure air flow, the Pitot tube pointed upstream with connection made to its static pressure element is probably the best. Care should be taken to avoid taking readings where pressure can be affected by impact or eddies in the air stream.

MEASUREMENT OF AIR MOVEMENT

The problem of measuring air movement may be divided into three main parts: when confined in ducts, when circulating in free spaces, and when entering or leaving such space through openings such as grilles. Other gases might be measured by the same methods, but emphasis here will be on air measurements[7].

For determining the velocity, and therefore the volume of air flowing in a duct, such as in the test of a fan or a complete ventilating system, the Pitot tube as described in the A.S.H.V.E. Code[8] is probably most often used. At low velocities the velocity pressure head is so low that it becomes difficult to get accurate gage readings. The velocities used in many ducts are below the lower limit of determination with gages available. The relation between velocity and velocity pressure may be used to determine the range of gage required.

$$V = 1096.5 \sqrt{\frac{h_V}{d}} \qquad (3)$$

where

V = velocity, feet per minute.
h_V = velocity pressure, inches of water.
d = density of air, pounds per cubic foot.

Air flow in a round duct is seldom uniform. In general, the velocity is lowest near the edges, and maximum at or near the center. In order to obtain higher velocities and more uniform flow across the measuring section, it is sometimes possible to reduce the duct to a smaller cross-section at the Pitot station by use of a long transition piece. In any case, a large number of readings along two diameters should be taken, with 20 being quite desirable. These should be taken at the centers of equal annular areas for correct determination of volumes[9]. For small pipes it is sometimes necessary to construct a Pitot tube smaller than the standard size. Such a small Pitot tube should be geometrically similar to the standard tube. Pulsating or disturbed flow will give erroneous results and every effort should be made to remove disturbances in the Pitot tube section.

Many forms of Pitot tubes other than the one described have been used and calibrated[10]. A double-ended tube[11], one end pointing down-stream,

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[7]For technical data refer to Fluid Meter Reports, Parts 1—1937, 2—1931, and 3—1933 (*American Society of Mechanical Engineers*).

[8]Loc. Cit. Note 4

[9]Loc. Cit. Note 4.

[10]Technical Notes No. 546 (*National Advisory Committee for Aeronautics*, November, 1935).

[11]The Characteristics of Double Pitot Tubes, by F. R. Ingram, E. Diez-Canseco and L. Silverman (A.S H V E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November, 1942, p. 708).

and one up-stream, is sometimes used for low velocities, but it should be carefully calibrated for accurate results. A special form of this tube design consists of two straight ⅛ in. tubes soldered together, closed at the end, and with a 0.04 in. hole in each tube opposite the line of contact. This tube is useful in exploring velocities on exhaust inlets, such as on hoods placed around grinding wheels.

The rounded approach orifice or nozzle of the general type described in the A.S.H.V.E. Unit Heater[12] and Unit Ventilator[13] Codes is an accurate air measuring device. When it is well made, the coefficient closely approaches unity. The discharge from such a nozzle is uniform[14] and provides a good location for calibration of air velocity instruments[15].

The Venturi meter is like the nozzle except for the addition of a downstream transition section that reduces the pressure drop through the measuring apparatus.

The thin-plate square-edged orifice has a decided advantage over the nozzle and Venturi meter in cost. Its coefficient is approximately 0.60. The exact value depends on the location of the connections, the pressure drop, the diameter ratio of orifice to pipe, and the sharpness of the edge[16].

Another method of air measurement uses the thermal electric principle where by means of a measured amount of current, heat is put into the air stream. The temperature rise is measured, and with the specific heat of the air mixture known, the weight of air flowing may be calculated. Heat should be applied uniformly to the mass of air passing, and the small temperature difference must be determined accurately.

Air Currents in Free Spaces

One of the instruments useful in determining the velocity of air currents in free spaces is the Kata-thermometer. It is essentially an alcohol thermometer with a large bulb. The stem has two marks, one corresponding to 95 F, and the other 100 F. The instrument is heated in water above 100 F, then dried and the time in seconds required for it to cool from 100 to 95 when placed in the air current gives a measure of the non-directional velocity. It is important to wipe the Kata-thermometer dry before taking the reading. Each Kata has its own factor etched on the stem, and this factor must be used with its cooling formula or chart for obtaining the velocity. The Kata-thermometer is useful in exploring ventilated spaces to determine whether the proper air movement and distribution are being maintained. It is also used in determining the cooling power of the atmosphere, since it loses heat by radiation and convection when dry, and by radiation, convection, and evaporation when the bulb is equipped with a wetted cloth covering[17].

Another instrument for measuring low velocity air currents is the heated thermometer anemometer[18]. This consists of an ordinary mer-

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[12]Standard Code for Testing and Rating Steam Unit Heaters (A S.H V E. TRANSACTIONS, Vol. 36, 1930, p. 165), adapted January, 1930 by A.S.H.V.E.

[13]A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 25), adopted June, 1932.

[14]Discharge Coefficients of Square Edged Orifices for Measuring the Flow of Air, by H. S. Bean, E. Buckingham and P. S. Murphy (*Bureau of Standards Journal of Research*, Vol. 2, 1929, p. 561).

[15]A.S.H.V.E. RESEARCH REPORT No. 1140—The Use of Air Velocity Meters, by G. L. Tuve, D. K. Wright, Jr. and L. J. Seigel (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 645).

[16]Flow Measurement by Nozzles and Orifice Plates (*A.S.M.E.* Power Test Codes, Chapter 4 of Part 5, 1940).

[17]Temperature, Humidity and Air Motion Effects in Ventilation, by O. W. Armspach and Margaret Ingels (A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 103).

[18]The Heated Thermometer Anemometer, by C. P. Yaglou (*Journal Industrial Hygiene and Toxicology*, Vol. 20, October, 1938, No. 8).

curial glass thermometer with a resistance winding on the bulb. Current is supplied from an external source in a measured amount. The temperature rise shown on this heated thermometer over that shown by an ordinary thermometer at the same location, and the current supplied, make it possible to calculate the non-directional velocity of the air stream. Since a smaller bulb is used than that on the Kata-thermometer, it is less affected by radiant heat sources.

The heated thermocouple anemometer employs a thermocouple instead of a thermometer[19].

Another instrument is the hot wire anemometer which has been made in several patterns. In general, a measured current is supplied to raise the temperature of a fine bare wire above the temperature of the surrounding air. With the use of a very fine wire, minute fluctuations in velocity may be measured, and the area exposed to radiant exchange with heated or cooled surfaces is at a minimum. This instrument is easily adapted to remote reading or recording. A group of them may be connected together to give the average velocity in a space, or the velocity at individual points within a test space, by suitable switching arrangements[20,21].

Deflecting Vane Anemometer

The deflecting vane anemometer consists of a pivoted vane enclosed in a case, against which air exerts a pressure as it passes through the instrument from an up-stream to a down-stream opening. The movement of the vane is resisted by a hair spring and a damping magnet. The instrument gives instantaneous readings of directional velocities on an indicating scale. When used in fluctuating velocities, it is necessary to average visually the swings of the needle to obtain average velocities. This instrument is very useful for studying mixing of air in a room[22] and in locating and measuring peak velocities that may be objectionable. Various attachments are available, such as the double tube arrangement for obtaining velocities in ducts, and a device for measuring static pressures. Each instrument and the attachments for it must receive individual calibration.

Propeller or Revolving Vane Anemometer

The propeller or revolving vane anemometer consists of a light revolving wheel connected through a gear train to a set of recording dials that read the linear feet of air passing in a measured length of time. It is made in various sizes, 3 in., 4 in., and 6 in. being most common. Each instrument requires individual calibration. At low velocities the friction drag of the mechanism is considerable. In order to compensate for this, a gear train that overspeeds is commonly used. For this reason the correction is often additive at the lower range and subtractive at the upper range with the least correction in the middle range of velocities. Most of these are not sensitive enough for use below 200 fpm.

Measurement of Velocities at Inlets and Outlets of Ducts

In the field it is often advisable to make volume measurements at the

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[19]A.S H.V.E. RESEARCH REPORT No. 1165—Development of Instruments for the Study of Air Distribution in Rooms, by A. P. Kratz, A. E. Hershey and R. B. Engdahl (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 351).

[20]Development of Testing Apparatus for Thermostats, by D. D. Wile (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 349).

[21]Linear Hot Wire Anemometer, Its Application to Technical Physics, by L. V. King (*Journal Franklin Institute*, 1916).

[22]A S H.V E. RESEARCH REPORT No. 1204—Entrainment and Jet-Pump Action of Air Streams, by G. L. Tuve, G. B Priester and D. K. Wright, Jr. (A.S H V.E. TRANSACTIONS, Vol 48, 1942, p. 241).

face of the supply openings. Often it is hard to get into the duct system, or it is difficult to find sections where the flow would be sufficiently uniform. For accuracy the instrument and its application should be checked on a similar approach and grille in the laboratory before use in the field.

Tests have shown that the propeller type anemometer can be used successfully on most of the common types of supply grilles[23, 24]. The core area is divided into equal squares, and the anemometer is held against the face of the grille for the same length of time in each. To get the air volume in cubic feet per minute, the average corrected velocity in feet per minute thus obtained is multiplied by the average of the gross and net free area of the grille (core) in square feet.

On exhaust openings, the anemometer traverse is made as described previously. The air volume may be determined by multiplying the corrected velocity in feet per minute by the gross core area of the grille in square feet and by a coefficient for average conditions of 0.85[25].

When a propeller type anemometer is held in a stream of varying velocities, it tends to indicate higher than the true average, that is, the speed of the propeller is nearer to the top velocity in its area than it is to the minimum velocity. This is the main reason for the large difference in ratings of unit ventilators by the anemometer method and by air volume measurements in a duct approach to the inlet[26].

Any of the other anemometers described can be used within their range at the face of supply grilles when properly applied. In principle it is a case of finding the velocity at many points and using the average thus found with the correct discharge area at that cross-section. The deflecting vane anemometer equipped with a jet on the end of a rubber tube has been found especially convenient and accurate on supply grilles[27]. On modern air conditioning grilles the core area is used without a correction coefficient when the jet is held one inch away from the face of the grille. At this distance the constriction due to the thin bars has disappeared since the small air jets have reunited, and the air stream has not yet spread beyond the core dimensions. With deflecting grilles the exploring jet should be turned to the angle giving a maximum reading. With suitable traversing tips and calibration, this instrument may also be used on exhaust grilles if proper grille factors are applied[28].

While hardly a quantitative instrument, smoke is very useful in studying air streams and currents. The application of a more accurate instrument is often made more exact by a preliminary exploration with smoke. A mixture of potassium chlorate and powdered sugar in equal portions gives a very satisfactory non-irritating smoke. It is fired by a match, and since considerable heat is evolved, it should be placed in a pan away from inflammable objects.

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[23]A.S.H.V.E. Research Reports Nos. 857, 911 and 966—Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies (A S.H.V.E. Transactions, Vol 36, 1930, p 201, Vol 37, 1931, p 619, and Vol. 39, 1933, p 373).

[24]A S.H.V.E. Research Report No 1162—Air Flow Measurements at Intake and Discharge Openings and Grilles, by G. L. Tuve and D K. Wright, Jr. (A.S H.V.E. Transactions, Vol. 46, 1940, p 313).

[25]A.S H.V.E. Research Report No. 1092—The Flow of Air Through Exhaust Grilles, by A M. Greene, Jr. and M. H. Dean (A.S.H.V.E Transactions, Vol 44, 1938, p 387).

[26]A S.H V.E. Research Report No 936—Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson and R. W. Kubasta (A.S.H.V E. Transactions, Vol. 38, 1932, p 463).

[27]A.S.H V.E. Research Report No 1076—Air Distribution From Side Wall Outlets, by D W. Nelson and D. J. Stewart (A.S.H.V.E. Transactions, Vol 44, 1938, p 77)

[28]Measuring Air Flow, by G. L. Tuve (*Heating, Piping and Air Conditioning*, December, 1941).

AIR CHANGE MEASUREMENTS

Atmospheric air contains a certain amount of carbon dioxide. Its concentration is increased within enclosures by the carbon dioxide given off by occupants. The total air change through open windows, infiltration, and mechanical ventilation, may be measured by the carbon dioxide concentration[29]. Since occupants also give off moisture, the increase in humidity may also be used as an index of ventilation within a space. Neither method is used at the present time, and more direct methods of measuring air supply and air distribution are in favor.

MEASUREMENT OF RELATIVE HUMIDITY

Wet- and dry-bulb mercurial thermometers are usually used to determine relative humidity. The sling psychrometer is a common mounting of the thermometers to permit swinging. The wet-bulb wick and water for wetting it must be clean, and the temperature of the water should preferably be slightly above the wet-bulb temperature. An air stream velocity of 900 fpm is recommended, although velocities from 300 fpm to 1000 fpm have been found satisfactory for passage over the wet-bulb wick. The velocity may be obtained by whirling the thermometer or by aspirating air over the wet-bulb. In ducts, the air flow itself gives the proper evaporating conditions. Several observations should be made until the minimum temperature is reached. Relative humidity may be obtained from tables or psychrometric charts[30]. Although it is common practice to use the charts which are based on a barometric pressure of 29.92 in. Hg, a correction for barometric pressure is necessary for extreme accuracy. This correction is made by multiplying the relative humidity as determined from the chart by the ratio of the observed barometric pressure and the standard barometric pressure.

For temperatures below 32 F, the water on the wick is allowed to freeze, during which time the temperature will drop below the true wet-bulb. A thin film of ice is more desirable than a thick one, and it is satisfactory to remove the wick and freeze a thin film directly on the bulb. Care must be taken to read the temperatures accurately due to the slight wet-bulb depressions. Tables for ice conditions must be used[31].

The dew-point apparatus for humidity measurements consists of a polished plated container cooled by the evaporation of a volatile liquid within. The temperature at which the first slight water vapor forms is the dew-point. If the temperature is below 32 F, the deposit will appear as frost. Another method of determining humidity is by chemical means in which the water vapor is removed by a drying agent and weighed on a chemical balance. A thermal conductivity method is available for temperatures above 212 F or for extremely low humidities[32].

DUST DETERMINATION

The measurement of dust is complicated by the many kinds involved. Some of the collecting methods are impingement on viscous surfaces,

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[29]A S H V.E. RESEARCH REPORT No. 959—Indices of Air Change and Air Distribution, by F. C. Houghten and J. L. Blackshaw (A S.H.V E. TRANSACTIONS, Vol. 39, 1933, p. 261).

[30]Psychrometric Tables for Vapor Pressure, Relative Humidity and Temperatures of the Dew-Point (*U. S. Department of Agriculture*, Weather Bureau, Washington, D. C.).

[31]A Review of Existing Psychrometric Data in Relation to Practical Engineering Problems, by W. H. Carrier and C O Mackey (*A S M E. Transactions*, January, 1937, p. 33; Discussion, *A S.M E. Transactions*, August, 1937, p 528).

[32]Gas Analysis by Measurement of Thermal Conductivity, by H. A. Daynes (*Cambridge Press*, 1933).

impingement at high velocity under water, collection on porous crucibles through which air passes, and electric precipitation. Determination may be by direct weighing of samples or by microscopic counting. The most commonly used methods are the modified Hill dust counter using microscopic count, the Smith-Greenburg impinger which collects samples in water and which are counted under a microscope in a Sedgwick cell[33], and the Lewis[34] sampling tube with the analytical determination of the increase in weight of a porous crucible. All reports should state the method of sampling and counting. The A.S.H.V.E. Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work specifies the porous crucible method[35].

HEAT TRANSFER THROUGH BUILDING MATERIALS

The A.S.H.V.E. Standard Test Code for Heat Transmission Through Walls[36] describes the construction and use of the guarded hot box for determining over-all heat transmission coefficients of built-up sections.

In June, 1942, the A.S.H.V.E. adopted a standard test procedure for determining the conductivity of materials by the use of a guarded hot plate[37]. The Nicholls heat meter is very useful for determining the heat flow through walls of buildings[38].

MEASUREMENT OF HEAT EXCHANGE FOR COMFORT CONDITIONS

Several instruments have been devised to measure the effect of various factors as they relate to the comfort of the body[39]. The principal ones are the Kata-thermometer, Dufton's eupatheoscope, Vernon's globe thermometer, Winslow and Greenburg's thermo-integrator, and Yaglou's heated globe[40, 41]. These instruments were designed to obtain a quantitative measurement of the thermal exchange between the human body and its environment.

COMBUSTION ANALYSIS

The analysis of flue gases to determine completeness and efficiency of combustion is usually made chemically with the Orsat apparatus. This consists of a measuring burette, a leveling bottle, and three pipettes. Carbon dioxide is absorbed in the first pipette by potassium hydroxide, oxygen in the second by potassium pyrogallate, and carbon monoxide in the third by cuprous chloride. A known volume of gas is drawn in, and after each of the three absorptions the reduced volume is again measured in the burette. Pressure and temperature of the gas sample are kept

[33]Public Health Bulletin, No. 144, 1925, (*U. S. Public Health Service*).

[34]Testing and Rating of Air Cleaning Devices Used for General Ventilation Work, by S. R. Lewis (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 277).

[35]A.S.H.V.E. Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work, adopted January, 1934 (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 225).

[36]A.S.H.V.E. Standard Test Code for Heat Transmission Through Walls (A.S.H V.E TRANSACTIONS, Vol. 34, 1928, p. 253), adopted 1928.

[37]Standard Method of Test for Thermal Conductivity of Materials by Means of the Guarded Hot Plate (Tentative), adopted July, 1942, by A.S.H.V.E.

[38]A.S.H.V.E. RESEARCH REPORT No. 685—Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 65).

[39]Measurement of the Physical Properties of the Thermal Environment, by D. W. Nelson, F. R. Bichowsky, L. M. K. Boelter, R. S. Dill, A. P. Gagge, John A. Goff, A. E. Hershey, F. C. McIntosh, F. W. Reichelderfer, G. L. Tuve and C. P. Yaglou (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June, 1942, p. 382).

[40]Instruments and Methods for Recording Thermal Factors Affecting Human Comfort, by C. P. Yaglou, A. P. Kratz and C.-E. A. Winslow (Year Book, *American Journal Public Health*, 36-37).

[41]The Thermo-Integrator—A New Instrument for the Observation of Thermal Interchanges, by C.-E. A. Winslow and Leonard Greenburg (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 149).

TABLE 1. RINGELMANN SMOKE CHART SPACINGS

NUMBER OF CARD	THICKNESS OF LINES, MM	DISTANCE IN CLEAR BETWEEN LINES, MM
1	1.0	9.0
2	2.3	7.7
3	3.7	6.3
4	5.5	4.5

constant while measuring. Several passes are made through each pipette which contains tubes or glass beads to increase the wetted surface. It is essential that each reaction be completed before the next reaction is started. Since the life of the reagents is limited, it is well to keep a record of the number of samples tested. Care is needed in operation to prevent the pulling of reagents out of the pipettes into the capillary tubing and burette. Many recording gas analyzers are available and are usually found in the larger plants.

Carbon Monoxide Measurement

A method of analyzing for low carbon monoxide concentrations completes the oxidation of the carbon monoxide in a known volume of sample, in the presence of a catalyst. The heat resulting is measured by a thermocouple calibrated in parts per 10,000 of carbon monoxide[42].

SMOKE DENSITY MEASUREMENTS

Smoke density may be judged by assigning to it the number of the Ringelmann Smoke Chart which appears to have the same color when observed at a distance of 50 ft. The charts are numbered 1 to 4 and are made of black lines cross-ruled on white as given in Table 1.

Apparatus using the photo-electric cell has been devised for recording smoke densities in large plants.

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[42]A Carbon Monoxide Recorder, by S. H. Katz, D. A. Reynolds, H. W. Frevert and J. J. Bloomfield (*U. S. Bureau of Mines, Technical Paper* No. 355, 1926).

CHAPTER 35

Motors and Motor Controls

Motor Rating; Functions of Motor Control Equipment; Direct Current Motors, Types, Control Equipment and Specifications; Alternating Current Motors, Types, Control Equipment and Specifications; Gear Motors; Glossary of Motor Terms, Enclosures, Speed Classification and Mounting

THE electric motor, available in many different types suitable for various services, is now the most widely used form of prime mover. The equipment for starting, controlling and protecting these motors varies with the type and with the functions it is desired to attain. Motors are divided into two general classifications, *alternating-current* or *direct-current*, depending on the power source to be used.

In selecting a motor for a particular application consideration must first be given to the type of power supply available. All machinery has different load characteristics which may vary with speed. Certain types may have a constant torque over wide ranges of speed, while others may have changing torques with changing speed. Consideration should be given to selecting the motor and the motor control which best suit the requirements of the drive.

MOTOR RATING

The rating of an electric motor depends upon the total temperature which the motor attains under operating conditions. This total temperature depends on both the ambient temperature and the temperature rise of the motor. As motor temperature rise is in turn determined by the ability of the motor to dissipate heat, circulation to the motor should not be restricted. Improper selection of motors with regard to temperature ratings may result in high motor operating temperatures with accompanying reduction in motor life.

In general, the electrical insulation is the portion of the motor most susceptible to injury from high operating temperatures. Of the several types of insulation which are available, the most common type, specified as Class A by the *National Electrical Manufacturers Association*, consists of cotton, felt, paper or similar organic materials and permits a 55 C rise in temperature over a 40 C ambient temperature. Class B insulation consists of mica, asbestos, fiber glass, or similar inorganic materials and permits a 75 C rise in temperature over the 40 C ambient temperature. Other types of insulation such as silicone resin are available and permit extremely high operating temperatures.

The mechanical construction of the different types of motor enclosures and the rise in temperature with Class A insulation for each type are enumerated in the glossary at the end of this chapter. Since the difference in temperature between the hottest spot and the nominal temperature, as measured by a thermometer, is greater for a completely unprotected machine than it is for an enclosed machine, the permissible temperature rise is smaller for an open motor.

FUNCTIONS OF CONTROL EQUIPMENT FOR MOTORS

In general, control equipment for all types of motors should provide: (1) means of disconnecting the motor from the power supply; (2) means

for starting the motor; (3) overload protection for the motor; (4) protection against low voltage; and (5) means for varying the motor speed.

Full voltage starting for motors is preferable because of its lower first cost and simplicity of control. Except for d-c machines, most motors are mechanically and electrically designed for full voltage starting. The starting inrush current, however, is limited in many cases by regulations of power companies because of the voltage fluctuations which may be caused by heavy current surges. It is therefore often necessary to reduce the starting current below that obtained by across-the-line starting. The power supplier should be consulted to determine the allowable inrush current for any given location.

The choice between full voltage and reduced voltage starting is governed almost entirely by inrush current limitations. The starting torque of all motors varies with the starting current and it is therefore necessary to insure that the motor is supplied with sufficient current to develop enough torque to accelerate the load.

In present practice overload protection of motors is obtained by use of thermal overload inverse time limit type protection. The usual setting of such protection devices is at 125 per cent overload, the element tripping after a definite interval of time. The *NEMA* standard practice requires the use of fuses to protect the overload elements from severe short circuit currents.

Two types of protection are available against low voltage at the motor terminals. One type, called low voltage release, permits the motor line contactor to drop out on low voltage and to close again when the voltage returns to normal, thereby restarting the motor when the abnormal condition is ended. The second type, called low voltage protection, causes the motor line contactor to drop out on low voltage but prevents restarting when the voltage returns to normal except by the action of an operator. This latter type of protection is desirable where it is necessary for the operator to make initial starting adjustments on the machine.

Manual control for an alternating or a direct current motor is usually located near the motor. When so located an operator must be present to start and stop or change the speed of the motor by operating the control mechanism. Manual control is sometimes employed only as a device to give overload protection and another device is employed to start and stop the motor. Manual control is used particularly on small motors which operate unit heaters, small blowers, and room coolers in an air conditioning system. In other cases manual control in the form of drums, when used with multi-speed motors, is only used as a speed setting device while the starting and stopping functions operate automatically through thermostats and pressure switches.

Because of the increasing complexity of air conditioning systems, heating, ventilating and air conditioning equipment is operated preferably by automatic control and less dependence is placed on manual operation and regulation.

Automatic control of motor starters may be accomplished by the use of remote push button stations, by a thermostat, float switch, pressure regulator, or other similar pilot devices. An added advantage of automatic control is that the main wiring for the starter may be installed near the motor, while the starter may be operated by a control device located elsewhere.

TABLE 1. CLASSIFICATION OF MOTORS

Power Supply	Type	Speed Characteristics	Full Voltage		Hp Range	Type of Application See Footnote†
			Starting Torque	Starting Current		
Constant Speed Drives						
D-C	1 Shunt	Constant	Normal	Normal (with controller)	All	(a) Fans and (c) centrifugal pumps and centrifugal compressors
	2. Compound	Variable	High	Normal (with controller)	All	(b) (c) (e) Reciprocating pumps and frequent or hand starting
	3. Series	Variable	High	Normal (with controller)	Small	(d) Fans direct connected
Poly-phase A-C	4 Squirrel-Cage General Purpose Class A	Constant	Normal 0 8–1.5 times	High 6–8 times	All	(a) Fans and (c) centrifugal pumps and centrifugal compressors
	5 Squirrel-Cage Class B	Constant	Normal 0 8–1 5 times	Normal 5–6 times	Medium Small	(a) Fans and centrifugal pumps and centrifugal compressors
	6 Squirrel-Cage Class C	Constant	High 2–2.6 times	Normal 5–6 times	Medium Small	(b) Reciprocating pumps (e) and compressors started loaded
	7 Wound Rotor	Constant or Variable	High 1–2.5 times	Low 1–3 times with secondary control	All	(a) Hoists (b) reciprocating pumps and compressors (c) and frequent (e) or hand start
	8. Synchronous High Speed	Exactly Constant	Normal 0 75–1.75 times	Normal 5–7 times	Medium Large	(a) Fans and centrifugal pumps and centrifugal compressors
	9 Synchronous Low Speed	Exactly Constant	Low 0 3–0 4 times	Low 3–4 times	Medium Large	(a) Reciprocating compressors starting unloaded
Single phase A-C	10 Capacitor	Constant	High	Normal	Small	(b) Pumps and compressors
	11 Capacitor Fan	Constant	Normal	Normal	Small	(a) Fans, centrifugal pumps
	12 Capacitor Start Induction Run	Constant	High	Normal	Small Fractional	(a) Fans (b) pumps and compressors
	13. Repulsion Induction	Constant	High	Normal	Medium Small	(a) Fans (b) pumps and compressors
	14 Split Phase	Constant and Adjustable	Normal	Normal	Fractional	(a) Fans (b) pumps and compressors (d) fans—direct

†Applications:

a. Drives having medium or low starting torque and inertia (WR^2) such as fans and centrifugal pumps or reciprocating pumps and compressors started unloaded.

b Drives having high starting torques, such as reciprocating pumps and compressors started loaded

c. Similar to (*a*) except where frequent or hand starting (large WR^2) requires a higher starting and accelerating torque.

d Fans direct connected.

e Stoker drives.

TABLE 1. CLASSIFICATION OF MOTORS—(Concluded)

Power Supply	Type	Speed Characteristics	Full Voltage		Hp Range	Type of Application See Footnote†
			Starting Torque	Starting Current		
			Adjustable Speed Drives			
D-C	15. Shunt Field Adjustment	Constant	Normal	Normal (with controller)	All	(a) (e) Fans and centrifugal pumps
	16. Armature Resistance Adjustment	Variable	Normal	Normal (with controller)	All	(a) (e) Fans and centrifugal pumps
	17. Variable Voltage Control	Constant	Normal	Normal	All	(d) Fans and centrifugal pumps
Poly-phase A-C	18. Squirrel-Cage High Slip, Transformer Adjustment	Variable	Normal	Normal	Medium Small	(a) Fans
	19. Squirrel-Cage Separate Winding or Regrouped Poles	Constant Multi-Speed	Normal or High	Normal or Low	All	(a) Fans (b) pumps and (c) compressors
	20. Wound Rotor, External Secondary Resistance	Variable	High	Low	All	(a) Fans (b) centrifugal pumps and compressors
Single Phase A-C	21 Repulsion	Variable	High	Normal	Low and Fractional	(a) Fans—centrifugal pumps (b) compressors
	22. Capacitor Low Torque Tapped Winding	Variable Two Speed	Low	Normal	Fractional	(d) Fans, direct
	23 Capacitor Low Torque Transformer Adjustment	Variable	Low	Low	Fractional	(d) Fans
	24. Split Phase Regrouped Poles	Constant	Normal	Normal	Fractional	(d) Fans

DIRECT CURRENT MOTORS

Direct current motors are classified (see Table 1) under three headings, depending on the type of winding: *shunt wound, compound wound,* and *series wound.*

Shunt Wound motors, being suitable for application to fans, centrifugal pumps, or similar equipment where the amount of starting torque required is relatively small, are used for the majority of direct current applications in the field of heating, ventilating, and air conditioning. They may be used on reciprocating pumps and compressors if started under unloaded conditions.

Without auxiliary control the shunt wound motor is designated as *constant speed.** Fig. 1 illustrates the characteristics of direct current motors, showing speed, horsepower, and torque as a function of current. The speed regulation* of small size shunt wound motors from ¾ hp to 5 hp is 12 per cent as specified by the *NEMA* while on larger motors it is 10 per cent.

*Refer to Glossary at end of chapter.

Compound Wound motors are required for application to reciprocating compressors, stokers, reciprocating pumps when started under loaded conditions, and other similar equipment requiring high starting torque. The characteristics of this type of motor are such that for starting torques above full-load torque the starting current required is somewhat less than for the shunt wound motor. Compound wound direct current motors are normally used whenever frequent starting makes high starting and accelerating torque desirable. Without auxiliary control, compound wound motors are designated as *varying speed*,* and have a speed regulation of 25 per cent.

Series Wound motors find only limited application in a few special cases and are available in a limited range of sizes. The motors are used

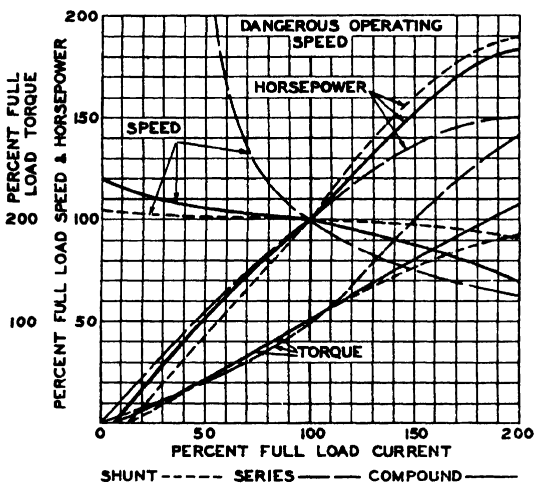


FIG 1. CHARACTERISTICS OF DIRECT CURRENT MOTORS

where extremely high starting torques are required and must be applied only to direct coupled continuous loads due to the fact that the speed of the motor becomes dangerously high when the motor is operated at a light load.

Typical d-c motor specifications are shown on page 634.

DIRECT CURRENT MOTOR CONTROL

Direct current motors are always started through starting controllers which use a resistance, in series with the motor armature, which is gradually cut out as the motor comes up to speed. Motors of small ratings may be linestarted providing the inrush current causes no serious voltage fluctuations in the power supply line.

Constant Speed and *Varying Speed* motors in ratings up to 2 hp may be linestarted. As shown in Fig. 2, the recommended practice for manual starting of motors over ½ hp requires the use of a disconnecting switch

*Refer to Glossary at end of chapter.

and a face-plate type starter. For automatic push button starting a disconnecting switch and an automatic starter are recommended.

Adjustable Speed motors are normally shunt wound and are operated at various speeds by varying a resistance connected in series with the motor field. A maximum range of speed of about 5 to 1 can be obtained by this means. Rated speed regulation is 22 per cent for motors of this type from 2 to 5 hp; 15 per cent is standard in larger sizes. The *NEMA* practice on rating adjustable speed d-c motors is defined in the Glossary at the end of the chapter.

The control for the adjustable speed motor consists of the addition of

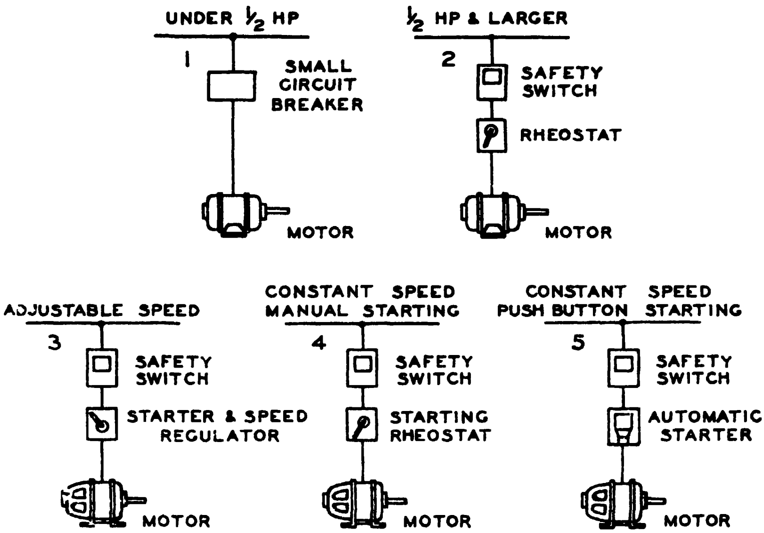


FIG. 2 RECOMMENDED CONTROLS FOR D-C MOTORS

a field rheostat for speed control to the equipment specified for the constant speed motor.

*Adjustable Varying Speed** motors are d-c motors in which the speed is varied by the addition of resistance in series with the armature. The speed of the motor by this means is always less than the rated full field speed and varies widely with a change in load, especially with high series resistance. Fig. 3 illustrates typical speed characteristics of this type of motor for different values of armature resistance.

The addition of series resistance in the armature circuit reduces the motor speed by lowering the voltage on the armature. At one-half speed the voltage is approximately one-half of line voltage. Consequently, with rated full load current the power delivered by the motor will be only one-half of the maximum, *e.g.*, it will be 5 hp from a 10 hp motor, because the other 5 hp will be lost in the resistance. It is, therefore, evident that the efficiency of the motor is reduced at reduced speeds due to the loss in the resistor.

Control for the adjustable varying speed motor is similar to that for

*Refer to Glossary at end of chapter.

Typical Specifications for D-C Motors

....................Hp, ...RPM, $\left\{\begin{matrix}\text{Constant}\\ \text{Adjustable}\\ \text{Varying}\end{matrix}\right\}$ Speed,

$\left\{\begin{matrix}\text{Shunt}\\ \text{Compound}\\ \text{Series}\end{matrix}\right\}$ Wound, $\left\{\begin{matrix}115\\ 230\\ 600\end{matrix}\right\}$ Volts D-C Motors

For Driving .... Motor Shall
(Application)

Be Arranged For $\left\{\begin{matrix}\text{Horizontal}\\ \text{Vertical}\end{matrix}\right\}$ Mounting And Shall

Be Provided With A..
(Open, Splashproof, etc.)

Type Of Enclosure, *NEMA* $\left\{\begin{matrix}\text{Class A}\\ \text{Class B}\end{matrix}\right\}$ Insulation,

And $\left\{\begin{matrix}\text{Ball}\\ \text{Sleeve}\end{matrix}\right\}$ Bearings

Typical Specifications for D-C Motor Control

Control For $\left\{\begin{matrix}\text{Constant}\\ \text{Adjustable}\\ \text{Adjustable Varying}\end{matrix}\right\}$ Speed D-C

Motors Shall Be $\left\{\begin{matrix}\text{Manual}\\ \text{Magnetic}\end{matrix}\right\}$ And Shall Consist of

a Safety Switch And an Enclosed Controller Providing

Overload Protection $\left[\text{And Low Voltage} \left\{\begin{matrix}\text{Protection}\\ \text{Release}\end{matrix}\right\}\right]$.

Typical Specifications for Squirrel-Cage Motors

..................................Hp, ...RPM, $\left\{\begin{matrix}25\\ 60\end{matrix}\right\}$ Cycles,

$\left\{\begin{matrix}208\\ 220\\ 440\\ 550\end{matrix}\right\}$ Volts, $\left\{\begin{matrix}2\\ 3\end{matrix}\right\}$ Phase, Squirrel-Cage Induc-

tion Motors of the *NEMA*

$\left\{\begin{matrix}\text{Class A—Normal Starting Torque, Normal Starting Current}\\ \text{Class B—Normal Starting Torque, Low Starting Current}\\ \text{Class C—High Starting Torque, Low Starting Current}\\ \text{Class D—High slip}\end{matrix}\right\}$ Type

Driving.................. .... Motor shall
(Application)

Be Arranged For $\left\{\begin{matrix}\text{Horizontal}\\ \text{Vertical}\end{matrix}\right\}$ Mounting And Shall

Be Provided With A..
(Open, Splashproof, etc.)

Type of Enclosure, *NEMA* $\left\{\begin{matrix}\text{Class A}\\ \text{Class B}\end{matrix}\right\}$ Insulation,

And $\left\{\begin{matrix}\text{Sleeve}\\ \text{Ball}\end{matrix}\right\}$ Bearings.

the constant speed motor with the exception that the starting resistor must be designed for speed regulating duty which means that it must be capable of carrying the motor current continuously. Speed controllers are available both for constant torque applications and varying torque drives, such as required by fans, in which the torque is reduced considerably at reduced speed.

The *Adjustable Voltage* type of speed control is also often known as the variable voltage or Ward-Leonard system. For machines requiring a wide range in speed control and a large number of steps of control this type of system is used most extensively. The drive consists of one or more d-c motors, the armatures of which are supplied with power from a d-c generator and the fields of all machines are excited from a constant

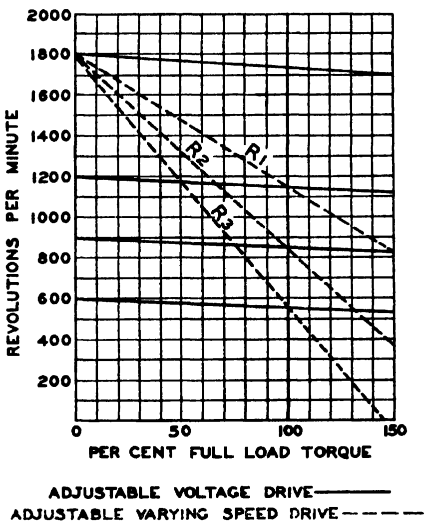


FIG. 3. SPEED-TORQUE CHARACTERISTICS OF ADJUSTABLE VARYING SPEED DRIVE AND ADJUSTABLE VOLTAGE DRIVE

voltage exciter. A schematic diagram of connections for an adjustable voltage drive is shown in Fig. 4. In most cases the d-c generator is a part of a three-unit set including a constant speed a-c driving motor, and a constant voltage exciter. As the voltage on the generator and consequently on the d-c motor or motors is adjusted by a rheostat in the generator field circuit, a great many steps are thus obtained in an efficient manner. With constant field excitation on the motor the speed of the motor will vary approximately as the voltage on the generator.

Extremely wide speed ranges are possible with the adjustable voltage type of drive. Ranges as high as 10 to 1 are common and by the addition of field control on the motors ranges as high as 40 to 1 are permissible. This type of drive provides the advantage of good speed regulation over the entire speed range, as shown in Fig. 3 in which this type of drive is compared with the adjustable varying speed drive.

Typical specifications for d-c motor control are shown on page 634.

ALTERNATING CURRENT MOTORS

Alternating current motors are divided into two main classifications: *polyphase* and *single phase* (see Table 1), according to the type of power supply used. They are further subdivided as to the type of motor winding.

When polyphase power is available it is usually found more economical to apply polyphase motors in preference to single phase motors. A typical 5 hp, 1200 rpm capacitor start-induction run single phase motor for instance, will cost approximately twice as much as the corresponding three phase Class II squirrel-cage motor. In addition, the polyphase motor has the advantages of higher power factor and higher efficiency.

Polyphase Motors

The three types of polyphase motors are: *squirrel-cage induction* motors, *wound rotor induction* motors, and *synchronous* motors.

Squirrel-Cage motors are specified by *NEMA* in classes providing a

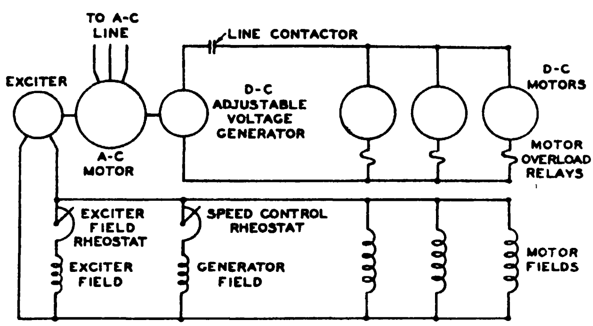


FIG. 4. COMPONENT PARTS OF AN ADJUSTABLE VOLTAGE DRIVE

variety of speed and torque characteristics. Class A motors provide normal starting torque at normal starting current and are suitable for constant speed application, to equipment such as fans and blowers, in which *starting current need not be limited.* Class B motors provide normal starting torque at low starting current and are used for the same type of application as Class A *where starting current must be limited.* Class C motors provide high starting torque with low starting current and are used on compressors, started without unloaders, and on reciprocating pumps. Class D motors have high slip* and are used with flywheels for widely pulsating loads on equipment such as reciprocating compressors and pumps where other motors would draw high peak currents.

Figs. 5, 6, 7, and 8 illustrate the characteristics of squirrel-cage motors. It will be noticed by inspection of Fig. 6 that both power factor and efficiency are improved if the motors are operating as near rated load as possible. In addition, as shown in Fig. 8, power factor and efficiency are better for higher speed motors.

Typical specifications for squirrel-cage motors are shown on page 634.

Wound Rotor motors are used for applications requiring high starting

*Refer to Glossary at end of Chapter.

torque at low starting current, because a wound rotor motor with its controller and resistance can develop full load torque when starting with about full load current. For comparison, a squirrel-cage motor would require from 3 to 5 times as much current to develop full load torque at starting. The wound rotor motor is also used for varying speed service to drive fans, blowers, and other continuous duty apparatus. Typical specifications for wound rotor motors are shown on page 640.

The addition of resistance to the secondary winding of the wound rotor motor changes the speed torque characteristics as indicated in Fig. 9. The motor speed, with the resistance added, is dependent on load and consequently the motor has very poor speed regulation when secondary resistance is added to reduce the speed to values below 50 per cent.

Synchronous motors are used for continuous duty applications at constant speed where efficiency and power factor are important. Another

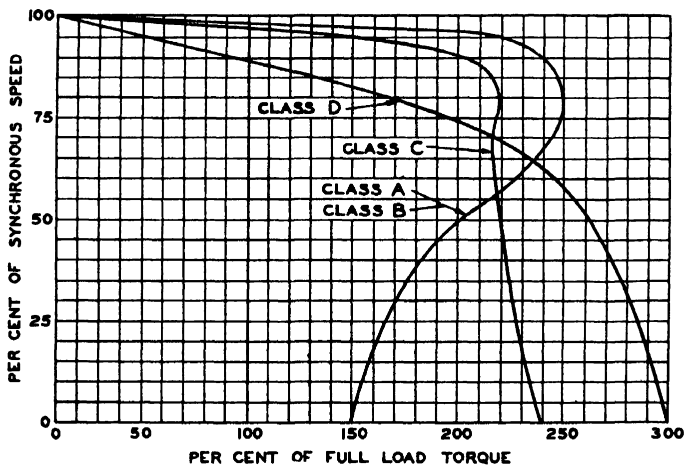


FIG. 5. SPEED TORQUE CHARACTERISTICS OF SQUIRREL-CAGE MOTORS

advantage of these motors is that of lower initial cost in large sizes when compared with squirrel-cage type motors.

The outstanding advantage of the synchronous motor is that its power factor can be changed to compensate for the low power factor of other drives in the same location. Lagging power factor is an inherent characteristic of all induction apparatus, such as induction motors and neon signs. Unless synchronous motors or capacitors are installed, the plant power factor may be comparatively low. This does not necessarily mean that corrective equipment must always be installed, but in most cases it is desirable to determine what advantages may be gained by improving the power factor. With purchased power, if the rates include a clause embodying a penalty for low power factor, or a bonus for high power factor, the saving in power costs may often make a very good return on the investment required for the corrective equipment.

Synchronous motors are used to drive fans, blowers, pumps, compressors and other applications. Compressor applications having a high peak torque require the use of flywheels to smooth out power peaks; and

should always be referred to the electrical manufacturer for recommendations.

Synchronous motors are provided with built-in damper windings on the rotor and operate during the starting period similarly to squirrel-cage motors. After the motor is nearly up to speed, field excitation is applied and the motor draws into step at synchronous speed. After excitation is applied the motor runs at exactly constant speed and will remain at this speed until a load approaching the pull-out load is reached, whereupon the motor pulls out of synchronism and stops.

In applying synchronous motors consideration must be given to the torque the motor can develop on pull-in, that is, at the instant when field excitation is applied. Table 2 tabulates typical application requirements of synchronous motor drives, listing starting, pull-in, and pull-out torques.

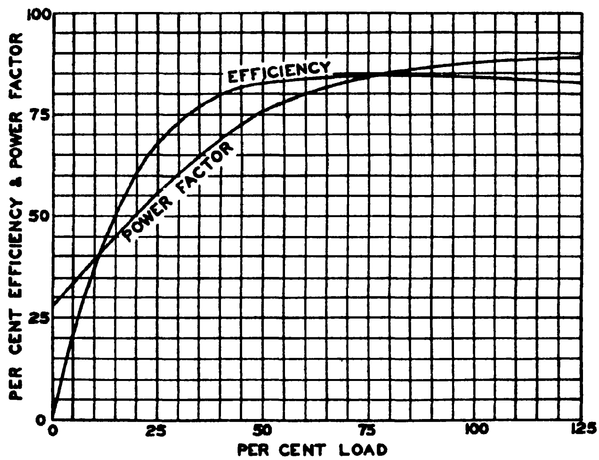


Fig. 6. Variation of Efficiency and Power Factor with Load for a Typical 5 Hp, 4 Pole 60 Cycle, Class B, Squirrel-Cage Motor

Typical specifications for synchronous motors are shown on page 640.

Multi-Speed motors provide flexibility in many types of drives. Synchronous motors can be furnished only with a 2 to 1 ratio in speed, single winding. Squirrel-cage induction motors may be 2, 3 or 4 speed. Two-speed induction motors are usually of single winding type, having a 2 to 1 speed ratio such as 600 rpm and 1200 rpm, or may be double winding. Three-speed induction motors are always two winding, and four-speed motors are usually two winding with a 2 to 1 speed ratio in each winding. Motors can be provided in constant torque, varying torque or constant horsepower ratings. The constant horsepower type of motor is considerably larger than the constant torque motor due to the fact that the same horsepower must be developed at either reduced speed or high speed.

In selecting two-speed motors for fan, pump, blower, or compressor applications, it is usually found that two winding motors are more expensive than the single winding type. The control cost for two-speed,

two winding motors, however, is more economical, and therefore the combined price of both motor and control for the two winding motor is only slightly higher. Because of the improved performance of the two winding motor and because of the factor of safety provided by two independent windings, the increased cost is frequently considered worth the difference.

Single Phase Motors

Single phase induction motors inherently develop no starting torque and are provided with auxiliary windings and devices for starting purposes. The motors are classified as to the type of auxiliary winding provided.

Capacitor Start-Induction Run motors develop high starting torque

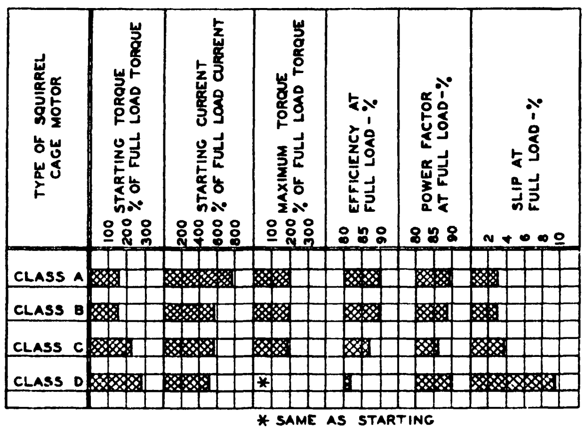


FIG. 7. COMPARATIVE PERFORMANCE OF SQUIRREL-CAGE MOTORS OF 30 HP AND SMALLER SIZES

with low starting current and are used for all types of constant speed heavy duty drives such as compressors, pumps, and stokers. During the starting period, a winding with a capacitor in series is connected in the motor armature circuit and when the motor comes up to speed a centrifugal switch cuts the capacitor and second winding out of the circuit.

Capacitor motors, which are ideally suited for small fan drives, are similar to the capacitor start-induction run type except that the capacitor is not cut out when running.

Repulsion Start-Induction Run motors develop extremely high starting torque. They are supplied with a short-circuiting switch which cuts out the commutator when the motor comes up to speed. These motors are suitable for applications such as industrial compressors where high break-away torque is required and where commutator and brush noise are not factors.

Split Phase motors have a high resistance auxiliary winding which is in the circuit during starting but is disconnected through the action of a centrifugal switch as the motor comes up to speed. Under running conditions it operates as a single phase induction motor with one winding

Typical Specifications for Wound Rotor Motors

.......................Hp, ...RPM, $\left\{\begin{matrix}25\\60\end{matrix}\right\}$ Cycles,

$\left\{\begin{matrix}208\\220\\440\\550\end{matrix}\right\}$ Volts, $\left\{\begin{matrix}2\\3\end{matrix}\right\}$ Phase, Wound Rotor Induction

Motors For Driving ....
(Application)

Motor Shall Be Arranged for $\left\{\begin{matrix}\text{Horizontal}\\\text{Vertical}\end{matrix}\right\}$ Mount-

ing And Shall Be Provided With A ....
(Open, Splash-

....Type of Enclosure, *NEMA*
proof, etc.)

$\left\{\begin{matrix}\text{Class A}\\\text{Class B}\end{matrix}\right\}$ Insulation and $\left\{\begin{matrix}\text{Ball}\\\text{Sleeve}\end{matrix}\right\}$ Bearings.

Typical Specifications for Synchronous Motors

.... Hp, .. RPM, ... · Per Cent P F, ...

Volts, $\left\{\begin{matrix}2\\3\end{matrix}\right\}$ Phase, $\left\{\begin{matrix}25\\60\end{matrix}\right\}$ Cycles, Synchronous

Motors of the $\left\{\begin{matrix}\text{Belted}\\\text{Coupled}\\\text{Engine}\end{matrix}\right\}$ Type For Driving

.... Motor Shall be Arranged
(Application)

for $\left\{\begin{matrix}\text{Horizontal}\\\text{Vertical}\end{matrix}\right\}$ Operation and Shall Be Provided

With A.. ..... Type of En-
(Open, Splashproof, etc.)

closure. Motor Shall be Capable of Developing A Starting Torque of .... Per Cent Full Load Torque, A Pull-In Torque of .... Per Cent Full Load Torque, And A Pull-Out Torque of .... Per Cent Full Load Torque. The Motor Field Shall

Be Excited From A

$\left\{\begin{matrix}\text{Direct Connected Exciter}\\\text{Belted Exciter}\\\text{M-G Set Exciter}\\\text{D-C Bus}\end{matrix}\right\}$ Which $\left\{\begin{matrix}\text{Shall}\\\text{Shall Not}\end{matrix}\right\}$ Be

Included With The Motor.

in the circuit. These units are available for the small horsepower ratings and when equipped with a high slip rotor may be used for adjustable varying speeds through line voltage control. The motors are ideally suited for fan duty.

Speed-torque characteristics of single phase motors are shown in Fig. 10.

Typical specifications for single phase motors are shown on page 642.

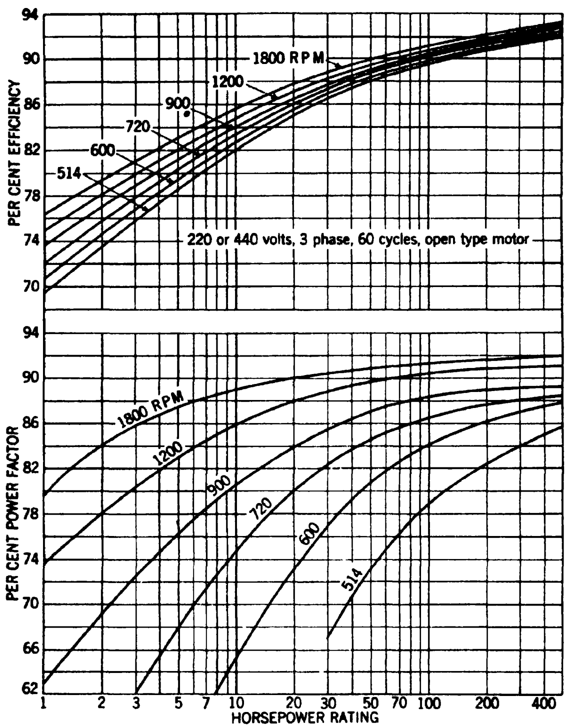


Fig. 8. Efficiencies and Power Factors for Squirrel-Cage Induction Motors

CONTROL FOR ALTERNATING CURRENT MOTORS

Squirrel-Cage motors are usually linestarted where power company limitations permit. In small sizes up to 2 hp the motors are started by means of snap switches with an overload current element for motor protection. In larger ratings a linestarter is usually provided, either with an additional safety disconnect switch or with a switch built into the starter. Reduced voltage starting may be either of manual or push button controlled magnetic type. In specifying this type of starter consideration should be given to the fact that starting torque of squirrel-cage motors varies as the square of the applied voltage. For example, a motor developing 100 lb-ft starting torque on full voltage would produce only 25 lb-ft torque on starting on half rated voltage. Fig. 11 illustrates recommended control practice for squirrel-cage motors.

Typical Specifications for Single Phase Motors

..................Hp, ...RPM, $\left\{\begin{matrix}25\\60\end{matrix}\right\}$ Cycles, $\left\{\begin{matrix}115\\230\end{matrix}\right\}$ Volts,

Single Phase Motor of the..
(Capacitor, Split Phase, etc.)

Type for Driving.. Motor Shall
(Application)

Be Arranged for $\left\{\begin{matrix}\text{Horizontal}\\\text{Vertical}\end{matrix}\right\}$ Mounting And Shall

Be Provided With a..Type
(Open, Splashproof, etc)

of Enclosure, *NEMA* $\left\{\begin{matrix}\text{Class A}\\\text{Class B}\end{matrix}\right\}$ Insulation, And $\left\{\begin{matrix}\text{Ball}\\\text{Sleeve}\end{matrix}\right\}$ Bearings.

Typical Specifications for Squirrel-Cage Motor Control

Control for Squirrel-Cage.. . .
(Fan, Pump, etc.)

Motors Shall Consist of An Enclosed $\left\{\begin{matrix}\text{Manual}\\\text{Magnetic}\end{matrix}\right\}$

Type $\left\{\begin{matrix}\text{Across The Line}\\\text{Reduced Voltage}\end{matrix}\right\}$ Starter Providing Over-

load Protection And Low Voltage $\left\{\begin{matrix}\text{Protection}\\\text{Release}\end{matrix}\right\}$.

Control Shall Include A Safety Disconnect

Switch $\left\{\begin{matrix}\text{Separately Mounted}\\\text{In a Common Enclosure}\end{matrix}\right\}$ With the Starting

Controller.

Typical Specifications for Wound Rotor Motor Control

Control for Wound Rotor Motor Control for.............Applications
(Fan, Pump, Etc.)

Shall Consist of a Safety Disconnect Switch,

a $\left\{\begin{matrix}\text{Separately}\\\text{Common}\end{matrix}\right\}$ Mounted Across the Line Starter

Providing Overload Protection, Low Voltage

$\left\{\begin{matrix}\text{Protection}\\\text{Release}\end{matrix}\right\}$, and a Secondary $\left\{\begin{matrix}\text{Starting}\\\text{Speed Regulating}\end{matrix}\right\}$

Controller for $\left\{\begin{matrix}\text{Separate Mounting}\\\text{Mounting in Same Enclosure}\end{matrix}\right\}$.

Primary and Secondary Control Shall Be Interlocked so as to Provide Complete Control from the Rheostat Handle.

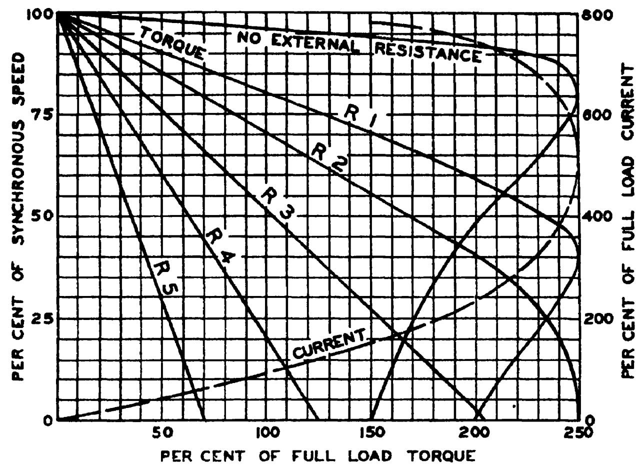


FIG. 9. PERFORMANCE CHARACTERISTICS OF A WOUND ROTOR MOTOR WITH EXTERNAL RESISTANCE

Typical specifications for squirrel-cage motor controls are shown on page 642.

Wound Rotor motors require control of both primary and secondary circuits. The primary* control may be the same as for squirrel-cage motors, manual or magnetic, at full voltage. Secondary* control provides means of varying secondary resistance for starting and speed control. The secondary controller should be specified for starting duty

*Refer to Glossary at end of chapter

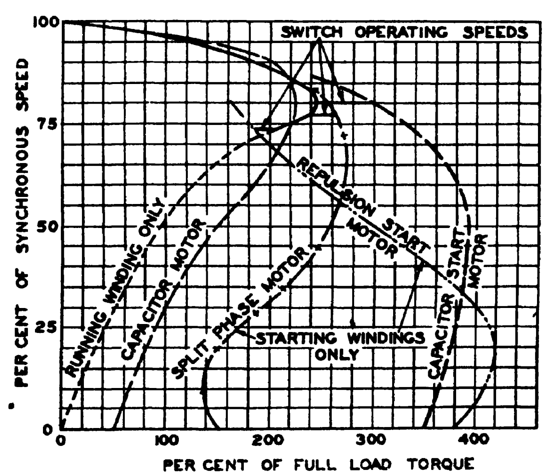


FIG. 10. SPEED-TORQUE CHARACTERISTICS OF SINGLE PHASE MOTORS

only or for speed regulating duty. Fig. 12 illustrates recommended control practice for wound rotor motors.

Typical specifications for wound rotor motor control are shown on page 642.

Synchronous motor controllers should provide pull-out protection, automatic synchronization or automatic stopping of the motor after pull-out, and insurance of complete starting sequence, as well as overload and low voltage protection. The control may be either magnetic or semi-magnetic at full or reduced voltage. Semi-magnetic starters provide automatic field control but require hand operation for closing the line contactors to start and transfer to full voltage.

In applying reduced voltage starters to synchronous motors it should

TABLE 2. TYPICAL APPLICATION REQUIREMENTS OF SYNCHRONOUS MOTOR DRIVES SHOWING STARTING, PULL-IN AND PULL-OUT TORQUES

Application		Method of Connecting Motor to Load	Starting Conditions	Torques			Remarks
				Starting	Pull-in	Pull-Out	
Fans	Exhaust and Ventilating	Coupled or Belted	Usually Loaded	50	60-125	150	WR^2 of Fan Must be Considered
Blowers	Cycloidal Positive	Coupled or Engine Type	Unloaded	40-60	40-60	150	Two-Speed Motors Sometimes Used
	Blowing Engines Reciprocating	Engine Type	Unloaded	40	40-60	150	
	Turbo High Speed	Direct Connected or Step Up Gear	Unloaded (Intake Closed)	30	50	150	WR^2 of Blower Must be Considered
Compressors	Air	Engine Type	Unloaded	40	40	150	Flywheel Effect Important
	Ammonia and Ammonia Booster	High Speed—Belted Low Speed—Engine Type Occasionally Coupled	Unloaded (By By-Pass)	40	40	150	Flywheel Effect Important
	Freon	High Speed—Belted Low Speed—Engine	Unloaded (By By-Pass)	45	60	150	Flywheel Effect Important
	Gas Reciprocating	High Speed—Belted Low Speed—Engine	Unloaded (By By-Pass)	40	40	150	Flywheel Effect Important

be remembered that, since these motors are started on damper windings and during the acceleration period function similarly to squirrel-cage motors, the starting torque varies as the square of the applied voltage. Consideration should be given to insure development of sufficient motor torque to accelerate the load.

Typical specifications for synchronous motor control are shown on page 645.

Multi-Speed control may be either manual or magnetic, and at full or reduced voltage. When using automatic magnetic control with two-, three-, and four-speed separate winding or consequent-pole motors, control may be obtained from a remote point by means of a push button master switch. The various speeds of the motor are obtained from the master switch by simply depressing the correct push button. This is known as selective speed control. It is commonly used in the smaller

Typical Specifications for Synchronous Motor Control

Synchronous Motor Control for.. ....
(Application)

Motors Shall Provide for {Full Voltage / Reduced Voltage}

Starting And Shall Be {Full Magnetic / Semi-Magnetic}. (Reduced

Voltage Starting Shall Be Obtained By Means of

{Autotransformers / Resistors / Line Reactors / Saturable Reactors} And Shall Limit The KVA

Inrush to a Maximum of .... per cent of Full Load KVA)

The Control Panel Shall Be For {Isolated / Switchboard}

Assembly And Shall Be of {Open Switchboard / Enclosed Cubicle} Con-

struction. It Shall Provide Overload, Under

Voltage Protection And After Pulling Out of Step

Will {Automatically Resynchronize / Automatically Stop} The Motor.

Typical Specifications for Single Phase Motor Control

Control For Single Phase . ...
(Fan, Pump, etc)

Motors Shall Consist of a {Manual / Magnetic} Type

{Across The Line / Reduced Voltage} Starter Providing Overload

Protection (And Low Voltage); And A Separate
Safety Disconnecting Switch.

theater installations where the fan and motor are located backstage and the speed control is located in the lobby.

Multi-speed motor controllers may be provided with compelling relays which make it necessary for the operator to press the first speed button before regulating the motor to the desired speed. This insures that the motor is always started at low speed before adjusting to one of the higher speeds.

Timing relays which provide for automatic acceleration may be used for control. With this feature the motor will always start at low speed and automatically accelerate to the desired speed. Decelerating relays may be used to reduce the shock effect of the braking action to the motor and drive when the speed is reduced from a higher to a lower speed.

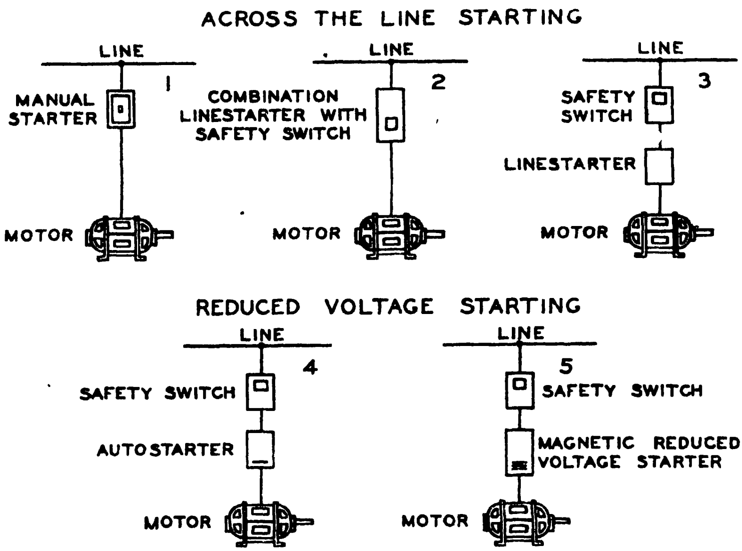
Single Phase motor control usually consists only of a linestarter, either

manual or magnetic. In some cases it is desirable also to provide a disconnect switch. Fig. 13 illustrates the recommended controls.

Typical specifications for single phase motor control are shown on page 645.

GEAR MOTORS

A gear motor is a self-contained combination of any type of a-c or d-c motor and an enclosed speed-reducing gear, providing a more compact and readily adaptable unit than is obtained by using a motor coupled to a gear reducer. Gear motors are available in sizes up to 75 hp with output



Arrangements 2, 3, 4 and 5 provide automatic push-button starting.

FIG. 11. RECOMMENDED CONTROLS FOR SQUIRREL-CAGE MOTORS

shaft speeds from about 4 to 1430 rpm, making it possible to couple or to connect by gear or chain to nearly any machine. High speed motors are used, generally 1800 rpm on 60 cycles, thus obtaining the advantages of high motor power factor and efficiency. The gearing efficiency is also high, usually about 98 per cent for a single reduction of the helical or spur type, that is, a 2 per cent loss for one reduction or 4 per cent loss for a double reduction. Consequently, the over-all performance of the gear motors is much higher than a combination of open gearing, belting, countershaft, or other arrangement, which would otherwise be required. Gear motors are used extensively to drive numerous types of slow speed drives. Besides being more effective than other combination drives in saving space, they are important in reducing maintenance and operating hazards.

GLOSSARY

General Definitions

NEMA is the abbreviation for the *National Electrical Manufacturers Association*.

Speed Regulation (d-c motors) is the change in speed between no-load and full-load,

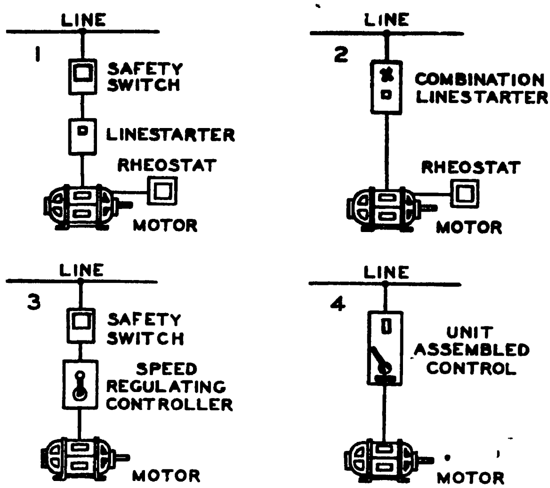


FIG. 12. RECOMMENDED CONTROLS FOR WOUND ROTOR MOTORS

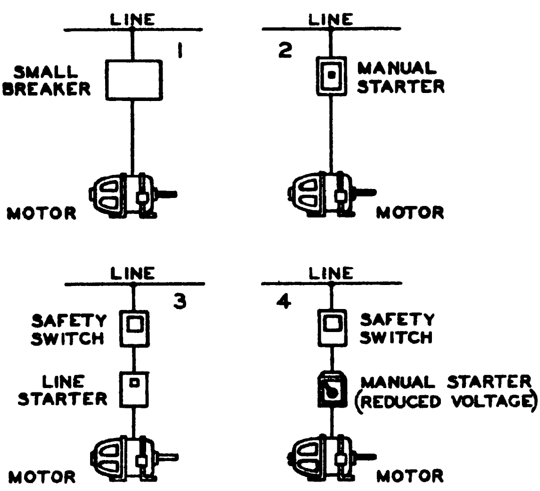
expressed in per cent of full-load speed; for example, a motor having a no-load speed of 1200 rpm and a full load speed of 1140 rpm would have a speed regulation of 5.6 per cent.

Slip (a-c induction motors) is the difference between the motor speed and synchronous speed expressed in per cent of synchronous speed, *e g.*, a 1200 rpm motor operating at 1140 rpm would have a slip of 5 per cent.

Torque is an expression of the turning effort developed by the motor at the shaft and is usually expressed in ounce-feet for fractional horsepower motors and in pound-feet for motors of larger ratings.

Primary is the term usually applied to the high voltage or line side of a transformer or motor. In the case of the wound rotor motor the primary is the stator winding.

Secondary is the term usually applied to the low voltage or load side of a transformer or motor. In the case of the wound rotor motor the secondary is the rotor winding.



Arrangements 2, 3 and 4 are optional for motors up to 7½ hp, 220 volts.

FIG. 13. RECOMMENDED CONTROLS FOR SINGLE PHASE MOTORS

NEMA Classification of Motor Enclosures

Open motors (40 C rise) are self-ventilated machines having no restriction to ventilation other than that necessitated by mechanical construction.

Protected motors (50 C rise) have all ventilating openings in the frame protected by perforated covers.

Semi-Protected motors (50 C rise) have the ventilating openings in the top half of the frame only protected by perforated covers.

Drip Proof motors (50C rise) are so constructed that drops of liquid or solid particles falling on the machine at any angle not greater than 15 deg from the vertical cannot enter the machine either directly or by striking and running along a horizontal or inclined surface.

Splash Proof motors (50 C rise) are so constructed that drops of liquid or solid particles falling on the machine or coming towards it in a straight line at any angle not greater than 100 deg from the vertical cannot enter the machine either directly or by striking and running along the surface.

Totally Enclosed Non-Ventilated motors (55 C rise) are so constructed as to prevent exchange of air between the inside and outside of the case.

Totally Enclosed Fan-Cooled Motors (55 C rise) are similar to totally enclosed, non-ventilated machines except that exterior cooling is provided by means of a fan or fans integral with the machine.

Explosion Proof motors* (55 C rise) have an enclosing case designed to withstand an explosion of a specified gas or vapor which may occur within it, and to prevent the ignition of the gas or vapor surrounding the motor by sparks, flashes, or explosion of the gas or vapor which may occur within the machine casing.

Water Proof motors (55 C rise) are so constructed as to exclude water applied in the form of a stream from a hose.

Dust Tight motors (55 C rise) are so constructed that the enclosing case will exclude dust.

Motor Speed Classifications

A Constant Speed Motor is one in which the speed remains practically constant with changes in load; *e.g.*, a d-c shunt wound motor or a-c squirrel-cage motor with low slip.

A Varying Speed Motor is one in which the speed varies with the load, usually decreasing when the load increases; *e.g.*, a d-c series motor or an induction motor with large slip.

An Adjustable Varying Speed Motor is one in which the speed can be adjusted gradually, but when once adjusted for a given load will vary in considerable degree with change in load; *e g*, a shunt wound d-c motor adjusted by armature resistance control.

An Adjustable Speed Motor is one in which the speed can be varied gradually over a considerable range, but when once adjusted remains practically unaffected by the load; *e.g.*, a d-c shunt motor with field resistance control. The standard ratings for open type, adjustable speed motors, having a speed range of 3 to 1 and greater are in accordance with the following:

(1) A standard continuous horsepower rating at 150 per cent of minimum speed with a temperature rise of 40 C.

(2) The next higher standard continuous horsepower rating at 3 times minimum speed with a temperature rise of 40 C.

(3) Between 150 per cent of minimum speed and 3 times minimum speed, the standard continuous horsepower rating with a temperature rise of 40 C will vary with the speed along a straight line connecting these two horsepower ratings. No further increase in horsepower is recognized above 3 times minimum speed

(4) Below 150 per cent of minimum speed the lower continuous horsepower rating (see preceding item 1) will apply with a temperature rise of 50 C.

(5) Motors may also be rated 1 hour with temperature rise of 50 C with the higher horsepower rating (see preceding item 2) throughout the entire speed range.

Example: 20/25 hp, 400 to 1600 rpm This motor may be rated 25 hp, 50 C, 400/1600 rpm; 1 hour.

Mechanical Modifications

Vertical Mountings are available for such applications as pumps, agitators, and so forth. This type of application may require a special umbrella-type hood to protect against dripping liquids.

Flanged Mountings are available for use where motors are built in as part of machines. Motors may also be supplied with flush plate mountings, suitable for close coupled pump and similar applications.

CHAPTER 36

Air Conditioning in the Treatment of Disease

Operating Rooms, Reducing Explosion Hazards, Nurseries for Premature Infants, Fever Therapy, Cold Therapy, High Temperature Hazards, Control of Allergic Disorders, Oxygen Therapy, General Hospital Air Conditioning

IN the past few years air conditioning has made considerable progress as an adjunct in the treatment of various diseases. Among the important applications are those in operating rooms, nurseries for premature infants, maternity and delivery rooms, children's wards, clinics for arthritic patients, heat therapy, cold therapy, oxygen therapy, X-ray rooms, the control of allergic disorders, and for the physiological effects in industry.

OPERATING ROOMS

The widest application of air conditioning in hospitals is in operating rooms. Complete air conditioning of operating wards is important because winter humidification helps reduce the danger of anesthetic gases; summer cooling with some dehumidification is needed to eliminate excessive fatigue and to protect the patient and operating personnel; and finally, filtering for the removal of allergens from the operating room air.

Reducing Explosion Hazard

Explosion hazards in operating rooms began with the introduction of modern anesthetic gases and apparatus. Ether administered by the old drop method gives rise to an explosive mixture, but in practice this method is still regarded as comparatively safe. When ether is mixed with pure oxygen, or nitrous oxide in certain concentrations, the explosion hazard may be as great as with ethylene-oxygen, or cyclopropane-oxygen mixtures[1].

During the course of ethylene anesthesia, the mixture, usually 80 per cent ethylene and 20 per cent oxygen, is so rich that the danger of explosion is slight in the immediate vicinity of the face mask, but leakage of ethylene into the air may accumulate to any lower concentration, and thus introduce a serious hazard. The most dangerous period is at the end of the operation when the patient's lungs and the anesthesia apparatus are customarily *washed out* with oxygen with or without the addition of carbon dioxide. Even when this procedure is omitted, it is difficult in practice to avoid dilution of the anesthetic gas with air during the normal course of breathing following the administration. In either case the mixture would pass through the explosion range and extraordinary precaution is necessary for the safety of the patient and operating personnel. Static sparks may result from accumulation of frictional charges on the rubber surfaces of the anesthetic apparatus, on woolen blankets, and on the bodies of the operators as they walk on insulated floors, when the humidity is low.

In a recent study[2] of 230 anesthetic explosions and fires, 70 per cent of the explosions and 60 per cent of the deaths were caused by igniting

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[1]Safeguarding the Operating Room Against Explosions, by Victor B. Phillips (*Modern Hospital*, 46, April and May, 1936).

See also Explosion Hazards of Combustible Anesthetics, by G. W Jones, R. E. Kennedy and G. J. Thomas (*U S Bureau of Mines, Technical Paper* No 653, 1943).

[2]The Hazard of Fire and Explosion in Anesthesia, by B A. Green (*Anesthesiology* 2 144, 1941).

agents other then static sparks. In 1941 the *National Fire Protection Association*[3] made certain recommendations for safe practice based on available information. Some of these recommendations are:

Windows should be kept closed to avoid explosive pooling of anesthetic gases. Twelve air changes per hour and a humidity of 55 per cent are advised. If a higher humidity were compatible with the well being of the patient and personnel, it should be maintained. All electrical installations should comply with the standards set by the *National Electrical Code* for use in explosive situations. Cautery equipment should not be used in hazardous locations. To prevent static sparks, all bodies in an operating room should be conductive or coupled. It is essential that adequate grounding be provided for the floor and every object in the operating room. Conductive rubber should be used on shoes, leg tips, operating table coverings and all rubber parts of the anesthesia equipment. All furniture in contact with the floor should be metal. In the absence of complete grounding facilities, the simple method of intercoupling patient, operating table, anesthetist and gas machine at ground potential may be used.

The carbon dioxide absorption method of administering anesthetics in a closed system confines explosive gases and obviates contact with various sources of ignition. Explosions in a closed system are likely to be more serious. However, proper precautions will reduce this hazard to a minimum.

It should be realized that when a room and the occupants have been completely grounded there is always the possibility that the patient or the operator might receive a dangerous shock if a short circuit developed in any of the electrical equipment.

A comprehensive study of the explosion problem and of the general causes and prevention of operating room hazards is being conducted by the *University of Pittsburgh*, the A.S.H.V.E. Research Laboratory, and the *U. S. Bureau of Mines*. The first result of this investigation has been a fruitful attempt to eliminate the explosive range of cyclopropane, one of the best but most difficult gases to handle. The use of helium as a diluent in the total gaseous mixture controls the oxygen concentration by replacement and since its flame quenching qualities are known it is the ideal gas for this purpose. In addition, a gaseous mixture containing helium is more difficult to ignite by electric discharges and this quality also increases the safety factor of anesthetic administration.

Operating Room Conditions

Little is known about optimum air conditions for maintaining normal body temperatures during anesthesia and the immediate post-operative period. An anesthetized patient displays dilatation of blood vessels in the skin resulting in profuse sweating and (it has been believed) inability to regulate body temperature. From this it was concluded that all anesthetized patients suffered considerable heat loss. In spite of this a recent paper[4] reports little more than 0.8 F variation in the rectal temperature during the course of the operation. The severe physiological effects, such as excessive sweating and rapid pulse, of high operating room temperatures on attendants and patients during the hot months

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[3]Control of Physical Hazards of Anesthesia, by R. M. Tovell and A W. Friend (*Canadian Medical Association Journal* 46.560, 1942).

[4]A.S.H.V.E. RESEARCH REPORT No. 1111—Air Conditioning Requirements of an Operating Room and Recovery Ward, by F. C. Houghten and W. Leigh Cook, Jr. (A.S H.V E. TRANSACTIONS, Vol. 45, 1939, p. 161)

signify the need for proper cooling. A comparison of surgeons' statements who operate in both air conditioned and non-air conditioned rooms strongly indicates lesser fatigue; and the greater recuperative power of the patient is confirmed by the study[5].

Although the comfortable air conditions for the operatives are not identical with those for the patient a compromise is as a rule not difficult; with a relative humidity of 55 to 60 per cent, temperatures from 72 to 80 F are used. The work just cited, reported that 68 to 70 F effective temperature not only furnished comfort for the operating room workers but apparently prevented exhaustion of the patient as evidenced by rapid convalescence in the recovery ward. Additional heat may be furnished to the patient locally or by suitable covering according to body temperature in individual cases.

The control of air-borne infection falls into two categories, namely, those which prevent dispersal of infectious material into the air and those employed or proposed for reducing the infectivity of already contaminated atmospheres, by removing or killing the disease-producing agents. Among others, two of the important preventative measures are isolation of the infected patient and adequate masking of the hospital personnel[6]. The means most commonly employed for reducing air-borne infectious agents is ventilation.

In an investigation recently conducted at the University of Pittsburgh, in a cooperative research program with the Society, comparative studies were made on bacterial content of conditioned and non-conditioned operating rooms. From these studies[7] it was concluded that the bacterial content of conditioned operating rooms was considerably less than that of non-conditioned rooms. The degree of air contamination can be reduced by proper ventilation, but recontamination of the air by occupants of the operating room prevents reduction in the bacterial level to a point adequate for the prevention of wound infections[8]. Research is in progress on the use of filtered air flowing through a system of complete mechanical barriers which would protect the patient completely against infection by attendants and bacteria-containing air in the corridor or ward[9].

Operations may be postponed on allergic patients during asthmatic manifestations through fear of complications. The removal of air-borne allergens, therefore, is in some cases an important function of the air conditioning system in preparing patients for operation.

Central system air conditioning plants and unit air conditioners prove satisfactory in operating rooms when producing between 8 and 15 air changes per hour of filtered and properly conditioned air without recirculation during the course of anesthesia. A separate exhaust fan system is usually necessary to confine and remove the gases and odors. Double windows are desirable and often necessary to prevent condensation and frosting on the glass in cold weather and to minimize drafts. The high air flow of 8 to 15 air changes in operating rooms is desirable for three reasons: (1) to reduce the concentration of the anesthetic to well below the pharmacologic threshold in the vicinity of the operating personnel,

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[5] Loc Cit. Note 4

[6] Air-Borne Infection, by O. H Robertson (*Science*, 97:495, 1943).

[7] Report on Air Conditioning in Surgery, by W Leigh Cook, Jr. (*Department of Industrial Hygiene, School of Medicine, University of Pittsburgh*, 1940)

[8] Unexplained Infections in Clean Operative Wounds, by Deryl Hart and S. E. Upchurch (*Annals of Surgery*, 114:936, 1941).

[9] The Control of Cross-Contamination by the Use of Mechanical Barriers, by J A. Reyniers (*Aerobiology, American Association Advanced Science Symposium*, 17:254, 1942).

(2) to remove the great amounts of heat and sometimes moisture, from sterilizing equipment if inside the operating room, from the powerful surgical lights, from solar heat, and from the bodies of the operatives, and (3) to provide extra capacity for quickly preparing the room for emergency operations. Much can be gained by careful insulation of sterilizing equipment and by thorough exhaust ventilation of sterilizing rooms adjoining the operating rooms.

A very common complication presumably traceable to operations is pneumonia. The difference in conditions between the operating room and the final hospital destination of the patient, including corridors and elevators, is conducive to post-operative pneumonia. A suggested remedy is a recovery ward in which conditions closely approximate those of the operating room and in which the patients remain from one to four days. Satisfactory conditions in the recovery ward not only hasten convalescence, but dispel the fear frequently found in patients who must undergo operations during the hot seasons[10].

Sterilization of Air in Operating Room

The role of light in deterring the spread of respiratory infection is extremely important. Direct sunlight, diffuse daylight from a blue sky and even light which has passed through window glass are all bactericidal in varying degrees[11]. Hence extensive window space is desirable in operating rooms and hospital wards. The ultra-violet range of the spectrum is the most potent for the sterilization of air. Results reported[12] would indicate that the post-operative temperature rise of patients during the first few days is in most instances caused more by bacterial contamination of the operative wound than by the absorption of blood and traumatized tissues. Operating room infections are being reduced subsequent to the installation of special ultra-violet lamps. In the use of ultra-violet radiation it is important to use the most efficient part of the spectrum with an intensity great enough to render the air relatively sterile and yet not injure tissue for the period of time it will be exposed[13]. The Council on Physical Therapy of the *American Medical Association* in a recent action voted acceptance of ultra-violet lamps for disinfecting purposes[14]. A subcommittee of the Council on Hospital Planning and Plant Operation of the *American Hospital Association* has recommended that the installation of ultra-violet light equipment should be considered when new contagious disease or surgical operating units are built[15].

The idea of employing bacterial mists as a method for controlling air-borne respiratory infection is not new, but until recently no one has succeeded in producing a sterile or relatively bacteria-free atmosphere by such means which could be tolerated by human beings[16]. The mists consist of droplets (liquid aerosols) 1 to 2 micra in diameter dispersed in air. Aerosol solutions of sodium hypochlorite (1 gram in 40 million cc of

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[10]Report of the Committee on Air Conditioning (*The American Hospital Association*, p 2, 1937)

[11]The Transmission of Certain Infections of Respiratory Origin, by L Buchbinder (*Journal of the American Medical Association* 118·718, 1942).

[12]Effect on Wound Healing of Bactericidal Ultraviolet Radiation from a Special Unit-Experimental Study, by Deryl Hart (*Archives of Surgery* 38 797, 1939).

[13]Considerations in the Use of Ultraviolet Radiation in Operating Rooms, by C. J. Kraissl, J. G Cimiotti and F. L. Meleney (*Annals of Surgery* 111.161, 1940)

[14]Acceptance of Ultra-violet Lamps for Disinfecting Purposes, by Council on Physical Therapy (*Journal of the American Medical Association* 118.298, 1942).

[15]Ultra-violet Rays as a Sterilization Agent in Hospitals, by the Council on Hospital Planning and Plant Operation (*American Hospital Association Bulletin* No. 203, 1940).

[16]Bactericidal Action of Propylene Glycol Vapor on Micro-organisms Suspended in Air, by O H Robertson, Edward Biggs, Theodore T. Puck and Benjamin F. Miller (*Journal of Experimental Medicine* 75.593, 1942).

air), a mixture of hexylresorcinal (10 per cent), and an alkysulfate (*lorol* 0.05 per cent) in alkaline propylene glycol (1 gram of the mixture to 4 billion cc of air) have been reported to sterilize contaminated air samples. Other compounds tested have been either inactive as aerosols or too toxic to be used. Recent work indicates that propylene glycol aerosols in concentrations as low as 1 gram of glycol to 50 million cc of air produce marked and rapid bactericidal effect on atmospheric micro-organisms in droplet form. Atmospheres containing propylene glycol are invisible, odorless and non-irritating. Intensive research in the search for more potent aerosols is in progress. A recent paper[17] reports the use of triethylene glycol mists which prove to be bactericidal in a dilution of 1 gram of glycol to 100 million to 200 million cc of air. Thus far, few data are available on the practical use of germicidal mists and vapors.

NURSERIES FOR PREMATURE INFANTS

One of the most important requirements in the care of premature infants is the stabilization of body temperature. This is necessary because their heat regulating systems are not fully developed; the metabolism is low and the infants generally exhibit marked inability to maintain normal body temperatures. The resistance to infection is low and mortality rate high.

Air Conditioning Requirements

The optimum air conditions for the growth and development of these infants were determined by extensive research[18] at the Infants Hospital, Boston, Mass., using four valid criteria, namely, stability of body temperature, gain in weight, incidence of digestive syndromes, and mortality. Individual temperature requirements varied widely (from 72 to 100 F) according to the constitutional state of the infants and body weights. The optimum relative humidity was about 65 per cent, and the air movement less than 20 fpm.

A single nursery conditioned to 77 F and 65 per cent relative humidity was found to fulfill satisfactorily the requirements of the majority of premature infants. Additional heat for weak (or debilitated) infants may be furnished in the cribs or by means of electric incubators placed inside the conditioned nursery, and the temperature adjusted according to individual requirements. In this way multiplicity of chambers and of air conditioning apparatus is obviated; the infants in the heated beds derive the benefit of breathing cool humid air, and the nurses and doctors need not expose themselves to extreme conditions.

Importance of Humidity: Although external heat is an important factor in the maintenance of normal body temperature, humidity appears to be of equal or greater importance. When the premature nurseries at the Infants Hospital were kept at relative humidity between 25 and 50 per cent for two weeks or longer, the body temperature became unstable, gain in weight diminished, the incidence of gastro-intestinal disturbances increased, and the mortality rose. On the other hand, continuous exposure to air conditions with 55 to 65 per cent relative humidity gave satisfactory results over a period of years. The initial physiologic loss of body weight (loss occurring within first four days of life) was found to

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[17]The Lethal Effect of Triethylene Glycol Vapor on Air-Borne Bacteria and Influenza Virus, by O. H. Robertson, T. T. Puck, H F. Lemon and C. G. Loosli (*Science* 97·142, 1943).

[18]The Premature Infant· A Study of the Effect of Atmospheric Conditions on Growth and on Development, by K. D. Blacktan, C P. Yaglou and K. McKenzie (*American Journal Diseases of Children*, 46. 1175, 1933)

vary inversely with the humidity. In the old nurseries with natural humidity it averaged 12.4 per cent of the birth weight; in the conditioned nurseries it was 8.9 per cent with 25 to 49 per cent relative humidity, and 6.0 per cent with 50 to 75 per cent relative humidity. The number of days required to regain the birth weight was correspondingly maximum in the old nursery and minimum in the conditioned nurseries under high humidity.

Maximum gains in body weight occurred in the conditioned nurseries under high humidity (55 to 65 per cent) in infants weighing less than 5 lb. The gains were less under low humidity (25 to 50 per cent) in the same nurseries, and in the old nurseries prior to the installation of air conditioning apparatus.

The incidence and severity of digestive syndromes, with diarrhea, persistent vomiting, diminishing gain or loss of body weight, and other symptoms, were generally from two to three times as high under low than under high humidity.

Summarizing, the best chances for life in premature infants are created by maintaining a relative humidity of 65 per cent in the nursery and by providing a uniform environmental temperature just sufficiently high to keep the body temperature within normal limits. Medical and nursing care are, of course, factors of equal and sometimes of greater importance.

Air Conditioning Equipment

Most of the installations now in use are of the central system type providing for filtration, for humidification and heating in cold weather, and for cooling and dehumidification in hot weather. A high ventilation rate, between 15 and 25 air changes, is desirable to remove odors and maintain uniformity of temperatures in extremes of weather. Recirculation is not used extensively in these wards owing to odors and the possibility of infection.

Control of Air-Borne Infection

The protection of the premature and older infant against infection is of the utmost importance. It was found in one installation equipped with air conditioning, germicidal lights and mechanical barriers that air conditioning alone did not prevent the spread of respiratory cross-infections. Bactericidal ultra-violet light barriers and air conditioning or mechanical barriers and air conditioning were efficient[19]. A brief preliminary report on the effect of propylene glycol vapor in reducing acute respiratory infections in a children's ward, suggests that it may be effective[20].

FEVER THERAPY

Artificial production of fever in man is an imitation of nature's way of overcoming invading pathogenic organisms. The action may be direct and specific by destruction of the invading organism within the safe limit of human temperatures, or indirect in the case of heat resistant organisms, by general mobilization of the defensive mechanisms of the body, which retard or neutralize the activity of pathogenic bacteria and their toxins. A serious challenge to the theory on which fever therapy is based comes from the demonstration that hyperpyrexia causes a reduc-

[19]Observations on the Control of Respiratory Contagion in the Cradle, by I. Rosenstern (*Aerobiology, American Association for the Advancement of Science, Symposium* 17:242, 1942).

[20]The Effect of Propylene Glycol Vapor on the Incidence of Respiratory Infections in a Convalescent Home for Children, by T. N. Harris and J. Stokes, Jr (*American Journal of Medical Sciences* 204 430, 1942).

tion in the titer of circulating antibodies in experimental animals[21].

The limits of induced systemic fever are usually between 104 and 107 F (rectal), and the duration from 3 to 8 hours at a time. The total period of fever treatment varies with the type of the organism involved from a few hours to 50 or more.

The diseases which respond favorably to artificial fever therapy are gonorrhea and its complications (which include arthritis, pelvic infections in women, and involvement of the eye), syphilis, chorea, infectious arthritis (non-gonorrheal), encephalitis, and some forms of asthma. There are other conditions which show promise under this treatment; but the most striking results are seen in gonorrhea and syphilis, since the causative organisms can be destroyed at temperatures compatible with human life[22].

Equipment for Production of Fever

Various means have been tried for producing artificial fever, including injections of various crystalloid or colloid substances, bacterial products of typhoid and malarial organisms; a number of physical methods, such as hot baths, radiant heat, diathermy, radiothermy, and in the last few years, air conditioned chambers. The relative advantages and disadvantages of various methods have been discussed in a paper[23]. The results by the use of air conditioned cabinets have not been fully explored, and it is therefore difficult to determine all the advantages and disadvantages of the value of air conditioning at this time.

In the earlier studies of the Society[24], temperatures were elevated more easily using saturated atmospheres. A fever therapy apparatus[25] using these same principles has proved efficient as a means of inducing and maintaining fever in a body with small likelihood of burns because of the comparatively low dry-bulb temperatures. This saturation factor is in great use today where fever is created by induction currents by placing the body in an electrical field. When the optimum body temperature has been reached by electrical induction, the atmosphere of the enclosure is kept at saturation to prevent heat loss, thus maintaining the patient's temperature at the desired point. Other apparatus[26] which uses electric heaters, centrifugal fans, and a water container for humidification has been used in the past, but the more recent trend is toward saturation with a lower dry-bulb temperature.

When heat is necessary in treating legs or arms, such media as short or long wave diathermy, infra-red, water baths, etc. have been used extensively. A recent development, a saturated atmosphere heating unit, similar to one previously described[27] has proven satisfactory, because heat may be administered over longer periods which render deep heating possible without fear of burns or shocks[28]. Local heating has been

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[21]The Influences of Artificial Fever on Mechanisms of Resistance, by H. V. Ellingson and P. F. Clark (*Journal of Immunology*, 43:65, 1942).

[22]Report of the First Year of Fever Therapy Research by the *Department of Industrial Hygiene, School of Medicine, University of Pittsburgh*, 1938.

[23]Fever Therapy by Physical Means, by Frank H. Krusen and E. C. Elkins (*Journal American Medical Association*, 112:1689, 1939).

[24]A S.H.V E Research Report No. 654—Some Physiological Reactions of High Temperatures and Humidities, by W. J. McConnell and F. C. Houghten (A.S.H.V.E. Transactions, Vol. 29, 1923, p. 129).

[25]A.S.H.V.E. Research Report No. 1054—Fever Therapy Induced by Conditioned Air, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet (A.S.H.V.E Transactions, Vol. 43, 1937, p. 131). A S.H V.E Research Report No. 1161—Fever Therapy Locally Induced by Conditioned Air, by M. B. Ferderber, F. C. Houghten and Carl Gutberlet (A.S.H.V.E. Transactions, Vol. 46, 1940, p. 307).

[26]Artificial Fever Therapy of Syphilis, by W. M. Simpson (*Journal of the American Medical Association*, 105:2132, 1935).

[27]Loc. Cit. Note 24.

[28]Saturated Atmospheres in the Treatment of Injuries, by M. B. Ferderber (*Industrial Medicine*, 8:256, June, 1939).

somewhat satisfactory in relieving the painful symptoms of peripheral vascular disease.

The final criteria for the use of fever therapy may be changed because of the introduction of certain drugs which appear prominent in the experimental treatment of some diseases for which fever therapy has been efficacious. Preliminary results would seem to indicate that the combination of fever therapy with chemotherapy (sulfonamide) may be the procedure of choice in the treatment of certain chemotherapy-resistant infections[29].

COLD THERAPY

In contrast to fever therapy the use of cold as a means of treatment is being investigated. From the available literature[30] the chief virtues of cold therapy (cryotherapy) in cancer are the reduction of pain due to extensive growths and the possibility that the process may be arrested. For a localized lesion, ice water between 36 to 48 F is circulated through tubing at the site of the disease for periods ranging from 4 to 48 hr. A later development was the principle of hibernation during which time the patient is kept in an air conditioned space with an environmental temperature between 50 to 60 F for five days. The body temperature is reduced below the critical level of 95 F. Most of the vital processes of life are at very low ebb and this period simulates the hibernation of wild animals. Although the relief of pain is reported, this form of treatment is still in the experimental stage.

It has been demonstrated that the cooling of limbs and other parts with ice-water or ice, cracked or pulverized, down to near the freezing point (5 C or 40 F) is harmless. Freezing must be avoided. There is a temporary retardation or suspension of life, with resumption of cellular activity as the temperature returns to normal. A human limb can remain bloodless and anesthetic below a tourniquet for at least 8 hr and perhaps up to 48 hr without injury, while the rest of the body remains warm. Where amputation is indicated it can thus be done without pain, loss of blood or strength and also without shock. There is no apparent interference with the subsequent healing of the stump. Refrigeration anesthesia is of importance not only in amputation but also in the control of hemorrhage, pain, infection and shock during the transportation of patients with traumatized limbs. A portable electric refrigerative apparatus is in use at the New York City Hospital[31]. There is a growing feeling that cooling the body is of great importance in the treatment of shock. Extensive research is in progress to determine what degree of heat or cold is most efficacious in obviating or treating secondary shock[32].

HIGH TEMPERATURE HAZARDS

Diseases due to heat are now classified as heat exhaustion, heat cramps, and heat strokes[33]. Heat exhaustion is due to circulatory failure; heat cramps to excessive loss of body chlorides and heat stroke to an inadequacy of the heat dissipating mechanism which results in heat retention. If the hyperthermia becomes excessive, the liver and the central nervous

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[29]Temperature Factors in Cancer and Embryonal Cell Growth, by L. W. Smith and Temple Fay (*Journal of the American Medical Association*, 113.653, 1939).

[30]Combined Artificial Fever-Chemotherapy, by H. W. Kendell, D. L. Rose and W. M. Simpson (*Journal of the American Medical Association*, 116:357, 1941).

[31]Experiments on Pelvic and Abdominal Refrigeration with Special Reference to Traumatic and Military Surgery, by F. M Allen (*American Journal of Surgery*, 55:451, 1942).

[32]Cooling in Shock—Editorial (*Journal of the American Medical Association*, 121.432, 1943).

[33]Heat Disease: Clinical and Laboratory Studies, by M. W. Hellman and E. S. Montgomery (*Journal of Industrial Disease and Toxicology*, 18·651, 1936).

system may be seriously injured and this damage may prove fatal. The hazards of high temperature are not understood. It is difficult to say whether short exposures at high temperatures are more harmful than longer exposures at lower temperatures. A new concept is evident from the observation of an increase in leucocytes (white cells) of the blood in workers subjected to high temperatures. These leucocytes are defensive factors which are increased when infection invades the body. A rise in temperature and leucocyte count indicates a mobilization of the body defenses. Since a recent study[34] showed that both temperature and cell count were increased, the question arises as to whether long exposures to very high temperatures might not cause exhaustion of these defense mechanisms.

ALLERGIC DISORDERS

Although there is some division of opinion over the ultimate cause of allergy, the prevailing belief is that it is due to an inherited or acquired hypersensitiveness to pollen or other foreign proteins in certain individuals who react abnormally to the offending substance. The reaction may be induced by inhalation, eating, or absorption (through the skin) of the allergens. Some of the clinical manifestations are hay fever, asthma, eczema, and contact dermatitis.

Symptoms of Hay Fever and Asthma

The respiratory tract is the usual site of allergic manifestations, *e.g.* hay fever and asthma. In hay fever, the nose and eyes are red and itchy, and there is considerable discharge. Nasal obstruction is the most common and distressing symptom. The severity of the symptoms varies widely from day to day depending chiefly on the amount of pollen in the air.

Seasonal asthma comes in attacks. The most popular theory concerning the mechanism of action is that the offending substance irritates the nerve endings in mucous membranes of the respiratory tract, causing spasmodic contraction of the small bronchioles of the lungs, which interferes with breathing, particularly with expiration. Non-seasonal allergic disturbances are sometimes attributed to house or street dusts, fungi, odors, animal dander, irritating gases, and heat or cold, particularly sudden temperature changes. It is often stated in the literature that heat regulation in asthmatic individuals is likely unstable, with a tendency toward the subnormal. Many allergic cases who are apparently well, develop their attacks when cold weather appears, or upon changing from warm to cool outdoor air.

Air Conditioning Apparatus

In recent years considerable effort has been directed toward the elimination of the principal cause of allergy from the air of enclosures by filtration or other air conditioning processes capable of removing pollens, in the hope of providing relief to individuals who fail to respond to medical treatment (desensitization or immunization).

Paper or cloth filters, mounted in inexpensive window or floor units, prove quite satisfactory, but since dust and smoke frequently cause asthmatic attacks, it is necessary that an air filter, to be of full value in the treatment of asthma, must remove all dusts and pollens regardless of size or amount. An electrostatic cleaner has proved extremely efficient

[34]A.S H.V E. Research Report No 1106—Air Conditioning in Industry, by W. L. Fleisher, A. E Stacey, Jr., F. C. Houghten and M. B Ferderber (A.S.H V E Transactions, Vol. 45, 1939, p 59)

in removing particles of 15 to 20 microns and smaller, besides dusts and smoke[35].

Although the chief remedial factor in the treatment by conditioned air is the filtration of pollen, a certain amount of cooling and dehumidification appears to be desirable. A comfortable temperature between 70 and 75 F and a relative humidity well below 50 per cent proved satisfactory[36]. Direct drafts, overcooling or overheating are apt to initiate or aggravate the symptoms.

Limitations of Air Conditioning Methods

The results obtained with air filtration or other air conditioning processes in the control of allergic conditions are fairly comparable to those obtained by desensitization treatment so long as the patients remain in the pollen free atmosphere. But while specific desensitization is preventive and in a few instances curative, for all practical purposes filtration gives only temporary relief. In mild cases sleeping in an air conditioned space may make it possible for the individual to pass more comfortable days. With rare exceptions, the symptoms recur on exposure to pollen laden air. Moreover the usefulness of air conditioning methods is limited because all cases are not caused by air-borne substances. Cases of bacterial asthma do not respond to the treatment with filtered air.

Despite these limitations air conditioning methods possess definite advantages in the simplicity of treatment, convenience, and under certain conditions almost immediate relief[37]. Pollen cases are usually relieved of most of their symptoms within 1 to 3 hr after exposure to properly filtered air. A pollen-free atmosphere is especially valuable in cases where desensitization has given little or no relief, and where desensitization is not advisable owing to intercurrent illness.

OXYGEN THERAPY

Oxygen therapy is the principal measure employed for preventing and relieving the distressing symptoms of anoxemia, which is a deficiency in the oxygen content of the blood. Some of the more important conditions in which oxygen treatment is believed to be beneficial are pneumonias, anemia, heart affections, post-operative pulmonary disturbances, certain mental disturbances, asphyxia, asthma and atelectasis in new-born infants. The effectiveness of oxygen therapy is not related to the type of administrative device except as the oxygen concentration is affected; that is, oxygen at 50 per cent concentration is equally effective whether given by any of the three common methods, catheter, face mask or tent[38].

The necessity of air conditioning in oxygen therapy arises from the fact that oxygen is too expensive a gas to waste in the ventilation of oxygen tents and oxygen chambers. The oxygen rich atmosphere in these enclosures is therefore reconditioned in a closed circuit by removal of excess heat, moisture, and carbon dioxide given off from the occupants being treated.

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[35]Air Cleaning as an Aid in the Treatment of Hay Fever and Bronchial Asthma, by Leo H Criep and M. A. Green (*Journal of Allergy*, 7:120, 1936).

[36]The Effect of Low Relative Humidity and Constant Temperature on Pollen Asthma, by B Z. Rappaport, T. Nelson and W. H. Welker (*Journal of Allergy*, 6:111, 1935).

[37]Hospital Air Conditioning, by C P. Yaglou (The Environment and Its Effect upon Man, *Harvard School of Public Health*, p. 244, 1939).

[38]Manual of Oxygen Therapy Techniques, by A. H. Andrews, Jr. (The Year Book Publishers, Inc , 1943).

Oxygen Tents

In oxygen tents the air enriched with oxygen is usually circulated by means of a small motor blower which sends the air over soda lime to remove carbon dioxide and then over ice to remove excess heat and moisture. The concentration of oxygen in the tent is regulated by means of a pressure reducing valve and flow meter. In an inadequately cooled tent, high temperatures and humidities are inevitable, increasing the discomfort of the patient and imposing an added strain on an already overburdened heart. Oxygen therapy under such conditions may do more harm than good. An ice melting rate of approximately 10 lb per hour gives satisfactory results in patients with fever in a medium size oxygen tent.

Oxygen tents are confining to the patient. They may terrify the restless and delirious patient. Medical and nursing care is complicated, as the tent must be opened or removed with attendant loss of oxygen. Oxygen concentrations of 50 per cent or more are difficult to maintain, and it is a problem to keep the temperature and humidity low enough in hot weather. However, with attention to details, the patient can be made quite comfortable.

Oxygen Chambers

The conventional oxygen chamber is an air-tight sheet metal enclosure of fire-proof construction, large enough to accommodate one or two patients. Trap doors or curtains are provided for the personnel, food and service, to avoid loss of oxygen. Glass windows in the ceiling and walls admit light from outside the chamber. The air conditioning system may be of the gravity type, or of the fan type using mechanical refrigeration or air drying agents.

The temperature and humidity requirement in oxygen therapy depends primarily upon the physical condition of the patient, and secondarily upon the type of disease. In pneumonias[39] prescribed conditions should be an effective temperature of 66 to 68 F, humidity of 50 per cent, air movement of not less than 50 linear feet per minute, oxygen concentration of 50 per cent, and carbon dioxide of less than one per cent.

Oxygen in Aviation

An important application of the principle of oxygen therapy is in aviation. At the present time all high altitude military airplanes in this country are provided with gaseous oxygen equipment and military personnel are required to utilize oxygen at all times while in flight above 15,000 ft, or between 12,000 to 15,000 ft for longer than two hours, or between 10,000 to 12,000 ft for longer than six hours. The use of oxygen in commercial aviation will depend on the height and duration of the flights as well as the state of health of the passengers. The necessity for portable, comfortable equipment, the possible fire hazards due to smoking, and the use of oxygen on sleeper planes are some of the difficulties facing civil airline operators. The pressure cabin airplane, capable of flights at 15,000 to 20,000 ft may be one of the solutions of the problem[40].

GENERAL HOSPITAL AIR CONDITIONING

Complete conditioning of a large hospital involves a capital investment and running expenses which may not be justified. Air conditioning has

[39] The Management of Pneumonia, by J. G. M. Bullowa, (*Oxford University Press*, p. 260, 1937).

[40] Principles and Practice of Aviation Medicine, by H. G. Armstrong (The Williams and Wilkins Company, 1939).

important applications in certain hospital wards such as nurseries for premature infants, oxygen therapy chambers, heat therapy rooms or cabinets, allergy wards[41], operating rooms and recovery wards. In clean and quiet districts, the requirements of almost all general and private wards during the cool season of the year can be satisfactorily fulfilled by the use of usual heating in conjunction with window air supply and gravity or mechanical exhaust. Insulation against heat and sound is much more important than humidification in winter; it will also help in keeping the building cool in warm weather. Excessive outside noise and dust may require the use of silencers and air filters in the window openings.

Cooling and dehumidification in warm weather are important. In new hospitals particularly, the desirability of cooling certain sections of the building should be given serious consideration. Financial reasons may preclude the cooling of the entire building, but the needs of the average hospital can be met by the use of built-in room coolers and a few portable units which can be wheeled from ward to ward when needed.

In the North and certain sections of the Pacific Coast, cooling is needed but a few days during summer, while in the South, it can be used to advantage from May to October, and in tropical climates almost continuously throughout the year.

Aside from comfort and recuperative power of the patients, cooling is of great assistance in the treatment of fevers in the new-born and in post-operative cases, in enteric disorders, fevers, heat stroke, heart failure, and in a variety of other ailments which often accompany summer heat waves.

Considerable research is in progress on the influence of air conditioning upon a wide variety of diseases such as pneumonia, upper respiratory diseases, tuberculosis, arthritis, nervous instability, hyper-thyroidism, essential hypertension, skin diseases, and vascular disorders.

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[41]Loc. Cit. Note 37.

CHAPTER 37

Transportation Air Conditioning

Railway Passenger Car Ventilation, Method of Air Distribution, Air Cleaning, Winter and Summer Air Conditioning, Humidity and Temperature Control, Summer Air Conditioning for Buses and Automobiles, Airplane Air Conditioning

THE principles of air conditioning used in connection with stationary applications such as stores, restaurants, hospitals, theaters, and homes are in general applicable to such mobile applications as railway passenger cars, passenger buses, automobiles, and ships. However, the equipment used for these mobile applications, differs from that used for stationary purposes in that it must meet additional requirements. Especially important are the features of compactness with the retention of ready accessibility for quick inspection and servicing, and low weight. Freedom from vibration which could be transmitted to the supporting vehicle and thus to the passengers is essential. (For Air Conditioning of Ships see Chapter 38.)

RAILWAY PASSENGER CAR VENTILATION

In non-air-conditioned cars, ventilation is accomplished by exhaust fans, roof ventilators, and open doors and windows. This practice provides an ample supply of outside air but does not prevent the entrance of smoke, cinders, and other dirt.

An average passenger car contains approximately 5000 cu ft of air and may seat as many as 80 passengers. The occupants are continually liberating heat, carbon dioxide, moisture, odors, and some organic matter from their breath, skin and clothing. The heat and moisture can be removed by cooling and dehumidification, but the other constituents can be successfully handled only by proper ventilation and air cleansing. In the average car from 2000 to 2500 cfm should be circulated by the air conditioning unit. Some of this air may be recirculated, but a portion of it should always be brought in from the outside. The amount of outside air required depends upon the type of car, number of passengers, air temperature, humidity, odors, and whether or not occupants are smoking, and will vary from 15 to 90 per cent of the total air circulated.

Careful attention must be exercised in specifying the rate of outside air taken in so as to fit the type of service adequately and yet not to supply more ventilation than is necessary. Conditioning this outside air is a major factor in determining the size of both summer and winter conditioning equipment. With present average ventilation requirements, about 30 per cent of the cooling equipment and sometimes as high as 50 per cent of the heating equipment is necessary to handle only the outside air load.

For normal conditions, 10 cfm of outside air per passenger is sufficient. When smoking is permitted, at least 15 cfm should be admitted. In some of the dining cars and de luxe sleeping cars, outside air rates as high as 20 and 30 cfm per occupant are used.

Method of Air Distribution

The fact that the amount of space devoted to railway passengers may be as low as 60 cu ft per person (ranging as high as 190 cu ft per person) and the high air flow rates made necessary by severe ventilation and sun

loads make the problems of air distribution and air delivery in railway cars critical.

Various methods may be used to distribute the air delivered to the interior of the car by the circulating fan or blower. The methods commonly used are:

1. A duct lengthwise along the center of the car.

2. One or two side ducts built on the outside of monitor-roofed cars, or on the inside of turtle-backed or arch-roofed cars.

3. A ceiling which is perforated for the emission of air into the car; the air being carried to the ceiling by an overhead duct. From one-half to the full area of the ceiling is used for the distribution of the air although practical considerations generally limit the available area to about two-thirds. As large a part of the ceiling area as possible should be used for air distribution.

Delivery grilles and plaques are used, and are often designed to give considerable entrainment and mixing to avoid cool drafts.

Smoking rooms present a special problem. The cloud of smoke that usually hangs near the ceiling can be broken up by having the incoming air directed along the ceiling in all directions at a velocity somewhat higher than that used for the rest of the car. The air should be exhausted from the room by a fan or through a grille to the washroom or lavatory, and then outside by a fan in a ventilator.

For compartments an adjustable supply duct outlet grille of suitable size and design should be provided and provisions made in the door or partition for the removal of the air to be recirculated.

Lower berths in sleeping cars and office cars should be provided with an adjustable air outlet which will discharge the amount of air desired at low velocity in any direction so that the occupant can regulate the ventilation to meet his own requirements.

In cars containing but one or two rooms or compartments, satisfactory results may be obtained by discharging the air directly from the conditioning unit into the upper part of the car. Care must be taken to have a proper discharge velocity. If the velocity is too low, the air will drop before reaching the end of the car and if too high it will discharge against the end bulkhead and be reflected back. Care must be exercised to secure proper circulation, otherwise objectionable drafts will be experienced.

The recirculating air grilles are usually of the straight flow type, and should be located so that objectionable drafts will not be created by the return air. The outside air intakes, located in the car vestibule, on the side of the car, or on the roof of the car, depending upon the location of the cooling coils, should be of ample size to permit the entrance of sufficient outside air. On many of the recently air-conditioned cars, there are no dampers or shutters at the outside air intakes, the percentage of outside air being controlled by adjusting the flow through the recirculating grille.

Air Cleaning

All of the air circulated by the blower is filtered before passing over the cooling coils. In some cars the outside and recirculated air is filtered separately before mixing, while in others the air from the two sources is mixed before passing through a common filter. Filters in use are made of metal, wool, cloth, spun glass, hemp, paper, hair, and wire screen. Most filters have a viscous coating of oil for greater cleaning efficiency. Some types may be cleaned, retreated, and returned to service while other types are discarded when dirty.

RAILWAY PASSENGER CAR WINTER AIR CONDITIONING

The majority of cars in service use steam from the locomotive or from a head-end, oil-fired boiler as a source of energy for winter heating. In some instances electrical energy from either a head-end generating set or motive power supply is utilized for resistance heating. In still other cases electrical energy and waste heat from individual car engine-generator sets are employed. The peak heating loads which depend largely upon the amount of insulation used in the car, the type of windows (whether single or double glazed), and the ventilation rate, may vary from 150,000 to 250,000 Btu per hour.

In order to temper the cold outside air, about 30 to 50 per cent of the total heat energy required is distributed by means of finned coils or resistance heaters located in the outside air duct. The remainder is usually transmitted to the car air by finned tubing located along the sides of the car near the floor, thus preventing cold convection currents falling from the car windows from reaching the feet of the passengers.

RAILWAY PASSENGER CAR SUMMER AIR CONDITIONING

Three general types of cooling or refrigerating equipment are being used in 13,362 (May, 1944) air conditioned railway cars and 31 air conditioned rail motor cars in the United States. Of these, 30 per cent are ice activated, 15 per cent use steam jet systems, and 50 per cent employ mechanical compression schemes. These systems which functionally are identical with those used for stationary applications (see Chapter 24) are modified in design to meet the requirements of mobile service. Contrasted with stationary applications of summer conditioning equipment, the use of water as a final means of heat disposal from condensers cannot be resorted to because water in such quantities cannot be transported economically. Accordingly, air cooled or evaporative condensers are always used, with the result that mobile cooling equipments operate at higher temperature, pressure, and power requirement levels than stationary equipment.

The maximum cooling and dehumidifying load which depends largely upon the amount of insulation, the type of windows, the ventilation rate, the sun intensity, and the number of passengers may vary from 60,000 to 96,000 Btu per hour.

An average ice-activated system for such capacities uses about 500 lb of ice and 1.2 kw per hour. The increase in car weight due to such a system is approximately 8500 lb.

The same service from a steam jet system is obtained with the expenditure of 180 lb of steam and 3.3 kw per hour, with an added weight per car of 8000 lb.

The mechanical compression systems, all of which use dichlorodifluoromethane as a refrigerant, may be classified by several types depending on the method of driving the compressor. The source of power for driving the compressor (approximately 10 hp) is complicated by the necessity of obtaining this power at all times whether the car is in motion or standing still on the right-of-way or in a terminal where auxiliary power plug-ins are available. In those cases where compressors are driven from car axles, additional refinements in the drive are necessary in order that a nearly constant cooling capacity may be obtained from a variable speed power source. Numerous combinations of electrical generating schemes for generating sufficient electrical energy from the car axle for lighting, ventilation, and summer air conditioning are in use, and their operation

is closely interlocked with compressor demands, need for pre-cooling, battery charging, etc. It is difficult therefore to state the additional weight imposed on a car because of such a compression air conditioning system, but it is probably in the vicinity of 6000 to 8000 lb. These systems, depending mostly upon the locomotive for supplying power for operation, impose a load, including the power required to pull the weight added by the equipment, of from 5 to 10 per cent of the total locomotive power.

Several schemes for relieving the locomotive of this compression load are used. Some of the articulated trains, which run as unit equipment—the same cars always in the same train—employ a head-end, engine-generator combination for supplying power to compressor motors. In other cases, especially in many of the new streamlined trains, propane fueled engine-driven generators on individual cars are used to supply power to motors, as well as to supply all power for car lighting and accessories. Gas engine compressor combinations on individual cars provide attractive low weight equipment where automatic engine operation is permissible under all circumstances. Diesel-powered generator units have been used experimentally on individual cars for supplying electrical energy and in some cases waste engine heat has been used either for modulating refrigeration with a reheat cycle or for car heating purposes.

RAILWAY PASSENGER CAR HUMIDITY AND TEMPERATURE CONTROL

The temperature to be maintained in a car depends upon the outside temperature and the humidity desired inside the car. With a low humidity it is necessary to maintain a higher temperature to establish a desirable comfort condition. Little humidity control has been attempted on cars up to the present time. A certain degree of automatic humidity control is secured with cooling, but the relative humidity obtained depends largely upon the temperature of the evaporator, which should be below the dew-point temperature of the air. With certain outside atmospheric conditions it may not be possible to operate the conventional equipment with a sufficiently low evaporator temperature to reduce the humidity without dropping the temperature too low. One method has been developed whereby the evaporator temperature is carried below the dew-point a sufficient amount to insure dehumidification and then the cold air is heated to the proper temperature by passing it over coils through which part of the high temperature liquid from the condenser is by-passed. Such a system is costly and has not been generally applied. The reheat cycle obtainable from waste engine heat may be used to good advantage in reducing the humidity without reducing the dry-bulb temperature.

During the heating season humidification is desirable from a comfort standpoint, but unless properly controlled, condensation will appear on the windows. A steam or water spray controlled by a humidistat will provide the necessary moisture for humidification. There are several cars with this feature now in use.

Temperature control for the most part obtained by rugged thermostats and relays capable of withstanding vibrations attendant with mobile service is usual equipment.

Manual zone control for varying outdoor conditions, as well as controls which regulate the car temperature automatically in accordance with outdoor conditions, are employed.

Simplified controls from the standpoint of operation by train crews and

especially from the servicing viewpoint are very desirable. The control of summer temperatures is accomplished mainly by cycling the complete cooling system; however, modulation is being effected by using multiple evaporators in which a fixed portion may automatically be cut out of operation to suit the cooling capacity requirements and to keep the equipment in operation for longer periods.

With motor or engine driven compression equipment modulation of compressor capacity to suit the reduced evaporator capacity is accomplished by changing motor or engine speed, or by varying the number of compressor cylinders in operation.

Steam ejector equipment is provided with dampers which by-pass the air around the cooling coil during the off cycle to prevent re-evaporation of moisture from the coil during the off period.

For further information on controls, see Chapter 33.

PASSENGER BUS SUMMER AIR CONDITIONING AND VENTILATION

The highways in the United States are now traveled by about 3500 summer air-conditioned passenger buses. Many of the facts stressed in connection with the design and installation of summer conditioning equipment in railway cars are even more important in these newer vehicles. Weight and space limitations are more stringent, and the problem of circulating from 900 to 1200 cfm of air in coaches carrying from 25 to 40 passengers with about 35 cu ft of space per passenger without drafts is no easy one.

Some bulkhead delivery systems have been used, and while the overhead package racks have served to break up drafts to some extent, these installations are not gaining in popularity. Longitudinal ducts in the corners above the package racks are sometimes used to carry conditioned air to a series of outlet louvers along the top of the windows. Other designs provide for false spaces below the package racks which serve as ducts to distribute air to either entrainment grilles in the bottom of the racks or distributing slots at the edges of the package racks. Some coaches employ a false ceiling to provide a duct, with delivery taking place from numerous perforations in the ceiling.

Return air grilles and filters are usually located near the rear ceiling where the evaporator is placed. Outside air intakes and filters are located preferably near the front of the vehicle so as not to contaminate this supply with exhaust fumes and road dust. Of the 30 cfm circulated per person, about 8 to 10 cfm are outside air and the remainder is recirculated. Power for the motor driving the centrifugal fans is obtained from the bus battery.

More recently a coach design has been brought out which provides for a number of return air outlets below the seats; these permit return air to enter a longitudinal duct below the floor. The filters and evaporator are located in this duct near the front of the vehicle. In this instance a central heating coil utilizing waste heat from the coach engine is also located in this duct. Conditioned air is delivered through a pair of vertical ducts to a package rack distribution scheme.

Summer conditioning systems for these vehicles range in cooling capacity from 36,000 to 48,000 Btu per hour. Mechanical compression systems using dichlorodifluoromethane are used, and are powered by water cooled, gasoline engines of approximately 14 hp.

Complete systems add from 800 to 1300 lb to the weight of a coach.

Sometimes an auxiliary generator driven by the air conditioning engine is used and serves to help charge the bus battery, thereby offsetting the power drain imposed by the ventilating blower. Belted reciprocating compressors and direct driven V-type and rotary compressors are used, with engine speeds up to about 1800 rpm. Air cooled condensers for this service require about 5000 cfm of outdoor air, and this is provided by either centrifugal or propellor type fans belted or direct driven by the air conditioning engine. Preventing noise and vibration from affecting passengers is of vital importance. Installations must be made so that quick daily servicing of the engine is possible. In all cases fuel is obtained from the main bus tanks, and in some cases the main engine jacket water cooling system is used to cool the air conditioning engine.

In the de luxe equipment, after the driver has started the air conditioning engine by means of its own cranking motor, the engine speed is modulated automatically as the refrigeration demand is partially met, and if this demand is then fully met, the engine is stopped thermostatically. Restarting when the cooling thermostat is no longer satisfied is accomplished either automatically or manually. The various protective and automatic devices on the refrigerant and engine systems make some of the bus air conditioning control systems quite complicated.

AUTOMOBILE SUMMER AIR CONDITIONING

Recently summer air conditioning has been applied to automobiles. The average present day automobile with little insulation, large, single glazed window areas, and high infiltration and exfiltration losses requires about 15,000 Btu per hour of cooling capacity. One system utilizes a reciprocating compressor belted from the main engine fan shaft thus operating at varying speeds up to 3000 rpm. The resulting refrigeration capacity varies from about 6000 Btu per hour at idling speed to 24,000 Btu per hour at maximum car speed.

A dry air condenser is placed in front of the engine radiator, and the liquid and suction refrigerant lines run back under the car floor to the evaporator which is located in back of the rear seat. Conditioned air is delivered into the car just above the shelf near the back of the rear seat. A return grille is provided under the rear seat, and the recirculated air is filtered. Outdoor air is provided by infiltration. Power for the air circulating blowers is obtained from the car storage battery. Equipment of this nature increases the car weight approximately 200 lb.

AIRPLANE AIR CONDITIONING

Air conditioning of airplanes is closely associated with heating and ventilation of airplanes, but air conditioning of planes in flight has received little attention to date, probably because of added weight of equipment necessary. Under ordinary conditions the cleanliness and humidity desired are fixed by natural conditions and therefore the supply of heat and temperature control become the chief consideration. Passenger planes and military planes even at low level flying are at altitudes which require no cooling and at extreme altitudes the design temperature of the atmosphere is 60 deg below zero.

In the operation of internal combustion engines the heating value of fuel appears approximately, one-third as power, one-third transferred to cooling fluid, and one-third in the exhaust gases.

Ordinarily, the heat in the exhaust gases is ample for all heating needs, despite the excessive transmission loss due to high speed, low tempera-

tures, and light wall construction. The maximum coefficient of heat transmission without insulation is found to be 2.28. The minimum coefficient with insulation is 0.33. For present day construction a value of 0.56 may be assumed. Naturally, the waste heat in the exhaust gases offers an attractive source for designers of heating equipment, who in the course of design have followed closely the development of heating as applied to land structures. In first attempts heat was obtained from an annular space surrounding the exhaust pipe, from which branches conducted the heated air to grilles or to seats of passengers. The speed of the plane eliminated the need for a fan or pump. In present practice danger of carbon monoxide contamination is avoided by the use of both primary and secondary heat exchangers.

Heating systems using hot air are in use for passenger and military planes. Steam systems have been used with boilers located in the path of exhaust gases but difficulty experienced in preventing freezing of the water has apparently rendered this system obsolete. A similar system using glycol is in use and requires special care because of the peculiar qualities of glycol. Thermostatic control, mixing dampers and air distribution vary little from standard practice. Controlling devices are built to withstand the extensive vibrations experienced. Humidity determinations under flying conditions show need for humidity control. Passenger planes or other planes requiring air conditioning when on the ground are satisfactorily serviced by portable air conditioning machines. High flying airplanes—above an elevation of 14,000 ft—require oxygen supply or pressurized cabins. The compression of the outside air by the supercharger raises the temperature of the cabin air from an extremely low to a comfortable high. The compression of the air also increases the humidity. Cleanliness at flying altitude is not a consideration, except under special conditions such as above deserts or upon encountering dust storms. In small planes carrying several passengers or crew, localized heating and ventilation are provided. Electrically heated clothing and oxygen masks meet requirements. At high altitudes pressurized cabins require no special oxygen supply.

Medium bombers at one time were equipped with separately fired heating units, but larger bombers, to meet the heating load imposed by crew and equipment, demand too much heat to be supplied in this manner. The recovery of heat from waste gases is therefore imperative. In airplane operations light weight is a more important requirement than in any other type of transportation. Noise should be reduced to a minimum and combined use may be made of material for insulating and noise reduction.

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CHAPTER 38

Marine Heating and Ventilation

General Considerations, Ship Construction Features, Factors Affecting Design, Requirement for Various Types of Space, Ship Insulation

THE importance of adequate shipboard heating and ventilation arrangements cannot be overemphasized. Installations must not only keep the passengers comfortable and the operating personnel physically fit and mentally capable of maintaining the ship as a profitable business investment, but must also satisfy the human element. The provision of satisfactory living and working conditions is one of the most economical means of keeping a high morale.

GENERAL CONSIDERATIONS

Unlike a shore establishment in connection with which personnel obtain their food, rest and recreation from outside sources, a ship must be self-sufficient and provide for all human needs. Accordingly, every vessel is a complete, independent community comprising, at least to some degree, the facilities available in the average urban development. These facilities must be contained in the minimum practicable space and weight to conserve dead weight and increase the pay load. The pay load may be expressed in terms of cargo carrying ability, passenger carrying capacity, fighting strength, or towing ability.

An atmosphere compatible with efficient operation must be maintained in working and machinery spaces, and temperatures must be such that materials or equipment will not be damaged. Conditions in living spaces, whether ashore or afloat, must permit adequate rest and comfort of occupants. Passenger accommodations must be treated to provide service and living conditions similar to those afforded by the various classes of hotels. Many of the expedients used ashore for this purpose are not applicable afloat. For instance, all living quarters aboard ship cannot be located at a distance from the power plant; but must often have boundaries in common with heat producing spaces. The thermal conductivity of shipbuilding materials such as steel, copper, brass, etc., is many times the value for building materials used ashore, and this, together with concentrated arrangement of equipment, produces a difficult heat transfer and insulation problem. Furthermore, the use of portholes, windows, skylights, and similar openings is greatly restricted in marine applications because of the necessity for strength, water-tightness and protection from the sea in foul weather. During wartime, the utility of portholes, windows, and skylights is greatly restricted because they must be fitted with light-excluding devices.

Much of the cooling and ventilation that is accomplished by natural means ashore must be handled by mechanical means afloat. Space does not exist for the large ventilation trunks required for natural ventilation. Also, because of the restricted space and shipbuilding materials the heat quantities that must be dissipated by ventilation air are often greater than would be required for a similar shore installation. Mechanical ventilation is not merely desirable, but is absolutely necessary in many shipboard spaces to enable the crew to operate the vessel. Furthermore there are numerous spaces which must be ventilated to prevent the accumula-

tion of objectionable, combustible and toxic gases. Shipboard equipment must also be reliable inasmuch as specialized servicing facilities are not available at sea, and failure during an emergency may jeopardize the vessel's safety. Experience has demonstrated that *simple* and *foolproof* heating and ventilating arrangements are essential for satisfactory service. Ventilation systems on ships must be designed to operate in heavy weather with little or no attention, and to resist the efforts of unauthorized personnel to readjust the heat or air distribution. Also, they must be laid out in such a manner that the watertight and fireproof integrity of the vessel is not impaired.

Contrary to the impression among some engineers and designers, that there is a basic difference between marine and shore installations in the application of heating, ventilating and air conditioning arrangements, the same fundamentals apply to both shore and shipboard applications. The air capacity handled by a system ashore will equal the capacity that an identical system will deliver afloat. The same laws of heat transfer apply, and the same heat balance must be maintained in the human body. The only variation between ship and shore applications is that emphasis is placed on different practical aspects of the design.

SHIP CONSTRUCTION FEATURES

The seaworthiness of the vessel is increased by subdividing the main watertight hull space into a number of watertight compartments. This is usually done by providing transverse watertight structures (bulkheads) so located along the ship's length that the vessel will not sink even when the shell is pierced in one or more watertight compartments, by collision, torpedoing, or other causes. It is obvious that the watertightness of these bulkheads is very important, particularly under war conditions. Every effort is made to lead ventilation ducts so that they do not pass through such structures. Where openings must be cut for ventilation or other purposes, special watertight closures are fitted to protect the watertightness of the bulkheads.

In addition to the obstructions caused by the main transverse bulkheads, the ship's interior is fitted with decks and flats (partial decks) that are spaced to suit the use to which the space is put. The deck height (floor height) in cargo spaces generally varies from 9 to 30 ft. This height in living quarters varies from 7½ to 9 ft. Usually in main machinery spaces, the entire depth is subdivided only by a few flats, with gratings provided where necessary to service the equipment.

Bulkheads, shell, and decks are designed to suit structural as well as watertight requirements of the vessel, and consequently, the designer must always contend with beams, stiffeners, stanchions and brackets.

Fig. 1 indicates the arrangement of a typical cargo vessel, and shows the main transverse watertight bulkheads, decks, structural members, and many other pertinent characteristics. Living quarters are usually located on and above the weather deck, this custom being based on the assumption that better natural air and light are available in this location. It is obvious that the upper 'tween deck and other spaces can be conveniently used for additional living and working spaces, but that natural ventilation is far less effective in this case.

The foregoing description mentions some of the limits and natural boundaries which affect the design and layout of ship ventilation and heating systems. It should be added that there are also equipment, wireways, and miscellaneous piping which must be considered. Since

there is very little unassigned space, fan rooms are usually quite congested and many studies must be made to ascertain the arrangement of equipment which will best suit the conditions. Cuts in structural members must be limited in number and size. Lack of headroom often causes poor aspect ratios, inaccessibility, and other undesirable conditions. It therefore follows that the design of a ship's heating and ventilating system is far more than a theoretical calculation. A good designer must be familiar with all phases of ship construction and operation in order to provide sufficient and efficient ventilation and heating, and to locate the equipment where it will interfere least with basic arrangements, major equipment, stability, and other important factors.

FACTORS AFFECTING DESIGN

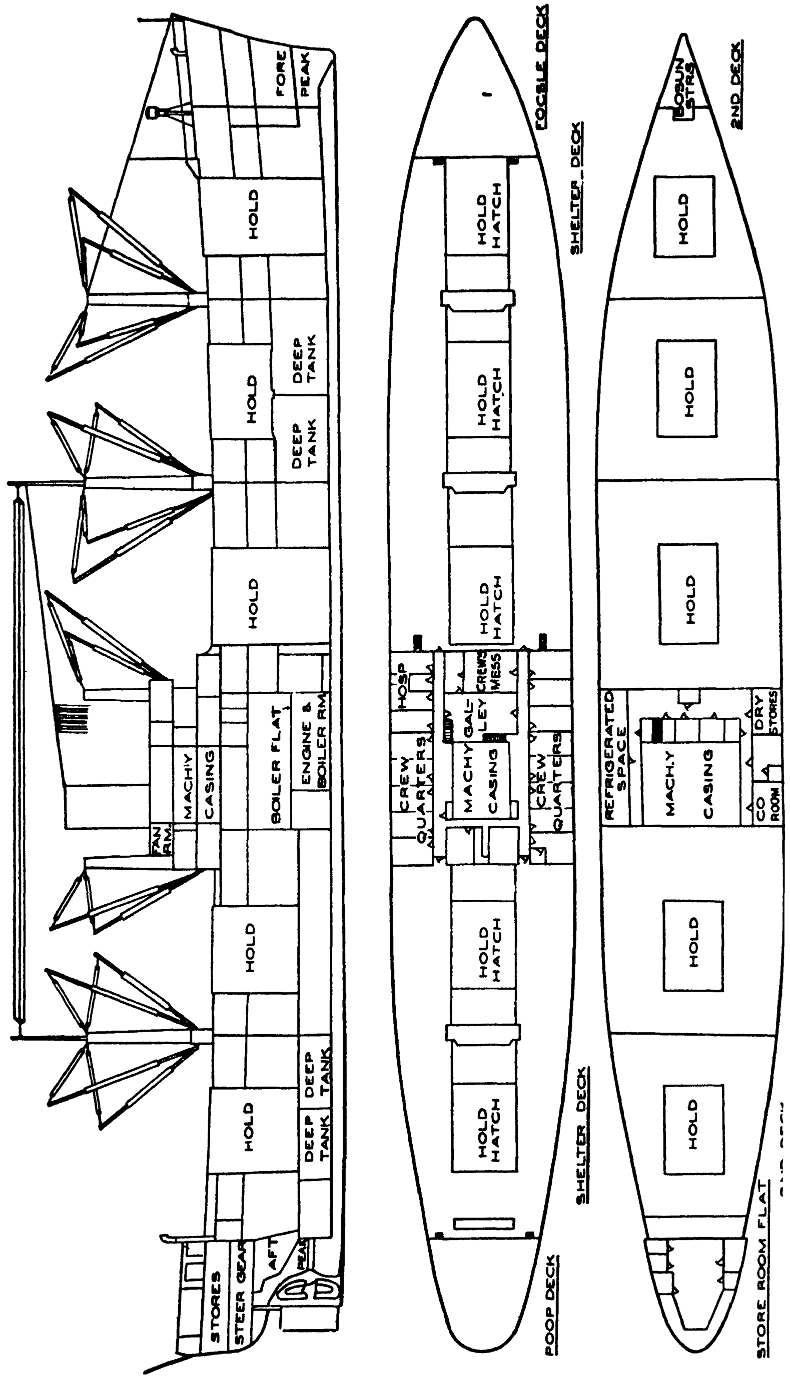
A ship may be favorably compared to a *floating hotel.* Its make-up includes living accommodations, dining rooms, shops, power plants, commissary spaces, etc., all of which must be so arranged and maintained as to make it self-sustaining for long periods of time. In contrast to a hotel, however, a ship may, on a single cruise, move from arctic to tropical climates. On the other hand, it may be built to operate only in one single locality. It is therefore necessary to assume maximum and minimum design air temperature condition based on the intended use of the vessel so that adequate cooling and heating may be provided. For vessels designed to operate generally in all parts of the world, a maximum summer temperature may be taken as 88 F and a minimum winter temperature as 10 F. For other ships, those which spend long periods in port or in inland waters, the maximum and minimum temperatures may be taken as 90 and 0 F respectively.

Preliminary design of heating and ventilating installations aboard ship is simplified by the uniformity of the conditions which apply to all ships. Among such conditions are (1) the necessity for the vessel to supply its own power, (2) the availability of an unlimited supply of sea water, (3) the drastic restriction of shipbuilding materials because of strength requirements, corrosion resistance and fire hazard, and (4) the limitation of available space and permissible weight. On the other hand, shipboard space, weight, and power limitations require precise layouts with minimum design safety factors, and this necessitates unusual attention to design details. Particular attention should be given also to the adjustment of a system after installation, because designed air quantities are usually close to the minimum acceptable and therefore, improper balance will result in unsatisfactory conditions in some spaces.

Each space aboard ship must be treated in accordance with the particular requirements of its use. The ventilation of most spaces is determined by empirical calculations. The quantity of air supplied must be not less than that which satisfies (1) maximum allowable temperature rise, (2) minimum allowable fresh air per person, and (3) permissible air change, named in order of importance.

The present practice of various ship designers may be to supply the quantity of air which satisfies only one of these requirements, but the most satisfactory ship ventilation will result from the use of that quantity which is large enough to satisfy all three.

When heating and cooling loads are estimated, the cooling effect of the water through which the vessel is sailing must be considered if the space is at or below the waterline. Maximum and minimum water temperatures of 85 and 35 F are usually used for such calculations. This factor is


FOC'SLE DECK
SHELTER DECK
2ND DECK
POOP DECK
SHELTER DECK
STORE ROOM FLAT
FORE PEAK
HOLD
DEEP TANK
DEEP TANK
HOLD
FAN RM.
MACH'Y CASING
BOILER FLAT
ENGINE & BOILER RM
HOLD
HOLD
DEEP TANK
DEEP TANK
STORES
STEER GEAR
AFT PEAK
HOLD HATCH
HOLD HATCH
HOLD HATCH
HOSP
CREWS MESS
GAL-LEY
CREW QUARTERS
MACHY CASING
CREW QUARTERS
HOLD HATCH
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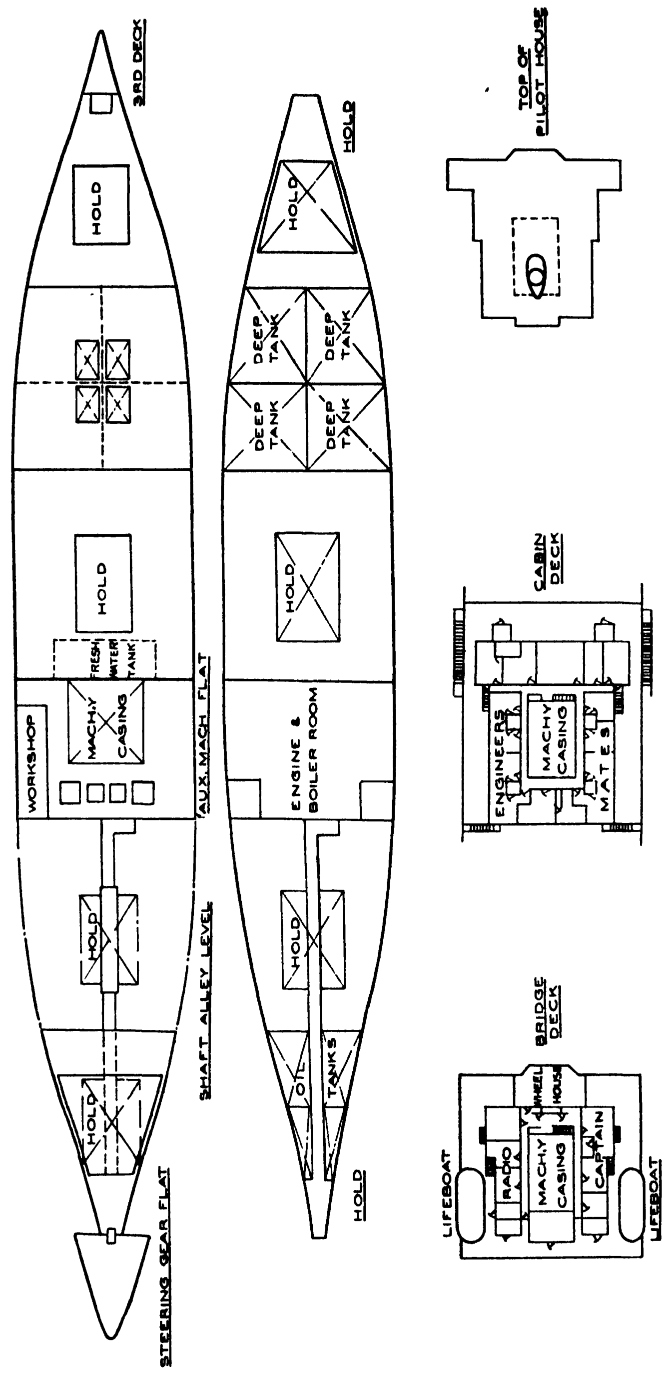

FIG. 1. ARRANGEMENT OF A TYPICAL CARGO VESSEL

important enough to justify the omission of ventilation from certain spaces which have only small heat gains and do not require fresh air for other purposes.

The relation between supply and exhaust ventilation depends on the type of space, and the space's relation to the weather. In any case, it should be remembered that positive means for supply and exhaust must always be provided. Where natural ventilation is anticipated, infiltration and exfiltration via open doors, windows, etc., cannot always be considered adequate for this purpose.

In addition to the ventilation requirements, but closely related to and usually forming an integral part of the study, is the manner in which heating shall be furnished for cold weather conditions. In all spaces which are provided with mechanical supply ventilation, where heat is also required, the simpler method and most economical from standpoint of weight is that in which duct type or blast heaters are fitted in the ventilating systems. Where this type of heating is furnished, convectors or radiators should be provided for spaces requiring heat but not furnished with mechanical supply ventilation. The most satisfactory arrangement for this method of heating, except where individual space control is essential, is zone heating. To accomplish this, the ventilation supply system, with the fan operating at slow speed (usually about 50 to 70 per cent delivery) passes the air over a preheater which raises the air to a temperature of about 40 to 60 F. This preheated air is then passed through the system and thence through one or more reheaters which are located in the ducts and so arranged that each serves a group of spaces which possess the same or similar heating requirements. It may be necessary to slightly revise the individual space air quantities, which have been determined on the basis of summer cooling, in order to bring them in line with the reheat or delivery temperature. In some cases central heating is used. By this arrangement all the air needed for heating is passed through the preheater and reheater fitted in series. This system has the disadvantage of affording poorer thermostatic control and cruder heating properties. In all installations of duct type heaters, however, care must be taken in the first stage of heating (preheater) to provide protection against freezing, either by heater design or compensating automatic valve control. Of course, many of the usual combinations or variations of duct type and convection or radiant heaters may be used with varying degrees of satisfaction, depending on basic design requirements such as weight and space limitations and economic justification for refinements.

REQUIREMENT FOR VARIOUS TYPES OF SPACE

In the paragraphs which follow the most important considerations are given for the treatment of the various spaces, but the resulting quantities of air should in all cases be checked with those shown in Table 1.

Machinery Spaces

The prime purpose of machinery space ventilation is to maintain a habitable temperature for the operating personnel. Of secondary importance is the necessity for preserving temperature conditions satisfactory for the successful operation of machinery, particularly of that designed for certain maximum ambient temperatures. High dissipation of heat, from high temperature steam equipment or from high powered electrical installations, make it more practicable to use *spot cooling* of working

TABLE 1. TYPICAL HEATING AND VENTILATION PRACTICES FOR SHIPS

SPACE	TYPE OF VENTILATION		TYPE OF HEATING	AIR CHANGE MINUTES	REMARKS
	Sup.	Exh.			
Crews Quarters	Mech.	Mech. or Nat.	H.B.[a] or D.R.[b]	5-6	
Toilets and Showers	Nat.	Mech.	D.R.	3-4	
Chart House	Mech.	Mech. or Nat.	H.B. or D.R.	5-6	
Passenger & Officers Qtrs.	Mech.	Mech. or Nat.	H.B. or D.R.	4-6	
Office & Similar Spaces	Mech.	Mech. or Nat.	H.B. or D.R.	5-6	
Mess Rooms	Mech.	Mech. or Nat.	H.B. or D.R.	4-6	May be exhausted through Galleys and Pantries.
Dining Salons	Mech.	Mech. or Nat.	H.B. or D.R.	3-6	May be exhausted through Galleys and Pantries.
Galley and Pantry	Mech. & Nat.	Mech.	H.B.	1-2	Tempered air Sup.
Deck Pantries (No cooking Equipment)	Mech. & Nat.	Mech.	H.B. or D.R.	3-6	
Stores Spaces (Adjacent to Living Quarters)	Mech. or Nat.	Nat. or Mech.	....	15-20	
Stores Spaces (Isolated Locations)	Mech. or Nat.	Nat. or Mech.	....	15-30	
Hospital Spaces	Mech.	Mech.	H.B. or D.R.	4	
Soiled Linens and Oilskin Lockers	Nat.	Mech.	....	15-20	
Shops	Mech.	Mech. or Nat.	H.B. or D.R.	6-10	
Group Berthing Spaces	Mech.	Mech. or Nat.	H.B. or D.R.	3-6	
CO_2 Bottle Space	Nat.	Mech.	....	10	
Battery Room	Nat.	Mech.	D.R.	2	
Laundry	Mech.	Mech.	H.B.	1-4	Tempered air Sup.
Dry Cargo Holds	Mech. or Nat.	Nat. or Mech.	....	20-30	
Stearing Gear Space	Mech. or Nat.	Mech. or Nat.	H.B. or D.R.	2-6	Unit Heater if desired.
Dry Stores	Mech.	Nat. or Mech.	....	6-10	
Gyro Room	Mech.	Nat. or Mech.	....	3-6	
Theaters and Halls	Mech.	Nat. or Mech.	H.B. or D.R.	6-8	
Passageways	Nat.	Nat.	D.R.	....	Exhausting or Supplying adjacent Quarters.
Engine & Boiler Rooms	Mech. or Nat.	Mech. or Nat.	....	1-2	
Misc. Machinery	Mech. or Nat.	Mech. or Nat.	....	2-4	
Resistor Spaces	Nat.	Mech.	....	10	
Stair Wells	Nat.	Nat.	D.R.	....	Same as passageways. Exh. at top.

[a]H. B.—Hot Blast System.

[b]D. R.—Direct Radiation.

areas, rather than to attempt to obtain uniform ambient temperature. The permissible temperature rise at working stations is usually about 15 F while the overall temperature rise is usually between 30 and 50 F.

When the necessary quantity of air has been determined, the ventilation arrangements should be designed so that all of this air will be distributed to locations where the personnel are to be stationed under operating conditions, thus giving them the maximum benefit from the cool air so introduced. This consideration is of vital importance because it must be remembered that the physical well being and alertness of the crew are more important than the operating ambients of the machinery; and by such installation the personnel receives the air before it becomes heated to the temperatures satisfactory for the equipment.

In addition to the air supply, these spaces must be exhausted, preferably by mechanical means. Every attempt should be made to remove air at or close to heat sources so that the mixing of ambient air with that which has been heated to a higher temperature will be at a minimum. The quantity or capacity of the mechanical exhaust systems should take into consideration the expansion of the supply air because of heating and should be also sufficient to insure an indraft through access openings to the space. Generally, in order to accomplish this, the quantity of mechanical exhaust should be about 120 per cent of the supply. In many installations, the combustion air for diesel engines and boilers is taken directly from the machinery space and serves as a partial mechanical exhaust system.

Normally, heat is not required for machinery spaces except for those fitted with electrically operated equipment, which may remain inactive during periods while in port. Such spaces should be equipped so as to heat them to about 50 F during such periods.

Living Spaces

The minimum quantity of air required for any space which is fitted for sleeping or office work, including hospital spaces, should be that which will limit the temperature rise over the outside air conditions to not more than 10 F (a rise of 7 F is more satisfactory if practicable), or a minimum of 30 cfm per person, whichever is the greater. For spaces fitted for eating, recreation, or manual work the rise may be taken at 10 F with not less than 20 cfm per person.

It will be noted that the quantities thus obtained are minimum and should be supplied by mechanical means. This is especially important if the spaces are located inside of the ship and have no direct weather exposures with ample openings, such as portholes or windows, which may remain open even in inclement weather. The same requirement applies to the necessity for mechanical exhaust, although natural exhaust may be used where only a short run of duct is required.

If the air to these spaces is supplied by mechanical ventilation it should enter through registers or diffusing type terminals. The installation of bracket fans within these compartments at proper locations and of sufficient number will aid materially in providing proper distribution for comfort.

Heat should be furnished to these spaces to maintain the following temperatures when operating in the minimum winter conditions:

Staterooms, Berthing, Messing and Office spaces	70 F
Working spaces and Shops	60 F
Hospital spaces	75 to 78 F

Toilets, Washrooms, Showers and Baths

Spaces for this purpose should be fitted with mechanical exhaust ventilation for odor and steam removal. They should have one or more well-located exhaust terminals overhead; one will usually suffice for the average size space. The supply may be taken naturally. Generally, the surrounding living or working spaces are exhausted through them. The minimum total quantity exhausted should be that which would change the air within the compartment in 6 min, plus 25 cfm for each fixture, plus 50 cfm for each shower head. Air requirements on merchant vessels are commonly estimated on the basis of a 4-min rate of change.

Heating should be obtained by use of convection-type heaters which should maintain 70 to 80 F for bathing or washing spaces and 70 F for toilet spaces when the outside winter temperature is at the design condition.

Galleys, Bakeries, and Food Handling Spaces

The problem of the ventilation of spaces fitted for cooking and food preparation is primarily one of heat and smoke or fume removal. Mechanical exhaust is always required for this purpose. The supply may be natural, but the required exhaust capacity will be most satisfactory if arranged for 50 per cent mechanical and 50 per cent natural.

The exhaust quantities are generally large and may be predicated on restricting the ambient temperature rise to about 15 F over the outside summer design air conditions. It will generally be found that the resulting quantity will change the air in these spaces in about 3/8 to 1 min, depending on concentration of equipment. All of the exhaust should be arranged so as to remove air from the space through hoods fitted over the cooking equipment and these hoods should contain grease filters over ranges and griddles and should be made readily accessible for frequent cleaning.

The mechanical supply when fitted should blow air directly on the personnel but away from the equipment and so as not to interfere with the flow of exhaust air to the hoods.

Generally, no heat is required for these spaces in winter except that the supply air temperature should not be too low. Usually a preheater delivering air at about 45 to 60 F should be provided.

Laundries

The problems in the ventilation of laundries are somewhat similar to those for galleys. Heat removal is necessary and best accomplished by the installation of mechanical exhaust through hoods fitted over the heat-producing equipment.

The supply for these spaces may be all mechanical, or part mechanical and part natural. The mechanical supply should be distributed through adjustable blast-type terminals at relatively high velocities directed to blow cool air on the torsos of the operating personnel. Experience indicates that this quantity should be sufficient to change the air in the spaces in from 1 to 4 min.

The exhaust should be at least one air change per minute and at least equal to 120 per cent of the total supply so as to insure an indraft of air through the access openings to the space. All exhaust openings within the space should be fitted to be readily accessible for frequent cleaning and lint removal.

No winter heating is required except that the supply air should be delivered at about 40 to 60 F.

Storerooms and Cargo Spaces

The ventilation of these spaces should be predicated upon the kind and type of stores or cargo to be carried.

For materials which would not be adversely affected by summer temperatures, no ventilation is required. Also, storerooms or cargo spaces below the water line in which temperatures would not normally exceed 100 F with the sea water assumed as 85 F maximum, may not require ventilation for certain cargoes. As spaces in these two categories are frequently damp some means of moisture removal must be provided. Chemical dessicants are satisfactory where it is essential to prohibit openings through watertight structure. In other cases a supply of dry air is provided from a central silica gel dehumidification system and distributing ducts, with recirculation used to accelerate the drying process.

Where storage spaces must be ventilated to obtain a change of air in about 15 to 30 min, in some cases the ventilation is determined by the maximum temperature which the cargo or stores can withstand without damage.

Spaces in which inflammable liquids are carried, or where inflammable vapors may be generated, require special consideration. They should be fitted with mechanical exhaust with terminals so located as to remove explosive or combustible vapors. The supply to these spaces may be natural and arranged so that good distribution, free from pockets, is assured.

If stores and cargo, which require some special and constant temperature or humidity control, are carried, special air conditioning equipment must be installed to suit the particular requirements.

SHIP INSULATION

In order to properly limit one of the major ventilation heat loads which is that made necessary by heat transmission, and to prevent condensation, it is necessary to use insulation judiciously. The principal sources of heat in a ship are the power plant and sun load. The confinement or exclusion of this heat in the structure of a ship is not easy, principally because of the complex structural nature of the beams, stiffeners, bulkheads, decks and hull. The continuous metal paths offer easy means of heat flow throughout the structure.

Insulation like any other component hull part of the ship cannot be used indiscriminately because of weight and space limitations. Therefore higher heat transmission coefficients are accepted for insulated structures of certain classes of vessels, than would be considered satisfactory ashore. On passenger and cargo vessels, the structure is frequently covered with a metal sheathing in order to improve appearance. Such sheathing reduces the resistance to heat flow because the necessary supports form a metallic contact that bypasses the insulation.

Hull insulation may be either sheathed fill or blanket and board type and should possess certain desirable physical characteristics, namely:

1. *Fireproofness*. The material must be incombustible and when subjected to high temperatures by fires within compartments it must not give off smoke or harmful gases. If cements are used to secure the material they too must satisfy the same combustible restrictions. When the exposed surface of the insulation is to be finished with paint, the paint should be fire retardant. Insulation properly used will retard the spread of fires within ships. Government regulations govern the construction and insulation of bulkheads to prevent the spread of fire on vessels.

2. *Density*. A 6,000 ton warship may have from 13 to 25 tons of hull insulation,

depending on the type of insulation. It is obviously desirable to minimize this dead weight commensurate with other considerations.

3. *Thermal Conductivity.* It is important that the conductivity of insulation used be 0.33 Btu per hour per square foot per degree Fahrenheit, or less.

4. *Ruggedness.* As any exposed or internally applied material is subjected to rough usage aboard ship it must be able to withstand much pounding from the seas and vibration from the ship's machinery.

5. *Verminproof.* For sanitary reasons it is essential that insulation harbor no vermin.

6. *Applicability.* Because of the necessity of speeding construction of vessels and minimizing costs, insulating materials must lend themselves to easy and ready application. Generally, when cements are used the application is slow and laborious.

For duct insulation—mineral wool or spun glass are the most commonly used materials. Corrugated asbestos is not recommended because the presence of moisture tends to disintegrate it. Semi-rigid insulation is generally used because it is simplest to install. Blanket type insulations are applied only to round or curved surfaces, and are secured with twine and lagged with sewed-on canvas. Semi-rigid type insulation, secured by adhesive, flat wire bands, and corner clips, is lagged with canvas only where exposed. Blanket type insulation is at least 1 in. thick. The thickness of semi-rigid insulation is frequently selected in accordance with the values given in Table 2.

TABLE 2. DUCT INSULATION THICKNESS FOR USE IN SHIPS

APPLICATION	THICKNESS IN.
Tempered air ducts in unheated spaces where a temperature differential greater than 35 F exists	½
Cold air ducts, supply and exhaust passing through heated spaces	1
Exposed ducts in heated spaces, carrying reheated air	½
Concealed ducts carrying reheated air, adjacent structure exposed	1
Concealed ducts carrying reheated air, adjacent structure not exposed	½
Ducts serving only the space in which they are located	None
Exhaust ducts carrying hot gases (galley, forges, etc.), in living and working spaces	1
Supply ducts in machinery spaces and similar hot spaces, serving spaces other than the hot spaces through which they pass	1
Heaters	1
Ducts subject to excessive sweating	½

Fans are seldom insulated. Preheaters are frequently located close to the fresh air intake in order to conserve insulation, and for the same reason zone reheaters are located as close to the zone as possible. Where a reheater serves only one space, the heater is commonly located in the space.

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CHAPTER 39

Industrial Air Conditioning

Atmospheric Conditions Required, General Requirements, Classification of Problems, Control of Regain, Moisture Content and Regain, Conditioning and Drying, Control of Rate of Chemical Reaction, Control of Rate of Biochemical Reactions, Control Rate of Crystallization, Elimination of Static Electricity

INDUSTRIAL air conditioning is primarily concerned with the atmospheric conditions required for the manufacture, processing, and preservation of material, equipment and commodities. The fundamental factors, one or more of which may govern these conditions are: humidity, temperature, air motion and air purity. The term air purity may have reference to the quantity of dust, bacteria, odors, or toxic gases present.

The optimum atmospheric conditions have been established for a large variety of industrial processes, but when the requirements are unknown it may be necessary to determine them by laboratory tests or by research. When the requirements are not definitely specified it is advisable to design the equipment to be flexible and to provide for future increase of facilities and capacity. Air conditioning requirements for human comfort are defined in Chapter 2. If the atmospheric conditions required for the process are not within the range of human comfort, separate consideration must be given to the maintenance of conditions which will not have a detrimental effect upon the health of the occupants.

ATMOSPHERIC CONDITIONS REQUIRED

The most desirable relative humidity for processing depends upon the product and the nature of the process. As far as the behavior of the material and its desired final condition are concerned, each material and process presents a different problem. The desirable relative humidity may range from a low of 5 per cent, as in certain industrial applications, such as insulation winding processes, up to a condition approaching saturation, as in processes relating to textiles, tobacco and baking industries.

Similarly, the most favorable temperature will vary according to the specific material and particular process. Frequently a compromise between the known optimum condition for processing and that required for reasonable worker comfort is desirable. This is particularly true where unconfined processes are required in departments where people are working and their health, comfort and productive efficiency must be considered.

It is generally recognized that relative humidities of 50 per cent or less are on the dry side. Such conditions are conducive to low regains in hygroscopic materials, drying out, increased brittleness of fibrous materials, prevalence of increased static electricity and tendencies toward increased dust liberation from the product. Relative humidities higher than 50 per cent are considered to be on the damp side. These conditions are conducive to high regain, promote softness and pliability in materials, decrease static electricity and tendencies toward reduced generation of product dust which represents a loss in weight of the material in process.

In many processes, the optimum desired air conditions are variable according to the stage and progress of the processing cycle, from the raw material to the finished product. Some materials, such as cotton textiles, begin with a low relative humidity in the carding and picking rooms, and after passing through the various intermediate steps with a gradual increase of relative humidity, the product is subjected to relative humidities of from 75 to 85 per cent in the final stage of weaving. Other processes are encountered that require the reverse of this procedure, starting with a high relative humidity and finishing with a low relative humidity, as is the case when producing glue and gelatinous materials and making gelatine capsules.

In some cases the temperatures listed in Table 1 have no direct influence upon the product itself, except for effect upon efficiency of the employees which in turn affects workmanship, uniformity and the cost of production. In this category may be included the automobile assembly line. The time necessary to assemble the many parts into a complete unit is a factor recognized and associated with the worker's comfort, and the avoidance of fatigue with subsequent loss of efficiency.

Air conditioning contributes an important role during the processing, machining and honing of precision metal parts, instruments, tools, engines, guns, etc., which demand micrometric accuracy of dimensions, and which are affected by small temperature variations. Hence, some uniform condition is usually selected, both as to temperature and humidity to serve the demands of the worker's comfort and the exacting requirements of the process.

The temperatures and relative humidities listed in Table 1 should be analyzed in relation to the qualified requirements of the process. Conditions generally acceptable for *industrial processing* and for *general storage* are listed in these tables. While it is true that many storage requirements demand the control of some fixed air temperature and relative humidity condition, to hold and preserve the contents, it is not generally referred to as *processing*.

Logically many phases of drying may be included in the category of air conditioning for industrial processing, especially where temperature and humidity, by direct influence to product, bring about some definite change in physical characteristics as well as in weight. (See also Chapter 41.) As an illustration, refer to the conditions that are required to control the rate of crystallization in coating pans in which sugar syrup is applied to various forms of pills, nuts, gum, etc., in consecutive liquid doses, until a crystallized coating or jacket is built up to the required size and thickness. Here, the primary problem is one of drying which requires the supply of air at some fixed volume and velocity along with regulated control of both dry- and wet-bulb temperatures. The wet-bulb depression determines the rate of moisture pick-up or drying by the air and may be termed the *drying head*. Of equal importance is the uniformity at which the wet-bulb is maintained. If this is allowed to vary, poor results will follow due to checking and cracking of the unfinished coating. This is obvious when it is realized that continuous evaporation of moisture is taking place during the process and also that the temperature of the material corresponds to the air wet-bulb temperature and will vary accordingly. With undue expansion and contraction with every temperature change, the thin crystallized coatings which are not elastic will check and crack before the process is completed.

GENERAL REQUIREMENTS

Air conditioning apparatus for industrial purposes must be capable of absorbing heat from various sources such as machinery power, electric lights, people, sunlight and chemical reactions; of warming or cooling to any desired temperature; and of providing ample air supply. Refrigeration may or may not be required, depending upon natural conditions, the required relative humidity and the maximum permissible temperature. Washing, purifying and treating the air may be desirable. Good distribution is essential for the control of air motion and for the prevention of uneven conditions. Air velocity should not be so high that it will cause damage to the commodity or interfere with the industrial process. Accurate, sensitive and reliable automatic control of humidity or temperature is vital in most cases.

Outside weather conditions and the ventilation required for workers are of secondary importance in relation to the total work to be done by the air conditioning system. In extreme cases of high concentration of industrial heat from machinery and ovens the error of entirely omitting the heat gain through the building structure would not be serious. At the other extreme, where low temperatures must be produced with refrigeration and where comparatively little power is required by the machinery, the heat gain through the building structure will become the major factor in determining the size of equipment and in this case the ventilation requirement assumes importance.

Buildings which are to be air conditioned should therefore be designed with careful consideration of over-all cost and efficiency. Condensation resulting from high humidities must be prevented by suitable materials and construction, or else collected and drained to prevent loss of product or quick deterioration of the structure. Air leakage or filtration may add greatly to operating costs or make the maintenance of low humidities (relative or absolute) wholly impossible. Low temperatures require good insulation.

It is apparent that the subject of air conditioning for industrial processes is extensive and greatly involved, and that a detailed treatment is therefore beyond the scope of this chapter.

CLASSIFICATION OF PROBLEMS

Any industrial air conditioning problem may be listed under one or more of the following five classifications: (1) control of regain, (2) control of rate of chemical reactions, (3) control of rate of biochemical reactions, (4) control of rate of crystallization, and (5) elimination of static electricity.

Control of Regain

In the manufacture or processing of hygroscopic materials such as textiles, paper, wood, leather, tobacco and foodstuffs, the temperature and relative humidity of the air have a marked influence upon the rate of production and upon the weight, strength, appearance and general quality of the product. This influence is due to the fact that the moisture content of materials having a vegetable or animal origin, and to a lesser extent minerals in certain forms, comes to equilibrium with the moisture of the surrounding air.

In industries where the physical properties of a product affect its value, the percentage of moisture is of special importance. With increase in moisture content, hygroscopic materials ordinarily become softer and more pliable. Standards of regain are firmly fixed in trade with fair

Table 1. Temperatures and Humidities Applicable to Industrial Air Conditioning

Industry	Process	Temperature Degrees Fahrenheit	Relative Humidity Per Cent
Automobile	Assembly line	65 to 80	40 to 55
	Precision parts—honing—machining	75 to 80	40 to 55
Baking	Cake icing	70	50
	Cake mixing	75	65
	Dough fermentation room	80	76 to 80
	Dough retarding	32 to 40	76 to 85
	Loaf cooling	70	60 to 70
	Make-up room	75 to 80	55 to 70
	Mixing room	75 to 80	55 to 70
	Paraffin paper wrapping	80	55
	Proof boxes	90 to 95	80 to 90
	Storage of flour	65 to 75	55 to 65
	Storage of yeast	32 to 45	60 to 75
Biological Products	Vaccines	below 32	
	Antitoxins	38 to 42	
	Blood bank	38 to 42	60 to 65
	Penicillin (dehydrated)	36 to 40	
Brewing	Fermentation in vat room	44 to 50	50
	Storage of grains	60	30 to 45
Ceramic	Drying of auger machine brick	180 to 200	
	Drying of refractory shapes	110 to 150	50 to 60
	Molding room	80	60
	Storage of clay	60 to 80	35 to 65
Chemical	General storage	60 to 80	35 to 50
Confectionery	Chewing gum rolling	75	50
	Chewing gum wrapping	70	45
	Chocolate covering	62 to 65	50 to 55
	Hard candy making	70 to 80	30 to 50
	Packing	65	50
	Starch room	75 to 85	50
	Storage	60 to 68	50 to 65
Distillery	General manufacture	60 to 75	45 to 65
	Storage of grains	60	30 to 45
Drug	Deliquescent powder	75	35
	Effervescent granulations	80	40
	Liver extracts (powdered)	70	20 to 30
	Storage of powders and tablets	70 to 80	30 to 35
	Tablet compressing	70 to 80	40
	Packaging	80	40
Electrical	Insulation winding	104	5
	Manufacture of cotton covered wire	60 to 80	60 to 70
	Manufacture of electrical windings	60 to 80	35 to 50
	Storage of electrical goods	60 to 80	35 to 50
Food	Butter making	60	60
	Dairy chill room	40	60
	Preparation of cereals	60 to 70	38
	Preparation of macaroni	70 to 80	38
	Ripening of meats	40	80
	Slicing of bacon	60	45
	Storage of apples	31 to 34	75 to 85
	Storage of citrus fruit	32	80
	Storage of eggs in shell	30	80
	Storage of meats (frozen)	0 to 5	85
	Storage of meats (above freezing)	36	85
	Storage of sugar	80	35
Fur	Drying of furs	110	
	Storage of furs	28 to 40	50 to 65

TABLE 1. TEMPERATURES AND HUMIDITIES APPLICABLE TO INDUSTRIAL AIR CONDITIONING—(Concluded)

Industry	Process	Temperature Degrees Fahrenheit	Relative Humidity Per Cent
INCUBATORS	Chicken	99 to 102	55 to 75
INSTRUMENTS	Repair and calibration	68	50 to 55
LABORATORY	General analytical and physical	60 to 70	60 to 70
	Storage of materials	60 to 70	35 to 50
LEATHER	Drying of hides	90	
	Mulling	95 to 100	95
LIBRARY	Book storage	65 to 70	38 to 50
LINOLEUM	Printing	80	40
MATCHES	Manufacturing	72 to 74	50
	Storage of matches	60	
MUNITIONS	Fuse loading	70	55
PAINT	Air drying lacquers	70 to 90	25 to 50
	Baking lacquers	180 to 300	
	Air drying of oil paints	60 to 90	25 to 50
PAPER	Binding, cutting, drying, folding, gluing	60 to 80	40 to 60
	Storage of paper	75 to 80	40 to 60
	Testing Laboratory	60 to 80	55 to 65
PHOTOGRAPHIC	Development of film	70 to 75	60
	Drying	75 to 80	50
	Printing	70	70
	Cutting	72	65
PRINTING	Binding	70	45
	Folding	77	65
	Press room (general)	75	60 to 78
	Press room (lithographic)	75 to 80	50 to 60
	Storage of rollers	70 to 90	50 to 55
RUBBER	Manufacturing	90	
	Dipping of surgical rubber articles	75 to 80	25 to 30
	Standard laboratory tests	80 to 84	42 to 48
	Cementing	80	25 to 30
SOAP	Drying	110	70
TEXTILE	Cotton— carding	75 to 80	50 to 55
	combing	75 to 80	60 to 65
	roving	75 to 80	50 to 60
	spinning	60 to 80	50 to 70
	weaving	68 to 75	85
	Rayon— spinning	70	85
	throwing	70	60
	weaving	75 to 88	60 to 75
	Silk— dressing	75 to 80	60 to 65
	spinning	75 to 80	65 to 70
	throwing	75 to 80	65 to 70
	weaving	75 to 80	60 to 70
	Wool— carding	75 to 80	65 to 70
	spinning	75 to 80	55 to 60
	weaving	75 to 80	50 to 55
	Testing Laboratory	70	65
TOBACCO	Cigar and cigarette making	70 to 75	55 to 75
	Softening	90	85
	Stemming or stripping	75 to 85	70

penalties for excesses. Deficiencies result in loss of revenue to seller and loss of desirable quality to buyer.

Manufacturing economy therefore requires that the moisture content be maintained at a percentage favorable to rapid and satisfactory manipulation and to a minimum loss of material through breakage. A uniform condition is desirable in order that high speed machinery may be adjusted permanently for the desired production with a minimum loss from delays, wastage of raw material and defective product.

In the processing of hygroscopic materials, it is usually necessary to secure a final moisture content suitable for the goods as shipped. Where the goods are sold by weight, it is proper that they contain a normal or standard moisture content.

Moisture Content and Regain

The terms *moisture content* and *regain* refer to the amount of moisture in hygroscopic materials. *Moisture content* is the more general term and refers either to free moisture (as in a sponge) or to hygroscopic moisture (which varies with atmospheric conditions). It is usually expressed as a percentage of the total weight of material. *Regain* is more specific and refers only to hygroscopic moisture. It is expressed as a percentage of the *bone-dry* weight of material. For example, if a sample of cloth weighing 100.0 grains is dried to a bone-dry weight of 93.0 grains, the loss in weight, or 7.0 grains, represents the weight of moisture originally contained. This expressed as a percentage of the total weight (100.0 grains) gives the moisture content or 7 per cent. The regain, which is expressed as a percentage of the bone-dry weight, is $\frac{7.0}{93.0}$ or 7.5 per cent.

The use of the term *regain* does not imply that the material as a whole has been completely dried out and has re-absorbed moisture. During the processing of certain textiles, for instance, complete drying during manufacturing is avoided as it might appreciably reduce the ability of the material to re-absorb moisture. A basis for calculating the regain of textiles is obtained by drying under standard conditions; a sample from the lot and the dry weight thus obtained is used as a basis in the calculations to determine the regain.

The moisture content of a hygroscopic material at any time depends upon the nature of the material and upon the temperature and especially the relative humidity of the air to which it has been exposed. Not only do different materials acquire various percentages of moisture after prolonged exposure to a given atmosphere, but the rate of absorption or drying varies with the nature of the material, its thickness and density.

Table 2 shows the regain or hygroscopic moisture content of several organic and inorganic materials when in equilibrium at a dry-bulb temperature of 75 F and various relative humidities. The effect of relative humidity on regain of hygroscopic substances is clearly indicated. The effect of temperature is comparatively unimportant. In the case of cotton, for instance, an increase in temperature of 10 F has the same effect on regain as a decrease in relative humidity of one per cent. Changes in temperature do, however, affect the rate of absorption or drying. Sudden changes in temperature cause temporary fluctuations in regain even when the relative humidity remains stationary.

The regain or moisture content affects the physical properties of textiles to a marked degree, changing the strength, pliability and elasticity.

The fact that the regain of textiles will come into equilibrium with the

TABLE 2. REGAIN OF HYGROSCOPIC MATERIALS

Moisture Content Expressed in Per Cent of Dry Weight of the Substance at Various Relative Humidities—Temperature, 75 F

Classification	Material	Description	Relative Humidity—Per Cent									Authority
			10	20	30	40	50	60	70	80	90	
Natural Textile Fibers	Cotton	Sea island—roving	2.5	3.7	4.6	5.5	6.6	7.9	9.5	11.5	14.1	Hartshorne
	Cotton	American—cloth	2.6	3.7	4.4	5.2	5.9	6.8	8.1	10.0	14.3	Schloesing
	Cotton	Absorbent	4.8	9.0	12.5	15.7	18.5	20.8	22.8	24.3	25.8	Fuwa
	Wool	Australian merino—skein	4.7	7.0	8.9	10.8	12.8	14.9	17.2	19.9	23.4	Hartshorne
	Silk	Raw chevennes—skein	3.2	5.5	6.9	8.0	8 9	10.2	11.9	14.3	18.8	Schloesing
	Linen	Table cloth	1.9	2.9	3.6	4.3	5.1	6.1	7.0	8.4	10.2	Atkinson
	Linen	Dry spun—yarn	3.6	5.4	6.5	7.3	8.1	8.9	9.8	11.2	13.8	Sommer
	Jute	Average of several grades	3.1	5.2	6.9	8.5	10.2	12.2	14 4	17 1	20 2	Storch
	Hemp	Manila and sisal—rope	2.7	4.7	6.0	7.2	8.5	9.9	11.6	13.6	15.7	Fuwa
Rayons	Viscose Nitrocellulose Cupramonium	Average skein	4.0	5.7	6.8	7.9	9.2	10.8	12.4	14.2	16.0	Robertson
	Cellulose Acetate	Fibre	0.8	1.1	1.4	1 9	2 4	3.0	3.6	4 3	5 3	Robertson
Paper	M F Newsprint	Wood pulp—24% ash	2.1	3.2	4.0	4 7	5.3	6.1	7.2	8.7	10.6	U. S. B. of S.
	H M F Writing	Wood pulp—3% ash	3.0	4.2	5.2	6 2	7.2	8.3	9.9	11.9	14.2	U. S. B. of S.
	White Bond	Rag—1% ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2	U. S. B. of S.
	Com. Ledger	75% rag—1% ash	3.2	4.2	5.0	5 6	6.2	6.9	8.1	10.3	13.9	U. S. B. of S.
	Kraft Wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9	U. S. B. of S.
Misc. Organic Materials	Leather	Sole oak—tanned	5.0	8.5	11.2	13.6	16.0	18.3	20.6	24.0	29.2	Phelps
	Catgut	Racquet strings	4.6	7.2	8.6	10.2	12.0	14 3	17.3	19.8	21.7	Fuwa
	Glue	Hide	3 4	4 8	5.8	6.6	7.6	9.0	10.7	11.8	12.5	Fuwa
	Rubber	Solid tire	0.11	0.21	0 32	0.44	0 54	0.66	0.76	0.88	0.99	Fuwa
	Wood	Timber (average)	3 0	4.4	5.9	7.6	9.3	11.3	14.0	17 5	22.0	Forest P Lab.
	Soap	White	1.9	3.8	5.7	7.6	10.0	12.9	16.1	19.8	23.8	Fuwa
	Tobacco	Cigarette	5.4	8.6	11.0	13.3	16.0	19.5	25 0	33.5	50.0	Ford
Food-stuffs	White Bread		0.5	1.7	3.1	4.5	6.2	8.5	11.1	14.5	19.0	Atkinson
	Crackers		2.1	2.8	3.3	3.9	5.0	6.5	8.3	10 9	14.9	Atkinson
	Macaroni		5.1	7.4	8.8	10.2	11 7	13.7	16.2	19.0	22.1	Atkinson
	Flour		2.6	4.1	5.3	6.5	8.0	9.9	12 4	15.4	19.1	Bailey
	Starch		2.2	3.8	5.2	6.4	7.4	8.3	9.2	10.6	12.7	Atkinson
	Gelatin		0.7	1.6	2.8	3.8	4.9	6.1	7.6	9.3	11.4	Atkinson
Misc Inorganic Materials	Asbestos Fiber	Finely divided	0.16	0 24	0.26	0.32	0.41	0 51	0.62	0.73	0.84	Fuwa
	Silica Gel		5.7	9.8	12.7	15.2	17.2	18.8	20.2	21.5	22 6	Fuwa
	Domestic Coke		0.20	0.40	0 61	0.81	1.03	1.24	1 46	1.67	1.89	Selvig
	Activated Charcoal	Steam activated	7.1	14 3	22 8	26 2	28.3	29.2	30.0	31.1	32.7	Fuwa
	Sulphuric Acid	H_2SO_4	33.0	41 0	47 5	52.5	57 0	61.5	67.0	73.5	82.5	Mason

conditions of the surrounding air and vary with its temperature and relative humidity is the fundamental basis for the control of physical qualities during manufacture. During the preparation processes in a cotton mill, the cotton fibers should be in a condition to be easily carded.

These preliminary processes are carried out best in a relative humidity

of 50 to 55 per cent. As the cotton fiber comes to the spinning operation, more flexibility is needed and the relative humidity is increased in this department. For many years, 65 per cent relative humidity was considered the optimum. To offset the extra work performed on the fiber as the spindle speed is increased, many cotton mills now carry 70 per cent relative humidity in the spinning rooms.[1] Winding, warping and weaving are all processes calling for great flexibility and a consequent need for higher humidity.

Other textile fibers, due to their different natural characteristics, are processed under relative humidities and temperatures applicable to each.

Rayons, on account of great loss of strength with the higher regains, should be processed in a relative humidity of 55 to 70 per cent. Acetate silk, another chemical fiber, with approximately 50 per cent of the regain of rayon, may be processed between 60 and 65 per cent relative humidity.

All hygroscopic materials when in the state of absorbing moisture from the surrounding air produce a sensible heat rise to the air equivalent to the latent heat released by air to the material. This adiabatic conversion may account for a small percentage of the total heat load of the conditioned space.

Conditioning and Drying

In general, the exposure of materials to desirable conditions for treatment may be coincidental with the manufacture or processing of the materials, or they may be treated separately in special enclosures. This latter treatment may be classified as conditioning or drying. The purpose of conditioning or drying is usually to establish a desired condition of moisture content and to regulate the physical properties of the material.

When the final moisture content is lower than the initial one, the term *drying* is applied. If the final moisture content is to be higher, the process is termed *conditioning*. In the case of some textile products and tobacco, for example, drying and conditioning may be combined in one process for the dual purpose of removing undesirable moisture and accurately regulating the final moisture content. Frequently conditioning or drying is made a continuous process in which the material is conveyed through an elongated compartment by suitable means and subjected to controlled atmospheric conditions.

Control of Rate of Chemical Reactions

A typical example of the second general classification, the control of the rate of chemical reactions, occurs in the manufacture of rayon. The pulp sheets are conditioned, cut to size, and passed through a mercerizing process. It is essential that during this process close control of both temperature and relative humidity should be maintained. The temperature controls the rate of reaction directly, while the relative humidity maintains a constant rate of evaporation from the surface of the solution and obtains a solution of known strength throughout the mercerizing period.

Another well-known example in this class is the *drying* of varnish which is an oxidizing process dependent upon temperature. High relative humidities have a retarding effect on the rate of oxidization at the surface and allow the internal gases to escape freely as the chemical oxidizers *cure*

[1]The Present Status of Textile Regain Data, by A E. Stacey, Jr. (*National Association of Cotton Manufacturers*, 1927).

the varnish from within. This produces a surface free from bubbles and a film homogeneous throughout. Desirable temperatures for *drying* varnish vary with the quality. A relative humidity of 65 per cent is beneficial for obtaining the best processing results.

Control of Rate of Biochemical Reactions

In the field of biochemical control, industrial air conditioning has been applied to many different and well-known products. All problems involving fermentation are classed under this heading. As biochemistry is a subdivision of chemistry, subject to the same laws, the rate of reaction may be controlled by temperature. An example of this is the dough room of the modern bakery. Yeast develops best at a temperature of 80 F. A relative humidity of 65 per cent is maintained so as to hold the surface of the dough open to allow the carbon dioxide gases formed by the fermentation to pass through and produce a loaf of bread, when baked, of even, fine texture without large voids.

Another example of a similar process is found in the curing of macaroni. The flour and water mixture is fermented and dried. As it is necessary to have a definite amount of water present to carry on a fermentation process, the moisture must be removed in a relatively short period to stop fermentation and prevent souring and in such a manner as to avoid setting up internal strains in the mixture. Best results are obtained with the correct cycles of both temperature and humidity.

The curing of fruits, such as bananas and lemons, also comes under this classification. Bananas are treated somewhat differently and to accomplish the required results, a cycle of temperatures and relative humidities is used. The starches in the pulp of the fruit must be changed and the skin cured and colored, after which the fruit is cooled to maintain as low a rate of metabolism as possible. Ideal conditions range between 55 to 57 F and in no case should the temperature go below 49 F, as the starches then would become fixed and be indigestible.

The curing of lemons is an entirely different problem. Bananas are cured for a quick market, while lemons are held for a future market. The process, therefore, varies in the temperature used. Temperatures from 54 to 59 F have been found to be best suited for this process. A high relative humidity of 88 to 90 per cent is necessary to hold shrinkage to a minimum and, at the same time, develop the rind so it will be sufficiently tough to permit handling.

Tobacco from the field to the finished cigar, cigarette, plug or pipe tobacco, offers another interesting example of what may be done by industrial air conditioning in the control of color, texture and flavor. In the processing of tobacco, the first three classifications of air conditioning are involved, and only through close atmospheric control can the best quality of the leaf be developed.

Control of Rate of Crystallization

The rate of cooling of a saturated solution determines the size of the crystals formed. Both dry- and wet-bulb temperatures are of importance, as the one controls the rate of cooling, while the other, through evaporation, changes the density of the solution.

In the coating pans for pills, gum and nuts, a heavy sugar solution is added to the tumbling mass. As the water evaporates, each separate piece is covered with crystals of sugar. A smooth, opaque coating is only accomplished by blowing into the kettle the proper amount of air at the right dry- and wet-bulb temperatures.

Elimination of Static Electricity

The presence of static electricity is very detrimental to the satisfactory and economical processing of many light materials, such as textile fibers, paper, etc. It is also extremely dangerous where explosive atmospheres or materials are present. Fortunately, this hazard is easily eliminated by increasing the relative humidity.

In attempting to eliminate static electricity, it must be borne in mind that for successful elimination the air that actually comes in contact with the material in the machine must be at a relative humidity of 50 per cent or more. As some machines consume a great deal of power which is converted directly into heat, the temperature in the machine may be considerably higher than the temperature adjacent to the machine where the relative humidity is normally measured. In such cases, the relative humidity in the machine will be appreciably lower than that elsewhere in the room, and it may be necessary to maintain a room relative humidity of 65 per cent, or even more, before the desired results can be obtained.

CALCULATIONS

The methods for determining the proper heating and cooling loads for the various industrial processes are similar to those outlined in Chapters 6 and 7. Because of the large number of motors and heat producing units usually prevalent in an industrial application, it is particularly important that operating allowances for the latent and sensible heat loads be definitely ascertained and used in the calculations to determine the total design load.

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CHAPTER 40

Industrial Exhaust Systems

Classification of Systems, Hood Design Principles, Hood Suction and Indraft Velocities, Duct System Design, Air Cleaning Equipment, Resistance of Systems, Air Flow Producing Equipment, Protection Against Corrosion

IN many industrial plants some type of exhaust system designed to collect and remove dusts, fumes, mists, vapors and gases is essential in order to promote efficiency, economy, and safety of operation. Definitions of these various contaminants are as follows:

Dusts are solid particles generated in manufacturing processes due to handling, crushing, grinding or impact of organic or inorganic materials such as rock, ore, metals, coal, wood, grain, etc.

Fumes are solid particles generated by condensation from the gaseous state, generally after volatilization of molten metals, etc.

Mists are suspended liquid droplets generated either by condensation from the gaseous to the liquid state or by breaking up of a liquid into a state of dispersion by some process such as splashing, foaming or atomizing.

Vapors are the gaseous form of substances which are normally in the solid or liquid state and which can be changed to either of these states either by increasing the pressure or by decreasing the temperature.

Gases are normally formless fluids which occupy the space or enclosure, and which can be changed to the liquid or solid state either by increasing the pressure or by decreasing the temperature, or by the combined effect of increasing the pressure and decreasing the temperature.

The theory of air flow in an exhaust system in the following paragraphs *A* to *D* will be found in a publication[1] of the *American Foundrymen's Association.*

A. When the air flow producing equipment of an exhaust system is properly operated it will produce a negative pressure (below atmospheric) in the exhaust side of the system sufficient to overcome all resistance and to sustain the desired air velocity; and, further, will overcome all resistances on the discharge or positive pressure side of the system so that the air drawn through exhaust inlets will be discharged against atmospheric pressure.

B. An exhaust system is entirely dependent on a *sufficient volume* of air flowing into the exhaust inlets to catch the matter to be exhausted before such matter has an opportunity to diffuse into the general atmosphere of the work place or room.

C. The velocity of the air flowing into an exhaust inlet is usually of secondary importance and becomes an essential factor only when a certain velocity is required to overcome some force acting on the matter to be caught. The velocity within an exhaust system is only important to the extent that it shall be sufficient to convey the entrained matter and prevent it from settling or dropping out in the piping system. Velocity in terms of velocity pressure is most essential in designing a system because it is the basis upon which all calculations are made. In testing and checking a system the velocity as determined from the velocity pressure reading obtained by means of a Pitot tube is the only true indication of the exact air flow in a pipe or system.

D. The total pressure within an exhaust system is only of importance in determining the power required to operate the system. Total pressure tests do not indicate the proper functioning of an exhaust system as related to the volume and velocity of the air flowing into an exhaust inlet.

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[1]Tentative Recommended Good Practice Code and Handbook on the Fundamentals of Design, Construction, Operation and Maintenance of Exhaust Systems, Page 21, Industrial Hygiene Codes Committee, *American Foundrymen's Association.*

General design information is included in this chapter which is intended to relate primarily to industrial exhaust systems.

CLASSIFICATION OF SYSTEMS

In general there are two basic layouts of exhaust systems, the central and the multiple unit system. In the central system a fan is located near the center of operations with a piping system radiating to the various machines to be served. In the multiple unit system, which is sometimes employed where the machines to be served are widely scattered, or where the operations are apt to be independent or intermittent, small individual exhaust fans are located at the center of the machine groups or at each machine. The unit arrangement has the advantage of flexibility.

Exhaust systems may be classified with respect to the nature of the material to be handled by them as (*a*) those handling dust and (*b*) those handling fumes, mists, vapors and gases. There is a marked difference in design details of systems serving dust producing operations and those exhausting the more vapor-like matter even though the same basic theories govern both classes. The type of hood and exhauster, the adaption of general ventilation, the conveying velocities and duct construction differ substantially.

Exhaust systems are also classified by the means employed to collect the material. The dust or refuse may be collected and controlled by enclosed hoods or open hoods with positive inward air movement or by diluting or exhausting the general air of the room. With some classes of machinery it is not feasible to hood the machines closely and in these cases open hoods over or adjacent to the machines are provided to collect as much as possible of the dust and fumes. This class includes such machines as rubber mills, package filling machinery, sand blast, crushers, forges, pickling tanks, melting furnaces, and the unloading points of various types of conveyors.

Open hoods should be placed as close to the source of dust or fumes as possible, with due regard to the movements of the operator, and should be placed so that the operator is in no case in the path of the exhausted material. When the hood must be placed at some distance above the machine it should be large enough to cover a large area as diffusion is usually quite rapid.

Some consideration should be given to the natural movement of the fumes. For those that are lighter than air, the hood may be over or above the machine; and where a heavy vapor, or dust-laden air at ordinary temperature is to be removed, horizontal or floor connections are sometimes preferable. In many cases there are convection currents and other atmospheric disturbances in the work room which should be given consideration. These disturbances diminish the tendency of dust and fumes to settle from the room air because of their density.

In another class of operation the main objective is to prevent the escape of dust into the surrounding atmosphere, and the removal of some dust from the machine or enclosure may be merely incidental. The dust-creating apparatus is enclosed within a housing which is made as tight as practicable, and sufficient suction is applied to the enclosure to maintain an inward air leakage, thus preventing escape of the dust. While the exhaust system is required to handle only the air which leaks in through the crevices and openings in the enclosure, yet in many installations leakages are very high and great care is required to obtain satisfactory

results with a system of this kind. The inward-leakage principle is utilized for controlling dust in the operating of tumbling barrels, grinding, screening, elevating, and similar processes.

Certain dust and fume producing operations are best carried on by isolating the process in a separate compartment or room and then applying general ventilation to this space. The compartment or room in which the work is performed should be as small as is consistent with convenience in handling the work. The ventilating system should be designed so that a current of clean air is drawn across the work in such a manner as to carry the dust or fume away from the operator and out of the work space. Another method of accomplishing the control of this type of installation, and one usually applied in the case of gases and fumes, is the dilution method. In this case sufficient clean air is introduced generally into the work space to dilute the contamination to a safe level.

HOOD DESIGN PRINCIPLES[2, 3]

The first step in the design of an exhaust system is to determine the number and size of the hoods and their connections. No general rules, however, can be given since hood and duct dimensions are determined by the characteristics of the operations to which they are applied. When a tentative decision regarding the set-up has been made, it is then necessary to obtain the suction and air velocities required to effect control. At this point the designer must rely upon the prevailing practice and on such physical data relating to hoods, duct systems and collectors as are available. The fan speed must be sufficient to maintain the estimated suction and air velocities in the system. In general, the most important requirements[4] of an efficient exhaust and collecting system are:

1. Hoods, ducts, fans, motors and collectors should be of adequate size and type.
2. The air velocities should be sufficient to control and convey the materials collected.
3. The hoods and ducts should be placed so as not to interfere with the operation of a machine or any working part.
4. The system should do the required work with a minimum power consumption.
5. When inflammable dusts and fumes are conveyed, the piping should be provided with an automatic damper in passing through a fire-wall.
6. Ducts and all metal parts should be grounded to reduce the danger of dust explosions by static electricity.
7. The design of an exhaust system should afford easy access to parts for inspection and care.

HOOD SUCTION AND INDRAFT VELOCITIES

The removal of dust or waste by means of an exhaust hood requires a movement of air at the point of origin sufficient to carry it into a collecting system. The air velocities necessary to accomplish this depend upon the physical properties of the material to be eliminated and the direction and speed with which it is thrown off. If the dust to be removed is already in motion, as is the case with high-speed grinding wheels, the hood must be installed in the path of the particles so that a minimum air volume may be used effectively. It is always desirable to design and locate a

[2]How to Design Exhaust Hoods, by J M. DallaValle (*Heating and Ventilating*, Series of 12 articles March, 1943 to February, 1944)

[3]Industrial Exhaust Ventilation in Industrial Hygiene, by Allen D Brandt (A.S H V.E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, July, 1944, p. 428).

[4]For more detailed requirements refer to Fundamentals Relating to the Design and Operation of Exhaust Systems, Z9-1936 (*American Standards Association*). Industrial Code Bulletin Nos. 10 and 12 (New York State Labor Department). Principles of Exhaust Hood Design, by J. M. DallaValle (*U. S Public Health Service*, 1939).

Table 1. Rates of Flow Through Branch Pipes Woodworking Machines

Pipe Diameter, In.	Air Volume, Cfm	Pipe Diameter, In	Air Volume, Cfm
3	200	6	800
4	350	7	1100
5	550	8	1400

Table 2. Branch Pipe Size for Woodworking Machine Hoods

Type of Machine	Size, In.		No. of Branches	Minimum Diameter, In.		
	Min.	Max		Bottom Branch	Top Branch	Others
Self feed table saw			2	5	4	
Other single saws		18	1		4	
	18		1		5	
Saws with Dado Head			1		5	
Band saws		2	2	4	4	
	2	3	2	5	4	
	3	6	2	5	5	
Disc sanders		18	1	4		
	18	28	1	5		
	26	32	2	4	4	
	32	38	2	5	4	
	38	48	3	5	4	4
Triple drum sanders		30	1	7		
	30	36	1	8		
	36	42	1	9		
	42	48	1	10		
Single drum sanders: (area in sq in.)		350	1	4[a]		
	350	700		5		
	700	1400		6		
	1400	2800		7		
Horizontal belt sanders		9	2	5	4	
	9	14	2	6	4	
Vertical belt sanders		6	1	4		
	6	9	1	5		
	9	14	1	6		
Jointers		8	1	4		
	8	20	1	5		
Single planers		20	1	5		
	20	26	1	6		
	26	36	1	7		
Tenoner			2	5	5	

[a]Not over 10 in. diameter.

Table 3. Rates of Flow Through Branch Pipes Grinding and Buffing Wheels

Pipe Diameter, In.	Air Volume, Cfm	Pipe Diameter, In.	Air Volume, Cfm
3	225	6	900
4	400	7	1200
5	600		

hood so that the volume of air necessary to produce results is as small as possible. This will reduce the size of equipment, the power required by the system, and also the heating load requirements in winter.

Air Flow from Static Readings

The static suction at the throat of a hood is frequently used in practice as a measure of the effectiveness of control. Where the hood coefficient is known the volume of air flow through any hood may be determined from the equation:

$$Q = 4005\, f\, A \sqrt{h_t} \tag{1}$$

where

Q = volume of air flow, cubic feet per minute.
A = area of connecting duct, square feet.
h_t = static suction measured 3 diameters from throat of hood, inches of water.
f = orifice or restriction coefficient which varies from 0.6 to 0.9 depending on the shape of the hood.

An average value of f is 0.71, although for a well-shaped opening a value of 0.8 may be used. The factor f is determined from the equation:

$$f = \sqrt{\frac{h_v}{h_t}} \tag{2}$$

where h_v is the velocity head in the connecting duct.

The *static suction* is not a good measure of the effectiveness of a hood unless the area of the opening and the location of the operation with respect to the hood are known. This is clearly indicated by Equation 3 which shows that the velocity at any point along the axis varies approximately inversely as the square of the distance. However, this formula coupled with Equation 1 should serve to indicate the velocity conditions to be expected when operations are conducted external to the hood opening.

Design Based on Total Air Flow

Where the foregoing factors are not known, the usual method of designing an exhaust system is to base the air flow through the system on rates of flow through each hood which have been found by experience to provide adequate control. For woodworking systems the rates of flow given in Table 1, calculated on the basis of a branch velocity of 4000 fpm, are adequate for control. Using these air flow rates, Table 2 gives the size of pipe connections to be used with the more common woodworking machines. Properly designed grinding and buffing wheel hoods have been found to be adequately controlled when the rates of air flow given in Table 3, calculated on the basis of a branch velocity of 4500 fpm, are used. Table 4 gives the minimum branch pipe sizes to be used on the more common sizes of grinding and buffing wheels.

In some states grinding, polishing and buffing wheels are subject to regulation by codes. (See Standards Chapter 48). The static suction requirements, which range from 1½ to 5 in. water displacement in a U-tube, must be followed in such states although in several instances they appear to be excessive. Frequently, in these operations, a large part of the wheel must be exposed and the dust-laden air within the hood is thrown outward by the centrifugal action of the wheel, thus counteracting useful inward draft. This tendency may be diminished by locating the connecting duct so as to create an air flow of not less than 200 fpm past the lower edge of the wheel.

Controlling Air Velocities

Exact determinations of hood control velocities are not available, but it is safe to assume that for most dusty operations velocities should not be less than 200 fpm at the point of origin. Recommended air velocities through hood openings for various processes are given in Table 5. For granite dust generated by pneumatic devices, velocities from 150 to 200 fpm, depending on the type of hood used, are recommended as sufficient for safe control[5]. Considering the character of the industry, air velocities of this order may be extended to similar dusty operations. The method for approximately determining these velocities in terms of the velocity at the hood opening is given in Equation 3.

No set rule can be given regarding the shape of a hood for a particular operation, but it is well to remember that its essential function is to create an adequate velocity distribution. The fact that the zone of greatest effectiveness does not extend laterally from the edges of the opening may frequently be utilized in estimating the size of hood required. Where complete enclosure of a dusty operation is contemplated, it is desirable to leave enough free space to equal the area of the connecting duct. Hoods for grinding, polishing and buffing should fit closely, but at the same time should provide an easy means for changing the wheels. It is advisable to design these hoods with a removable hopper at the base to capture the heavy dust and articles dropped by the operator. Such provisions are of assistance in keeping the ducts clear. Air volumes used to control many dust discharges may often be reduced by effective baffling or partial enclosure of an operation. This procedure is strongly urged where dusts are directed beyond the zone of influence of the hood.

TABLE 4. BRANCH PIPE SIZES FOR GRINDING AND BUFFING HOODS

Type of Wheel	Wheel Size Diameter, In.		Maximum		Branch Pipe Minimum Diameter, In
	Min.	Max	Width In	Area Sq In.	
Grinding	....	9	1	30	3
	9	18	3	175	4
	18	24	4	300	5
	24	30	5	500	6
	30	36	6	700	7
Disc Grinding	....	20	....	300	4
	20	30	....	....	5
Buffing, Polishing and Scratch Brushing	....	8	2	50	3½
	8	16	3	150	4
	16	24	4	300	5
	24	30	6	600	6

Axial Velocity Formula for Hoods

When the normal flow of air into a hood is unobstructed, Equation 3 may be used to determine the air velocity at any point along the axis[6]:

[5]Control of the Silicosis Hazard in the Hard Rock Industries. I. A Laboratory Study of the Design of Dust Control Systems for Use with Pneumatic Granite Cutting Tools, by Theodore Hatch, Philip Drinker and Sarah P. Choate. (*Journal of Industrial Hygiene*, Vol. XII, No. 3, March, 1930).

[6]The Control of Industrial Dust, by J. M. DallaValle (*Mechanical Engineering*, Vol. 55, No. 10, October, 1933).

$$V = \frac{0.1\,Q}{x^2 + 0.1\,A} \tag{3}$$

where V = velocity at point, feet per minute.
Q = volume of air handled, cubic feet per minute.
x = distance along axis, feet.
A = area of opening, square feet.

Velocity Contours

It is possible by use of a specially constructed Pitot tube[7] to map contours of equal velocity in any axial plane located in the field of influence. It has been found that the positions of these contours for any hood can be expressed as percentages of the velocity at the hood opening and are purely functions of the shape of the hood[8].

Further, the velocity contours are identical for similar hood shapes when the hoods are reduced to the same basis of comparison. These facts are applicable to all hood problems so that when the velocity contour

TABLE 5. RECOMMENDED AIR VELOCITIES THROUGH OPENINGS IN HOODS ENCLOSING OPERATIONS OR LOCATED OVER ZONES OF GENERATION OF DUSTS, FUMES, VAPORS AND GASES RELEASED IN CERTAIN MANUFACTURING PROCESSES

CONDITION OF GENERATION OF CONTAMINANT	MINIMUM AIR VELOCITY, FPM	PROCESS
Released without noticeable movement.........	50–100	Evaporation of vapors, exhaust from pickling, washing, degreasing, plating, welding, etc.
Released with low velocity	100–200	Paint spraying in booth; inspection, sorting, weighing, packaging, low speed conveyor transfer points, rotating mixtures, barrel filling.
Active generation	200–500	Foundry shakeout, high speed conveyor transfer points, crushers, screens.
Released with great force...	500–2000	Grinding, tumbling mills, abrasive cleaning

distribution is known, the air flow required can be determined. Fig. 1 shows the contour distribution in two axial planes perpendicular to the sides of a rectangular hood with a side ratio of one-half. The distribution shown is identical for all openings with a similar side ratio provided the mapping is as shown in the figure. The contours, of course, are expressed as percentages of the velocity at the opening.

Low Velocity Systems

On multiple installations of the same operation it is often possible to institute a great saving in power cost by designing an exhaust system using low velocities in the main ducts. Such a system for use in grinding and shaping porcelain has been described[9]. In these operations, the separate machines are grouped around a central plenum chamber and exhausted by means of a low pressure fan connected to the plenum. In this case a power saving of over 90 per cent was obtained. A similar design technique[10] has been described for use in ventilating plating tanks.

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[7]Studies in the Design of Local Exhaust Hoods, by J. M. DallaValle and Theodore Hatch (*A.S.M.E. Transactions*, Vol. 54, 1932).

[8]Velocity Characteristics of Hoods under Suction, by J. M. DallaValle (A.S.H.V.E. TRANSACTIONS, Vol 38, 1932, p. 387).

[9]Low Velocity Exhaust Systems, by Theodore Hatch (*Heating and Ventilating*, October, 1940, p. 27).

[10]Tank Ventilating Power Costs Cut by Low Velocity Systems, by William B. Harris (*Heating and Ventilating*, July, 1942, p. 42).

Large Open Hoods

Large hoods, such as may be used for electroplating and pickling tanks, should be sub-divided so the area of the connecting duct is not less than one-fifteenth of the open area of the hood. Frequently, it will be found necessary to branch the main duct in order to obtain a uniform distribution of flow. *Canopy hoods* should extend 6 in. laterally from the tank for every 12 in. elevation, and wherever possible they should have side and rear aprons so as to prevent short circuiting of air from spaces not

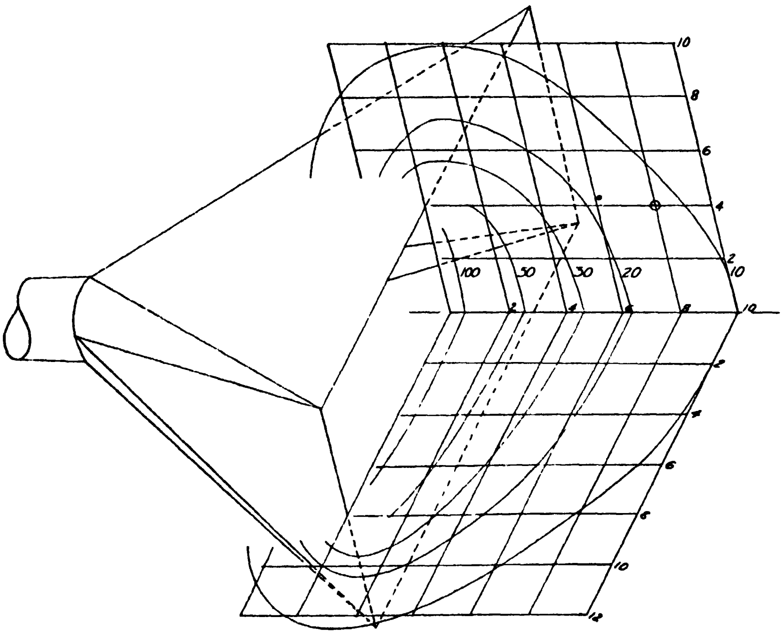

FIG. 1. VELOCITY CONTOURS FOR A RECTANGULAR OPENING WITH A SIDE RATIO OF ONE-HALF. CONTOURS ARE EXPRESSED AS PERCENTAGES OF THE VELOCITY AT THE OPENING

directly over the vats or tanks. In most cases, hoods of this type take advantage of the natural tendency of the vapors to rise, and air velocities may be kept low. Cross drafts from open doors or windows disturb the rise of the vapors and therefore provision must be made for them. The air velocities required also depend upon the character of the vapors given off, cyanide fumes, for example, requiring an air velocity of approximately 75 fpm on the surface of the tank and acid and steam vapors requiring velocities as low as 25 to 50 fpm. The total volume of air flow necessary to obtain these velocities may be approximately determined from the equation:

$$Q = 1.4\,PDV \qquad (4)$$

where

Q = total volume of air handled by hood, cubic feet per minute.
P = perimeter of the tank, feet.
D = distance between tank and hood opening, feet.
V = air velocity desired along edges and surface of tank, feet per minute.

Lateral Exhaust Systems

The lateral exhaust method, as developed for chromium plating[11], is applicable in many instances in preference to the canopy type hoods. The method makes use of drawing air and fumes laterally across the top of vats or tanks into slotted ducts at the top and extending fully along one or more sides of the tanks. The slots are 1 in. wide and for effective ventilation a 2000 fpm exhaust air velocity at the slot face is advisable. In addition, the duct should not be required to draw the air laterally for a distance of more than 18 in. and the level of the solution should be kept 6 to 8 in. below the top of the tanks. If width of tank is over 18 in. a double lateral exhaust may be used with slots on both sides.

It has also been determined that a similar control may be used for tanks wider than 3 ft when the same velocity (2000 fpm) is maintained through a slot which is increased ¼ in. for every foot of width greater than 3 ft. When these slots must be extended more than 6 ft in length some method of spreading the flow is necessary to provide even air flow distribution through the entire slot length. This can be accomplished by tapering the slot, which incidentally will add to the resistance of the system. A more economical approach is to place properly spaced vanes in the side ducts, or to branch the side ducts[12].

The flexible exhaust tube method may be advantageously used for removing dust or fumes. Flexible tubes having one end connected to an exhaust system and a slotted hood attached to the other end may be shaped at will to fit in with industrial processes without affecting the ease of operation. Efficient dust or fume removal may be had with use of relatively small exhaust volumes. This type of system may be used on swing grinders, portable grinding wheels, soldering operations, stone cutting, rock drilling, etc.

Spray Booths

In the design of an efficient spray booth, it is essential to maintain an even distribution of air flow through the opening and about the object being sprayed. While in many instances spraying operations can be performed mechanically in wholly enclosed booths, the volatile vapors may reach injurious or explosive concentrations. At all times the concentrations of these vapors, and particularly those containing benzol, should be kept well below 100 parts per million in the breathing zone of the worker. Spray booth vapors are dangerous to the health of the worker and care should be taken to minimize exposure to them.

It is recommended in the design of spray booths that the exhaust duct be located at the end of the booth opposite the opening. In front of this duct should be placed baffle plates which will cause a uniform air velocity distribution across the frontal area. The air volume should be sufficient to maintain a velocity of not less than 100 fpm over the open area of the booth (150 fpm is preferable where benzol or lead is present in the paint) and the vapors should be discharged through a suitable stack to permit dilution. It is good practice to pass the fumes or vapors through baffle type washers or scrubbers designed for efficient spray removal.

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[11]Health Hazards in Chromium Plating, by J. J. Bloomfield and William Blum (*U. S. Public Health Report*, Vol. 43, No. 26, September 7, 1928).

[12]New Data for Practical Design of Ventilation for Electroplating, by W. P. Battista, Theodore Hatch and Leonard Greenburg (*Heating, Piping and Air Conditioning*, February, 1941, p. 81). Ventilation of Plating Tanks, by Allen D. Brandt (*Heating, Piping and Air Conditioning*, July, 1941, p. 434).

Hoods for Chemical Laboratories

Hoods used in chemical laboratories are generally provided with sliding windows which permit positive control of the fumes and vapors evolved by the apparatus. Their design should offer easy access for the installation of chemical equipment and should be well lighted. Air velocities should exceed 50 fpm when the window is fully open.

Kitchen Hoods

The length and width of kitchen hoods should be such as to extend beyond the extreme projection of the ranges, broilers, etc., over which they are installed. The minimum projection or overlap should be 12 in. Where space conditions permit, range hoods should be about 2 ft high so as to provide a reservoir to confine momentary bursts of smoke and steam until the exhaust system can evacuate the hood. Range hoods should be located as low as possible to increase their effectiveness.

A steel plate placed horizontally within a kitchen hood and located sufficiently high above the level of the bottom of the hood to permit use of a row of lights within the hood along the edge has been found effective in improving the operation. The area between the edges of the plate and the edges of the hood should be selected according to the exhaust volume and velocity required.

In general the amount of air to be exhausted from restaurant range hoods is at the rate of 100 fpm per square foot of face area. Thus, a hood 4.5 ft wide by 30 ft long has a face area of 135 sq ft, which multiplied by 100 fpm velocity results in a total air quantity to be exhausted of 13,500 cfm. In some cases, where the application is principally frying and where it is not practical to install a hood 2 ft high, it is recommended that the face velocity be increased from 100 to 150 fpm, depending on peak load conditions in the kitchen. Exhaust connections to range hoods should always be made at the top and back of hoods, and should be spaced preferably not more than 6 ft apart and be rectangular in shape with the long side parallel to the back of the hood. Exhaust openings into range hoods should be designed to maintain a velocity of 1500 to 1800 fpm.

An approved fire damper with fusible link should be (and is required by code in many states) installed in the main exhaust duct or branch adjacent to the range hood. Should there be more than one hood connected to a common duct, then the branch duct to each hood should be provided with a fire damper. Access doors should be provided at the fire damper for purpose of inspection, cleaning, or for renewal of fusible link. All exhaust piping to range hoods, commonly called grease ducts, should be provided with tight fitting cleanout doors of adequate size to permit easy removal of grease. Some engineers use filters to advantage in hoods which are subject to grease conditions.

Hoods over steam tables should be of similar construction to range hoods. It is good practice to design such hoods with a face velocity of 60 to 70 fpm. Hoods over dishwashing machines are usually relatively small and generally 1500 to 2000 cfm per hood is allowed, which is equivalent to a velocity of approximately 100 fpm per square foot of face area. Range hoods in diet kitchens are constructed the same as restaurant range hoods but with less exhaust air per square foot of face area, depending upon the nature of the food cooked.

Hoods are not often used in private residences unless they are quite large and the consideration of expense is not important. For such

residences the hoods should be designed on the same basis as diet kitchens. Most all residence kitchens can be effectively and economically ventilated by the installation of a built-in kitchen ventilator, which should be located in an outside wall and in close proximity to the kitchen range. It has been found that the capacity of the built-in kitchen ventilator should be at least 350 cfm regardless of the size of kitchen. This can be justified on the basis that the smaller the kitchen the more concentrated the heat will be thus requiring a more rapid rate of air change. Standard size built-in kitchen ventilators are generally available in three sizes, namely 350, 500 and 800 cfm. The proper size to use will depend on design conditions and available wall space.

DUCT SYSTEM DESIGN

In designing a duct system it is necessary to recognize a few fundamental principles (see also Chapter 31). Knowing the quantity of air required, the size of the duct may be computed from Equation 5:

$$A = \frac{Q}{V} \tag{5}$$

where

A = cross-section area of duct, square feet.
Q = air quantity to be handled by the duct, cubic feet per minute.
V = velocity of air, feet per minute.

Air Velocities in Ducts

Where it is necessary to transport the particulate material collected in an exhaust system, minimum carrying velocities must be maintained in the ducts preceding the collector. It has been found that good results are obtained when design air velocities in horizontal runs are not less than 2000 fpm or not greater than 5000 fpm. When the dust being carried is organic and other than wood flour, or similar material, a velocity of 2500 fpm is adequate. Approximate required conveying velocities are given in Table 6.

For duct systems wherein the air has no dust or solid load, a lower velocity is desirable, which may range from 1200 to 2000 fpm. In view of the fact that the horsepower required by a system depends directly on the resistance and the resistance is a function of the velocity, economical design requires velocities of this magnitude.

The equal friction method is generally used for designing a duct system as this insures equal resistance to air flow in all branches throughout the system (see Chapter 31). Long main ducts do not generally provide the most economical layout. Where it is necessary to ventilate a large number of machines, or machines which are widely separated, it is desirable to

TABLE 6. APPROXIMATE CONVEYING VELOCITIES

MATERIAL CONVEYED	DESIGN VELOCITY FPM
Vapors, gases, fumes, very fine dust	2,000
Fine dry dusts	3,000
Average industrial dusts	3,500
Coarse particles	3,500–4,500
Large particles, heavy loads, moist materials	4,500 and over

TABLE 7. GAGES OF METALS FOR EXHAUST SYSTEMS[a]

Diameter of Round Pipe or Greatest Dimension of Rectangular Pipe, Inches	Thickness of Duct Material U. S Gage Number	
	For Highly Abrasive Matter	For Other Matter
Up to 8 inclusive	20	22
Over 8 to 18 inclusive	18	20
Over 18 to 30 inclusive	16	18
Over 30	14	16

[a]Fundamentals of Design, Construction, Operation and Maintenance of Exhaust Systems (*American Foundrymen's Association*, p 53).

locate the fan at approximately the center of the system. With this arrangement it is possible to choose a fan which will deliver the required air quantity against a lower resistance pressure, and this will generally result in a horsepower saving.

When a system carrying dust is designed with an oversize main duct to allow for future extension, the air velocity may be found to be too low to carry the dust, and serious plugging may occur. In this case it is desirable to install an orifice in the end of the pipe to allow for the lower air quantity.

Construction

The interior of all ducts should be smooth and free from obstructions at joints and soldered air-tight. Other sealing mediums are permissible where soldering is impracticable.

Ducts should be constructed of galvanized sheet metal except when the presence of corrosive fumes or gases, temperatures above 400 F, or other factors would make galvanized material impractical. For the usual exhaust systems the metal thicknesses shown on Table 7 are recommended. Elbows and angles should be a minimum of two gages heavier than straight lengths of equal diameter. Hoods should be a minimum of two gages heavier than straight sections of a connecting branch.

Longitudinal joints of ducts should be lapped and riveted or spot-welded on 3-in. centers maximum. Girth joints or ducts should be made with lap in direction of air flow, with 1 in. lap for duct diameters through 19 in. and 1¼ in. lap for diameters over 19 in. Elbows and angles should have an inside or throat radius of two pipe diameters whenever possible. Large radii are recommended for heavy concentrations of highly abrasive dusts. Elbows 6 in. or less in diameter should be constructed of at least 5 sections and if over 6 in. in diameter of 7 sections, with angles pieced proportionally. Hoods should be free of sharp edges or burrs and reinforced to provide necessary stiffness. Transitions in mains and sub-mains should be tapered with a taper 5 in. long for each 1 in. change in diameter whenever possible. All branches should enter the main at the large end of the transition at an angle not to exceed 45 deg or preferably 30 deg. Branches should be connected only to the top or sides of mains, with no two branches entering diametrically opposite to each other. Dead end caps should be provided within 6 in. from last branch of all mains and sub-mains. Cleanouts should be provided every 10 ft and near each elbow, angle, or duct junction in horizontal sections. Ducts should be supported sufficiently to place no loads on connected equipment and to carry weight of a system plugged with material. The maximum distance between supports should be 12 ft for 8 in. or smaller ducts and 20 ft for larger ducts. Six in. minimum clearance should be

Table 8. Loss Through 90-Deg Elbows

Elbow Center Line Radius in Per Cent of Pipe Diameter or Depth	Approximate Loss in Per Cent of Velocity Head
50	75
100	26
150	17
200–300	14

provided between ducts and the ceiling, wall or floor. Blast gates for adjustment of the system should be placed near the connection of a branch to the main and means of locking gates after the adjustments have been made should be included. Rectangular ducts should be used only when clearances prevent the use of round construction. Rectangular ducts should be as nearly square as possible. The weight of metal and the lap, and other construction details, should be the equal of round duct construction having a diameter equal to the longest side. All pipes passing through roofs should be equipped with collars so arranged as to prevent water from leaking into the building.

The main trunks and branch pipes should be as short and straight as possible.

Cleanout openings having suitable covers should be placed in the main and branch pipes so that every part of the system can be easily reached in case the system clogs. Either a large cleanout door should be placed in the main suction pipe near the fan inlet, or a detachable section of pipe, held in place by lug bands, may be provided.

Every pipe should be kept open and unobstructed throughout its entire length, and no fixed screen should be placed in it, although the use of a trap at the junction of the hood and branch pipe is permissible, provided it is not allowed to fill up completely. The passing of pipes through fire-walls should be avoided wherever possible, and floor sweep connections should be so arranged that foreign material cannot be easily introduced into them.

At the point of entrance of a branch pipe with the main duct, there should be an increase in the latter equal to their sum. Some state codes specify that the combined area be increased by 25 per cent. While this is not always good practice and is frequently done at the expense of a reduced air velocity, it is often done where future expansion of the exhaust system is contemplated.

Duct Resistance

The resistance to flow in any galvanized duct riveted and soldered at the joints may be obtained from Fig. 2, Chapter 31. The pressure drop through elbows depends upon the radius of the bend. For elbows whose centerline radii vary from 100 to 300 per cent of pipe diameter, the loss may be estimated from Table 8. It is sometimes convenient to express the resistance of an elbow in terms of an equivalent length of duct of the same diameter. Thus with a center line radius equal to the pipe diameter the resistance is equivalent to a section of straight pipe approximately 10 diameters long, while with a center line radius 1½ times the diameter, the resistance is apparently the same as that of seven diameters of straight pipe.

AIR CLEANING EQUIPMENT

Primary dust separators depend for their air cleaning action on centrifuging out the dust particles or on imparting to the transporting air stream a quick change in direction, accompanied by a sudden reduction in velocity. They may be applied to exhaust systems from polishing and buffing wheels and, in some cases, as a preliminary separator ahead of a final dust arrester.

Dust particles which cannot be caught in inertia type separators are usually removed by one of the following methods:

1. *Filtration.* Of the filter arresters the cloth type, either of the screen, envelope, or bag construction, is most generally used for industrial exhaust systems. Cloth arresters properly engineered for this job and handling dust for which they are suited operate at a good efficiency.

2. *Wet Collectors.* Wet collectors have been in use for many years, but only recently have been introduced for collection of industrial dust from exhaust systems. They are particularly applicable where moisture is present in the air being exhausted. The resistance is relatively low in wet collectors suitable for dust removal and remains constant for a given exhaust system.

3. *High-tension Electrical Precipitation.* High-tension electrical precipitation has been applied to removal of dust, fumes, and mists from air and other gases. (See section on Electric Precipitators in Chapter 28).

4. *Dynamic Precipitators.* This equipment effects removal of entrained dusts by means of a specially designed fan in which the rotating element imparts a centrifugal force to the dust particles. Originally these precipitators were intended for the collection of dry dust, but recently the construction has been modified to permit the introduction of water into the dirty air intake, thus increasing its application to more kinds of dust. Consideration must be given to the type of dust handled[13].

For additional information on dust and cinders, see Chapter 28, Air Cleaning Devices.

RESISTANCE OF SYSTEM

The maintained resistance of the exhaust system is composed of three factors: (1) loss through the hoods, (2) collector drop, and (3) friction drop in the duct system.

The loss through the hoods is usually assumed to be equal to the suction maintained at the hoods. Where possible the resistance of the particular collector to be used should be obtained from the manufacturer.

Friction drop in the pipes must be computed for each section where there is a change in area or in velocity. The velocities should be found in each section of pipe starting with the branch most remote from the fan. The friction drop for these sections can be determined by reference to Table 8 in this chapter and Fig. 2, Chapter 31. Total friction loss in the piping system is the friction drop in the most remote branch plus the drop in the various sections of the main, plus the drop in the discharge pipe.

EFFICIENCY OF EXHAUST SYSTEMS

The efficiency of an exhaust system depends upon its effectiveness in

[13]The National Silicosis Conference Report (*Bulletin No 13*, U S. Department of Labor, February 3, 1937).

TABLE 9. ACCEPTED STANDARDS FOR TOXIC CONCENTRATION OF FUMES, DUSTS AND MISTS[a]

FUMES	MILLIGRAMS PER CUBIC METER
Cadmium oxide	0.1
Chlorodiphenyl	1.0
Lead or oxides of lead	0.15
Mercury or mercury compounds	0.1
Pentachloronaphthalene	0.5
Trichloronaphthalene	5.0
Zinc oxide	15.0
DUSTS	MILLION PARTICLES PER CUBIC FOOT[b]
Asbestos	5
Cement	100
Gypsum	100
Lead or compounds of lead	0.15[c]
Manganese	6[c]
Marble	100
Silica (more than 70 per cent free silicon-dioxide)	5
Silica (more than 10 per cent free silicon-dioxide)	10
Silica (less than 10 per cent free silicon-dioxide)	100
MISTS	MILLIGRAMS PER CUBIC METER
Chromic acid	0.1
Sulphuric acid	5.0

[a]Adapted from Safe Concentrations of Certain Common Toxic Substances Used in Industry, by M Bowditch, C. K. Drinker, P. Drinker, H. A. Haggard, and H. Hamilton (*Journal of Industrial Hygiene and Toxicology*, Vol. 22, No. 6, June, 1940). Industrial Code Bulletin No. 35 (New York State Labor Department). Study of Asbestosis in Asbestos Textile Industry (*U. S. Public Health Bulletin* No. 241, 1938). Chronic Manganese Poisoning in an Ore Crushing Mill (*U. S. Public Health Bulletin* No. 247, 1940).

[b]Determined by light field or equivalent technique.

[c]Milligrams per cubic meter.

reducing the concentration of dusts, fumes, vapors and gases below the safe or threshold limits[14].

Too much emphasis cannot be placed on the necessity of testing exhaust systems frequently by determining the concentration of atmospheric contamination at the worker's breathing level[15]. Commonly accepted values of threshold limits for usual atmospheric contaminants, such as fumes, dusts and mists are given in Table 9. Similar data covering gases and vapor will be found in Chapter 27.

AIR FLOW PRODUCING EQUIPMENT

In any type of exhaust system some form of air flow producing equipment should be required to create the pressure necessary to cause the air to flow through the system to the discharge stack. Natural draft should not be depended on to remove harmful or dangerous gases, fumes, mists, dusts, or other matter when it is imperative that such matter be removed from the work place or room atmosphere.

The principal types of air moving equipment are chimney exhausts,

[14]Criteria for Industrial Exhaust Systems, by J. J. Bloomfield (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 353).

[15]Keeping Dust Under Control, by John M. Kane (presented to the 31st National Safety Congress, October 28, 1942, and reprinted in part in *National Safety News*, January, 1943).

TABLE 10. CORROSION RESISTING MATERIALS FOR EXHAUST SYSTEMS[a]

MATERIAL	ACID[b]							
	Acetic	Chromic	Hydro-chloric	Hydro-fluoric	Nitric	Phos-phoric	Sul-phurous	Sul-phuric
Metals	Dil / Conc	Dil / Conc	Dil. / Conc	Dil. / Conc	Dil. / Conc.	Dil. / Conc.	Dil / Conc	Dil / Conc.
Aluminum	Good	Fair	Poor	No Data	Poor / Good	Poor	Poor	Poor
Magnesium and Alloys	No Data	Good / Poor	No Data	Poor / Good	No Data	No Data	No Data	No Data
Lead and Lead-Coated	Poor	Good	Poor	Poor	Poor	Poor	Good	Good / Poor
Moly Alloy (60 *Ni*—20*Mo* —20 *Fe*)	Good	No Data	Fair	No Data	Poor	Poor	No Data	Good
Monel Metal	Fair	Poor	Fair / Poor	Good[d]	Fair / Poor	Fair	Poor	Good[e] / Poor
Bronze	Poor						Good	
Silicon Iron	Fair / Good	No Data	Fair	Poor	Good	Good	No Data	Good
Stainless Steel[c] (18 *Cr*-8 *Ni*)	Good	Good	Poor	No Data	Good	Poor	Good	Poor / Good[e]
Enameled Steel	No Data	No Data	Good	Poor	Good	Poor	No Data	Good
Miscellaneous								
Asbestos Comp.			Good except against strong acids and alkalies					
Wood	Some woods are decomposed or softened faster than others.							
Rubber					Poor			Poor
Plastics	In general plastics resist weak acids and are decomposed by concentrated acid.							

[a]Standard Practice Sheet No. 115 (Division of Industrial Hygiene, New York State Labor Department).
[b]Acid mists in air are more corrosive than as liquid in storage tank. Galvanized iron not resistant to acid.
[c]Stainless steel of (24 *Cr*—10 *Ni*) fairly resistant at low temperature for HCl and H_3PO_4
[d]Under most conditions
[e]At room temperatures.

venturi ejectors, centrifugal exhaust fans, disc or propeller fans, and axial flow fans[16].

Manufacturers generally provide special fans for the collection of various industrial wastes. These are available for the collection of coal dust, wood shavings, wool, cotton and many other substances. When substances having an abrasive character are conveyed, the fan blades and housing should be protected from wear. This may be accomplished by placing a collector on the negative side of the fan or by lining the housing and blades with rubber.

PROTECTION AGAINST CORROSION

The removal of gases and fumes in many chemical plants requires that metals used in the construction of the exhaust system be resistant to chemical corrosion. A list of the materials which may be used to resist the action of certain fumes is given in Table 10. Hoods and ducts, when short, may frequently be constructed of wood and be quite effective. Rubberized paints are available and may be applied as protective coatings in handling such gases and fumes as chlorine and hydrochloric acid.

[16]The Axial Flow Fan and Its Place in Ventilation, by W. R. Heath and A. E. Criqui (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, February, 1944, p. 107)

CHAPTER 41

Drying Systems

Drying Methods, Radiant-Heat Drying, Conduction or Direct Contact Drying, Convection or Air Drying, Mechanism of Drying, Omissions in the Cycle, General Rules for Drying, Humidity Chart, Dryer Calculations, Design, Estimating Methods

DRYING, in its broader sense, refers to the removal of water, or other volatile liquid from either a gaseous, liquid, or solid material. In practice, the process of direct drying gaseous material is referred to generally as dehumidifying, or condensing, and in some cases chemicals are used in the adsorption or absorption of moisture. The subject of dehumidification is treated in Chapter 23. Drying a liquid is called evaporation or distillation. The common usage of the word *drying* refers to the removal of water or other liquid, such as a solvent, by evaporation from a solid material.

When the solid to be dried contains large amounts of free water, the actual drying process is frequently preceded by the removal of part of the water by some mechanical means, such as filtration, settling, pressing or centrifuging. Removal of as much water as possible by such methods is usually advisable, as the cost of these operations, per pound of water removed, is generally much less than by evaporation. In some drying processes the evaporation of the liquid is accompanied by a chemical change, as in the drying of paint and varnish.

DRYING METHODS

Heat must be supplied in order to dry a solid by evaporation. Since this latent heat is large compared with the specific heat of the materials, drying becomes largely a problem in heat transfer. Hence drying methods are often classified according to the method of heat transfer used, as follows:

1. Radiant-heat drying.
2. Drying by direct contact, and conduction.
3. Convection or air drying.

Radiant-Heat Drying

Drying by sun heat is still practiced where danger of rain is slight, atmospheric pollution is negligible and sufficient time can be allowed.

Radiating surfaces, (heated by steam, electricity or other means), afford a good method of heat distribution and control. Radiant heating sets up convection currents, and in low-temperature dryers only about one-third to one-half of the total heat for evaporation is actually supplied to the material by radiation. At high temperatures the radiation output increases rapidly, according to the *fourth-power law*. The total radiation may be computed by the equations and tables given in Chapter 3. In general, fins and irregular surfaces do not increase radiation, hence the area to be used in calculations is the area of a smooth-surface envelope enclosing the radiating elements.

A certain amount of air circulation is required through a radiant dryer, in order to carry off the vapor.

Conduction or Direct Contact Drying

Drying rolls or drums, flat surfaces, open kettles and immersion heaters are examples of the direct-contact method. Intimate contact of the material with the heating surface is important, and in some cases agitation is desirable to increase the uniformity of heating or to prevent overheating.

Greatest resistance to heat transfer occurs on the air side of the material being dried. The rate of heat transfer from the surface of the heated material to the air, and hence the rate of drying, may be increased by: (a) forced convection or air circulation and (b) vacuum operation to lower the boiling point of the liquid being evaporated.

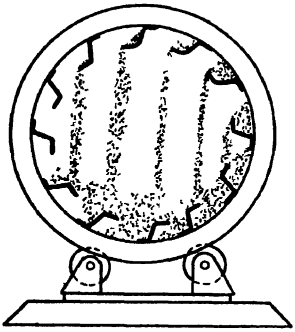

FIG. 1. SECTION OF ROTARY DRYER

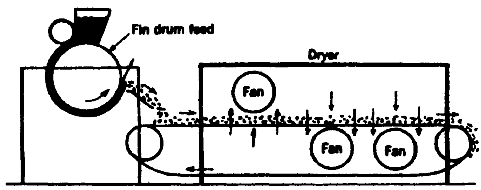

FIG. 2. SECTION OF CONTINUOUS DRYER, BLOW-THROUGH TYPE

Convection or Air Drying

The circulation of heated air or other gases over the material being dried is termed convection drying. Some convection drying occurs in practically all types of dryers, but if the main source of heat is from the air or gases, the dryer may be called a convection dryer. Typical forms or examples may be enumerated:

1. Rotary drum dryers, (Fig. 1).
2. Tunnel or oven dryers, batch type.
3. Tunnel dryers, conveyor type, (Figs. 3 and 4).
4. Tower or column dryers.
5. Through-circulation dryers, (Fig. 2).

In any type of convection dryer the heat transfer and hence the drying depends primarily upon the surface area of material exposed to the air and the velocity of the air over the surfaces (see Dryer Calculations, later).

A general classification of several types of dryers is given in Table 1. The chief basis of classification in this table is that of intermittent or batch operation as opposed to continuous operation. Another important basis of classification would be the method of handling the material to be dried. In an effort to secure maximum contact of the air with the product, as well as uniformity of heating, the effectiveness of cascading the material in an inclined drum or of blowing heated air through a bed of granular or pre-formed material is at once apparent, and where continuous drying at high capacity is required, these types are preferred. Extensive experimental studies on both types are available, (see References).

Simple *drying ovens* are often used for drying smaller quantities of material.

The heat and humidity supply for low temperature work up to 250 F is often steam; steam coils either in the oven or outside, heat the air used for drying. Circulation of heated oil is used to a limited extent, but the danger of leaks is serious, for if the oil is hotter than the flash point, a fire may start if the oil is released to the atmosphere. In many cases where steam is not available, direct or indirect-fired heaters are used with gas or oil as fuel. Indirect heaters should be carefully selected from a standpoint of long life and efficiency. The heat exchange surface should be adequate in area and easily accessible for cleaning and removal. For extremely high temperatures, alloy surface may be used. With direct-fired equipment care must be used in the selection of burners and sufficient com-

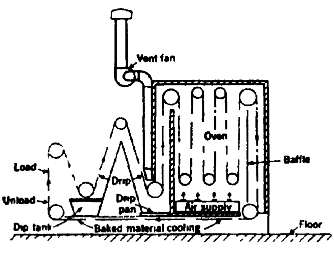

FIG. 3. SMALL PART MULTIPLE PASS OVEN

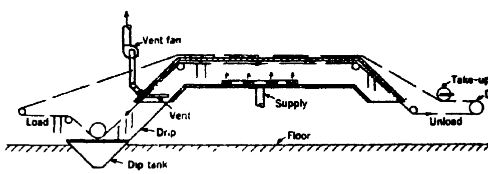

FIG. 4. INCLINED END ENAMELING OVEN

bustion space allowed to insure complete combustion of fuel. Humidity can be obtained in dryers by the use of steam spray, humidifiers, or recirculation.

For low temperature work up to 200 F ovens and dryers are commonly built of two thicknesses of insulating board (fireproof preferred), with air space between. As the temperature increases materials better able to withstand the heat must be used. Metal lined ovens are easy to keep clean, and many high temperature dryers up to 1000 F are made of metal panels with insulation between. Care should be taken to avoid *through metal* (metal extending through the oven wall from inside to out). Batch type ovens are entirely closed while in use and control of air leakage is easily taken care of. In the continuous dryer where the ends are open, heat and air leakage becomes important. Warm air leaking out of the ends of ovens means a heat loss, and often the temperature and humidity outside the oven becomes unbearable. For this reason, inclined or bottom entry ovens are used, as the warm air leakage can be more easily controlled. See Figs. 3 and 4.

MECHANISM OF DRYING

The modern theory of drying may be summed up as follows: Assuming uniform velocity and distribution of air at a constant temperature and humidity over the surface to be dried, the drying cycle will be divided into two distinct stages:

1. Constant rate period.
2. Falling rate period.

The *constant rate period* occurs while the material being dried is still very wet, and continues as long as the water in the material comes to the

TABLE 1. DRYERS FOR EVAPORATION OF WATER

Type	Kind	Materials Handled	Means of Handling	Temp. Range Deg F	Heat Supply	Uses and Remarks
Batch or Intermittent	Com-partment	Paper, Leather, Yarns, Lumber, Foodstuffs	Suspended, Truck, Tray	80 to 180	Steam Coils, Air, Electricity	When production does not warrant continuous drier
	Agitated	Chemicals too sticky for Rotary Drier	Shoveled into Drum or Pan	100 to 330	Water, Steam Jacketed, may have Vacuum on top	Where dust must be saved
	Vacuum	Chemicals, Explosives, Pharmaceuticals, Food Products	Tray, Basket, Tumbling Drum	80 to 300	Water, Steam	Cost of operation high, for expensive materials
Continuous	Tunnel	Ceramics, Chemicals, Lumber, Food Products	Truck, Tray, Belt	100 to 350	Steam Coils, Air, Electricity, Products of Combustion	For high production
	Rotary	Bulk	Cascades through	80 to 500	Air, Steam, Products of Combustion	Where material will stand rough handling and is not subject to balling up
	Drum	Liquids, Slurries	Flowed on Drum, Dry Material Scraped off	to 310	Steam, may have Vacuum on Top	Hygroscopic materials dried with vacuum, and packed immediately
	Cylinder	Paper, Textiles, Chemicals	Continuous Sheets, Endless Chain Belt	to 350	Steam inside of Drum	Where material comes in sheets or rolls, and will stand direct contact with heating surface
	Festoon	Paper, Chemicals	Continuous Sheets, Suspended on Metal Screens	to 200	Air, Steam Coils	Where one side cannot come in contact with supports until dry
	Tower or Column	Grains, Sand	Falls through by Gravity	125 to 250	Air, Steam Coils	Where headroom is available
	Spray	Solutions over 30% Solids	Sprayed into Chamber	120 to 350	Air, Products of Combustion	Drying is almost instantaneous
	Induction	Metals, for removal of traces of Water	Placed in High Frequency Field	to 400	Electricity	Where heating of metal from inside out is important

surface so rapidly that the surface remains thoroughly wet, and evaporation proceeds at a constant rate, precisely as from a free water surface. The material tends to assume a temperature corresponding to the wet-bulb temperature of the surrounding air. But the actual temperature is often slightly higher, due to radiation and conduction from dry surfaces adjoining the material. The constant rate period continues until a time is reached when the moisture no longer comes to the surface as fast as it is evaporated. This point is called the *critical moisture content* in the drying process.

As the drying proceeds, a period of *uniform falling rate* is entered. During this period, the surface of the material is gradually drying out, and

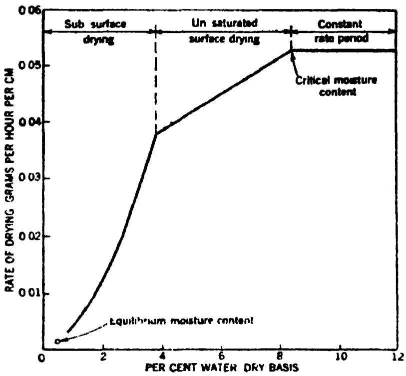

FIG. 5. RATE OF DRYING OF WHITING SLAB

the rate of drying falls as the remaining wet surface decreases in area. This period is also known as unsaturated surface drying.

As drying continues, the surface is completely dry and the water from the interior evaporates and comes through the surface as vapor. As the plane of water recedes, the diffusion of the vapor becomes more difficult and hence the period is known as *varying falling rate period*, or sub-surface drying.

Drying ceases when the *equilibrium moisture content* has been reached. The final moisture content of the product depends on the relative humidity of the air in contact with it. Equilibrium is established when the vapor pressure of the moisture in the air and the vapor pressure of the moisture in the material are equal. The equilibrium moisture content varies with the hygroscopic properties of the material, (see table of Regain of Hygroscopic Materials, Chapter 39).

The drying of a slab of whiting is shown in Fig. 5 and illustrates the principles referred to previously. The factors affecting the variations of drying rates during the periods mentioned are outlined in Table 2.

Omissions in the Cycle

Many solids, such as lumber, are so dry at the beginning of the drying operation that the constant rate period of free surface evaporation does not occur. Frequently the surface of the material is dry enough so that no surface drying can take place, in which case only the final stage of sub-surface drying is involved. In other instances, the critical moisture con-

tent of a wet solid is sufficiently low that sub-surface drying starts almost immediately after the conclusion of the constant rate period. Thus the intermediate state of unsaturated surface drying does not occur and the drying is of the sub-surface type during practically the whole of the falling rate period. With other kinds of material, particularly thin sheets, such as newsprint paper, sub-surface drying may occur at such a low moisture content that it is not encountered in commercial work, the falling rate period being confined in practice to unsaturated surface drying.

Table 2. Factors Influencing Drying

Factor	Drying Period	
	Constant Rate, Unsaturated Surface	Sub-Surface
Temperature	Increase in temperature increases drying rate	Increase in temperature increases drying rate, because with decreased viscosity, capillary flow is increased.
Humidity	Drying rate increases as humidity is decreased	No effect until equilibrium content is reached; drying then ceases
Air Velocity	Drying rate varies approximately as the 0.6 power of the velocity	No effect
Air Direction	Drying rate increases, the more nearly the air blows perpendicular to surface; for dead air film becomes thinner	No effect
Thickness of Material	Drying rate is not affected by the thickness	Drying rate varies inversely as the square of the thickness

GENERAL RULES FOR DRYING

Temperature

The highest temperature possible should be used because of faster drying and smaller requirements for ventilation. The amount of moisture that can be carried by a pound of air increases rapidly with rise in temperature as shown in the humidity chart of Fig. 6. Too high a temperature may cause spoilage of materials; many materials calcine or change their chemical properties if heated too hot; gypsum and glauber salts lose some of the chemically combined water, fall apart, and change their chemical properties. Too high or rapid rise in temperatures in drying lumber or ceramics may create a liquid vapor tension within the material so high that the cells explode, causing permanent injury to the fiber. If too high a temperature is used on some chemicals, they begin to react exothermally; a temperature rise and chemical action from within will burn the materials, *e.g.*, bakelite products, gunpowder, etc. During the constant rate period of drying, the material heats only to the wet-bulb temperature of the surrounding air, consequently high temperatures will not injure the material in this stage.

Humidity

Moisture in the drying air may be very important. Many materials tend to case-harden, dry on the outside, forming a skin which retards the moisture flow from the inside to the surface, or stops it completely, and so

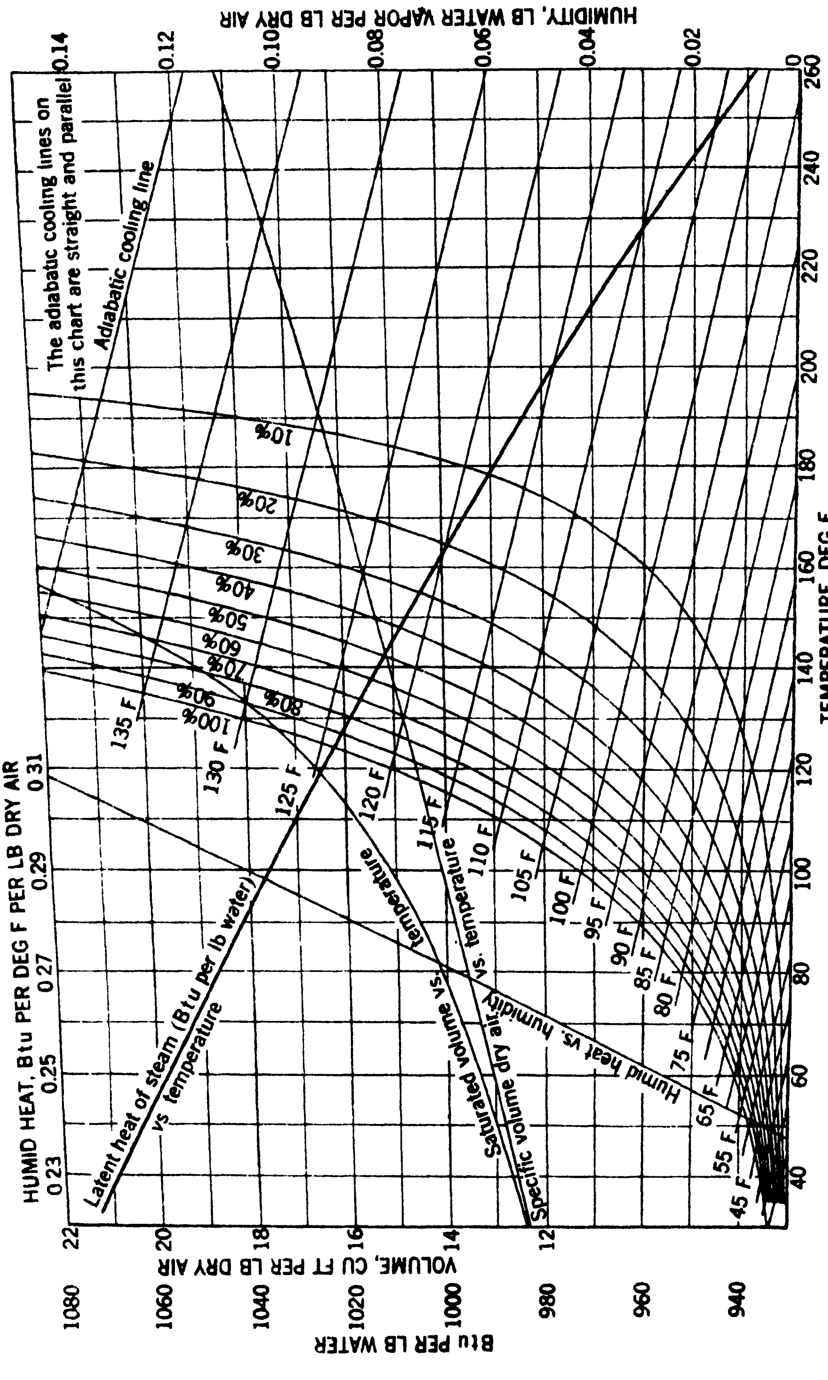

FIG. 6. HUMIDITY CHART

increases the drying time very much or causes a change of the physical properties of the material. It is often necessary to add humidity to the air in the initial stage of drying. Lumber case-hardens, cracks, and warps if the outside is dried too fast. Ceramics crack if not heated through before drying commences. Elastic materials warp or crack if not evenly dried. Many paints case-harden if not dried under high humidity.

On the other hand, in the case of those materials whose physical or chemical properties require that they be dried at relatively low temperatures, high humidity tends to retard drying in the first stage and may even stop it altogether in the final stage. Where drying temperatures below 120 to 140 F are used, the drying rate may be highly dependent on atmospheric humidity conditions. In such instances it is often desirable to dehumidify the air entering the dryer during periods of high atmospheric humidity; where a high degree of uniformity is required, it is often possible to secure complete independence of atmospheric conditions by recirculating the air in a closed system which includes a suitable dehumidifier. For this purpose absorptive dehumidifying systems have the advantage of accomplishing the desired reduction of humidity without appreciably elevating or lowering the dry-bulb temperature of the air; for this reason after-cooling is not required, and reheating is reduced to a minimum. Dehumidifying systems are described in Chapter 23.

Air Circulation

As noted under Mechanism of Drying, air velocity is more important in the first two stages of drying than in the last, and for this reason zone drying in continuous dryers is frequently considered. It permits accurate regulation of temperature, humidity, and velocity in the different zones. High velocity results in more rapid drying, more even distribution of temperature and consequently more even drying in the first period. Too high a velocity may be detrimental because of excessive power needed for creating it, or because the material may blow away if it is light and fluffy. In the drying of paints, varnishes, and enamels, high velocity or improper distribution of the air even with the use of filters, may cause dust already in the dryer to be blown against the material, ruining the finish.

DRYER CALCULATIONS

The fundamental calculations for the design and performance of dryers are based on the thermodynamics of air and water mixtures treated in Chapter 1, and the fundamentals of heat transfer treated in Chapter 3. For the humidity calculations a high-temperature psychrometric chart is given in Fig. 6. In addition to the fundamental heat transfer calculations of radiation, conduction and convection, the heat losses through the walls of the dryer will be computed by the methods illustrated in Chapter 4 and Chapter 31. Data on radiation calculations are given in Chapter 45.

Where products of combustion are used directly in a dryer, a knowledge of the properties of fuels and combustion products is important. Data on fuels and combustion are given in Chapter 8. For determining the heat available in products of combustion, a specific heat of 0.25 Btu per pound per degree Fahrenheit may be used.

The calculations for drying during the constant-rate period are different from those applying to the falling-rate period.

Constant-Rate Period

The rate of drying by air passing over a wet surface is directly proportional to the vapor pressure difference, and also proportional to the

0.8 power of the air velocity. For practical calculations the wet surface is assumed to attain the wet-bulb temperature of the air passing over it, and evaporation takes place at constant rate under equilibrium conditions. The equation may then be expressed in three forms:

$$R = C\,A\,V^{0.8}(\Delta P) \qquad (1)$$

$$R = C'\,A\,V^{0.8}(\Delta H) \qquad (2)$$

$$R = C''\,A\,V^{0.8}(\Delta T) \qquad (3)$$

where

R = rate of drying during constant-rate period, pounds of moisture per hour.

A = area of bed or material in contact with air, square feet.

V = air velocity over material, feet per minute.

ΔP = difference between vapor pressure at wet-bulb (surface) temperature and at dew-point of air.

ΔH = difference between humidity ratio of saturated air, at the surface temperature, and the actual humidity ratio of the air stream, pounds of water per pound of dry air.

ΔT = difference between dry-bulb and wet-bulb temperatures of air, *i.e.*, the wet-bulb depression.

C, C', C'' = proportionality constants (for numerical values consult references).

These equations are useful mainly for computing the effects of changes in operating conditions, such as changes in air velocity, air temperature, humidity and surface area. The equations assume that the material is in equilibrium at the wet-bulb temperature of the air. If equilibrium has not been reached, or if heat is being added to the charge by radiation or conduction, such conditions must be taken into account. For large tray dryers or continuous surfaces, the logarithmic mean difference should be substituted for the simple difference in ΔP, ΔH and ΔT.

When the constant of proportionality is known for a given set of conditions, Equations 1, 2 or 3 may be applied for basic design, as illustrated in *Example 1*.

Example 1. Compute the rate of drying of a granular material initially 35 per cent moisture (dry basis), if the material is spread in trays and is to be dried by blowing air horizontally over the surface at 1000 fpm. The air is 140 F dry-bulb, 90 F wet-bulb. Density of the dry material is 85 lb per cubic foot, and the drying constant C'', in Equation 3, has been found to be about 1/25,000. Find the size of dryer for a capacity of one ton per hour (dry basis), and the time required for drying each batch from 35 to 10 per cent moisture content, if the material is spread in trays, in a layer one inch thick. (The critical moisture content of the material is below 10 per cent, hence the drying is at constant rate.)

Solution: Assume that the surface of the material attains the wet-bulb temperature of the air, then $\Delta T = 140 - 90 = 50$ F. The rate of drying by Equation 3 is:

$$R = C''\,A\,V^{0.8}\,\Delta T = \frac{1000^{0.8} \times 50}{25{,}000} = 0.50 \text{ lb of water per hour per square foot of surface.}$$

The total water evaporated per square foot of surface is:

$$W = \frac{85}{12}(0.35 - 0.10) = 1.77 \text{ lb (per batch).}$$

Then the time required per batch is:

$$T = \frac{1.77}{0.50} = 3.54 \text{ hr.}$$

The size of dryer required to dry the material at the rate of one ton of dried material per operating hour will be:

$$A = \frac{2000 \times 3.54}{85/12} = 1000 \text{ sq ft, total area of trays.}$$

Falling-Rate Period

The *critical moisture content* marks the end of the constant-rate period and the beginning of the falling-rate period. This falling rate may be due to the fact that the surface is no longer completely wetted, or it may result from a condition in which the moisture cannot reach the surface as fast as it can be evaporated. When this high resistance to capillary flow and diffusion is the governing factor in the drying process, the time of drying increases rapidly with the thickness of the material. During constant-rate drying the time required is directly proportional to the thickness of the bed (see *Example 1*), while the time required for drying during the falling-rate period is often proportional to the square of the thickness of the material.

Actual calculations of drying during the falling-rate period are not highly satisfactory because of the number of variables. It has been demonstrated empirically that the rate of drying is approximately proportional to the free water content of the material.

An approximate value of the critical moisture content which marks the beginning of the falling-rate period may be obtained from Table 3.

TABLE 3. CRITICAL MOISTURE CONTENT FOR VARIOUS MATERIALS[a]

Material	Critical Moisture Content, Per Cent
Sand	3 to 20
Clay; soil; pigments	5 to 40
Paper, book or coated	30 to 40
Paper, newsprint	60 to 70
Paper pulp and pulp-boards (wallboard)	75 to 150
Leather (heavy)	90 to 120

[a]The critical moisture content is expressed as weight of water in per cent of weight of dry material

DESIGN

In all drying problems, data regarding temperatures, time, and humidity must be obtained by experiment or previous experience. Experiments are best performed at the temperatures, humidities, and velocities to be actually used in the full sized dryer, and with full size samples.

The following nomenclature and explanation of terms will be used in the discussion of design calculations:

H = humidity ratio of air, pounds of water vapor per pound of dry air.
G = pounds of dry air supplied to the dryer per unit of time.
S = pounds of stock dried per unit of time in a continuous dryer.
S' = pounds of stock charged per batch to a discontinuous dryer.
Θ = time.
Q = total heat supplied to the dryer.
t = air temperature.
t' = stock temperature.
t'' = average stock temperature over short time interval, in a batch dryer.
t_w = wet-bulb temperature.
s' = specific heat of the stock.
B = total radiation and conduction losses per unit time.
w = pounds of water per pound of dry stock.

r = heat of evaporation of water.

s = humid heat of air, *i.e.*, heat necessary to raise 1 lb of dry air + H lb of steam 1 F.

Subscript (1) designates conditions at the point where the material in question (air or stock) enters, and (2) where it leaves the dryer.

Air dryers may be divided into two classes, those in which ***all moisture*** evaporated from the stock *leaves the dryer as vapor* in the effluent air, and those in which *part or all* of the moisture *is condensed* from the air ***in the drying equipment itself.*** In any continuously operating dryer of the first type the relation between moisture content of the stock and quantity of air required for the drying operation is given by the equation:

$$G\,(H_2 - H_1) = S(w_1 - w_2) \tag{4}$$

In discontinuous dryers, *e.g.*, compartment dryers, the drying operation is given by the equation:

$$G\,(H_2 - H_1) = S'\,\frac{dw}{d\Theta} \tag{4a}$$

In the continuous dryer, the heat consumption per unit time is:

$$\frac{Q}{\Theta} = Gs_1(t_2 - t_1) + G(r_2 + t_2 - t'_2)\,(H_2 - H_1) + S(t'_2 - t'_1)\,(s' + w_1) + B \tag{5}$$

Equation 5 assumes continuity of operation. For charge or batch operations, the total time of the drying cycle may be broken up into a number of periods, sufficiently short so that over each period average values of t, t' and H may be employed provided the third term of the right hand member of the equation is modified to read:

$$S'\,(t''_2 - t''_1)\,(s' - w_1)$$

and in the second term t'_2 be replaced by

$$\frac{t'_1 + t''_2}{2}$$

Theoretically these periods should be very short and the equation integrated. Practically the error introduced by using a small number of long periods and employing average values of the variables over each, rarely introduces serious error. The evaluation of Equation 4a may be approximated in a similar manner.

The first term of the right hand member of Equation 5 represents heat lost as sensible heat in the effluent air. In many drying operations this becomes excessive. Each pound of air supplied should remove the maximum amount of moisture. This is best accomplished by bringing the air into contact with the stock with sufficient intimacy so that the air leaving the dryer is saturated, or nearly so. Counter-current as against parallel flow of air and stock gives rise to optimum operating conditions, resulting in a minimum quantity of air required (G), and a corresponding minimum loss, as sensible heat, in the exit air. Similarly, continuous operation is superior to intermittent operation.

Despite the fact that the sensible heat loss increases with the rise in temperature of the air, the percentage of heat lost from this source decreases, provided the increase in moisture carrying capacity of the air, due to high temperature, is actually utilized. To secure maximum thermal efficiency in drying, a high drying temperature and high saturation of the outlet air are imperative.

Ventilation Phase

The technique of attack of the *ventilation phase* of a drying problem is best made clear by an illustration. Assume that a material containing 40 per cent moisture is to be dried until this quantity of moisture is reduced to 5 per cent by weight. The material will stand an air temperature of 150 F and it is possible to provide sufficiently good contact between the material and the drying air so that the effluent air can be brought up to 50 per cent humidity at 150 F. The dryer is to use room air, the temperature and humidity of which may be assumed to average 70 F and 50 per cent. A counter-current dryer will be employed and the air in this dryer will be kept at a substantially constant temperature of 150 F by heaters thermostatically controlled. The stock enters at 70 F, rises quickly to the wet-bulb temperature of the air, with which it is in contact, and is found experimentally to maintain wet-bulb temperature until the moisture content has fallen to 20 per cent From this point its temperature rises progressively as it dries. In this range the difference in

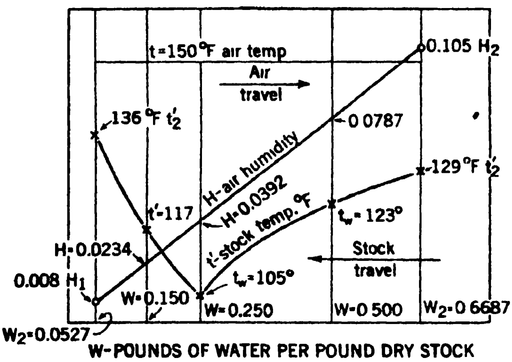

FIG. 7. TEMPERATURE HUMIDITY RELATIONS IN A DRYER

temperature between stock and air, divided by the wet-bulb depression, may be assumed proportional to the moisture content.

The moisture content of the entering stock, in the units here employed, is:

$$w_1 = \frac{40 \text{ per cent water}}{60 \text{ per cent dry stock}} = 0.6667: w_2 = \frac{5 \text{ per cent water}}{95 \text{ per cent dry stock}} = 0.0527$$

$w_1 - w_2 = \Delta w = 0.614$ lb water evaporated per pound of dry stock. Since the air leaving the dryer is 50 per cent saturated at 150 F from Fig. 6, $H_2 = 0.105$. Similarly, $H_1 = 0.008$, corresponding to 50 per cent humidity at 70 F. Consequently $H_2 - H_1 = \Delta H = 0.097$ lb water evaporated per pound dry air.

An analysis of Equation 4 shows that (H) is linear in w. Hence, one can construct on Fig. 7, the line marked (H) being drawn connecting the initial and final points just computed.

Since the air leaving the dryer has a temperature of 150 F and a humidity of 0.105, Fig. 6 shows that its wet-bulb temperature is 129 F. This is plotted at the right hand side of Fig. 7. Since the stock maintains a wet-bulb temperature down to 20 per cent moisture, where $w = 0.25$, the corresponding humidity can be computed by the use of Equation 4 or by reading directly from the diagram, the value being 0.0392. Fig. 6 shows that the corresponding wet-bulb temperature is 105 F. Any

intermediate point on the wet-bulb temperature curve can be calculated similarly. The points for $w = 0.5$ are shown in Fig. 7.

Below the point, $w = 0.25$, the temperature of the stock begins to rise appreciably above the wet-bulb temperature. Its temperature at any given point in this range, for example at $w = 0.15$, may be computed as follows: At this point, $H = 0.0234$ (from Equation 4) and from Fig. 6, $t_w = 95$ F. Hence the wet-bulb depression, $t - t_w = 150 - 95 = 55$ F.

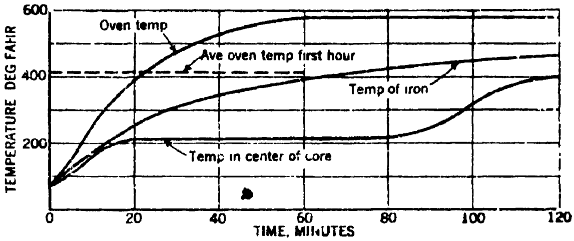

FIG. 8. CORE DRYING TIME TEMPERATURE RELATIONS

The assumption made regarding the relation between stock temperature and moisture content in this range may be formulated:

$$\frac{\Delta t'}{t - t_w} = \frac{w}{0.25}$$

At the point $w = 0.15$, $\Delta t' = 33$ F, $t' = 117$ F. The temperature of the stock leaving the dryer, similarly computed, is 136 F.

Fig. 7 thus computed gives in graphical form the information as to the temperature humidity relationships in the dryer. The air requirements can be computed by Equation 4. Thus, per 100 lb of dry stock, it is necessary to supply 633 lb of dry air. Furthermore, since from Fig. 6 it is seen that the volume of 50 per cent saturated air at 70 F, is 13.55 cu ft per pound; 8580 cu ft of room air must be supplied per 100 lb dry stock. Similarly, since the volume of 50 per cent saturated air at 150 F is 18.0 cu ft per pound, the volume of hot wet air discharged from the dryer is

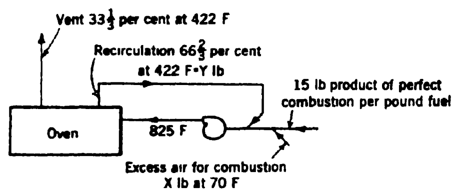

FIG. 9 CORE DRYING DIAGRAM OF COMBUSTION PRODUCTS AND AIR

11,400 cu ft per 100 lb of dry stock. Finally, the heat necessary to supply to the dryer, as a whole, or to any section of it, may be computed from Equation 5.

High Temperature Dryer

In the design of a high temperature dryer unit a method of approach to the necessary calculations involved is outlined as follows:

Example 2. Cores 4 and 5 in. thick are to be dried by heating to a temperature at 400 F. An intermittent type box oven is to be used, size 12 x 14 x 10 ft with 856 sq ft surface having an average heat transfer of 0.3 Btu per square foot per degree per hour.

Drying time as determined by test is 2 hr (Fig. 8). Cores weighing 6 tons, and 15-ton steel plates, trucks etc. are delivered to the dryer at 70 F. The oven is heated by an external heater; the products of combustion and 66⅔ per cent recirculated air will be delivered to the oven at 825 F. Fuel oil of 19,980 Btu gross and 18,830 Btu per pound net heating value, weighing 6.75 lb per gallon and having 15 lb product per pound fuel for perfect combustion. Cores consist of 91 per cent sand, 3 per cent oil binder, and 6 per cent water.

Solution. Heat required per ton of cores:

	Lb Material × Temp. Rise × Sp. Ht.	=	Btu
Sand	$0.91 \times 2{,}000 \times (400 - 70) \times 0.2$	=	120,120
Binder	$0.03 \times 2{,}000 \times (400 - 70) \times 0.4$	=	7,920
Water heating	$0.06 \times 2{,}000 \times (212 - 70) \times 1.0$	=	17,040
Water evaporation	$0.06 \times 2{,}000 \times$ 970 (Fig. 6)	=	116,520
Water superheating (approx. 50 per cent reaches 575 F)	$= 0.5 \times 0.06 \times 2{,}000 \times (575 - 212) \times 0.45$	=	9,800
Total Heat			271,400 Btu

Heat in 1 lb fuel oil	=	18,830 Btu
Heater Loss (10 per cent)	= 1883	
Duct Loss (5 per cent)	= 942	2,825 Btu
Btu available to heat oven		16,005 Btu
Heat content of gases in 1 lb fuel oil at 825 F is 205 Btu		
Sensible heat in products of perfect combustion	= (15 lb + 205) =	3,075 Btu
Btu to heat air X and Y (Fig. 9).	=	12,930 Btu

$$Y\,(S_{825} - S_{422}) + X\,(S_{825} - S_{70}) = 12{,}930 \qquad (6)$$

$Y = 2\,(X + 15)$ for 66.7 per cent recirculation

where

S = heat content of air at temperature noted taken from Fig. 6.

(Recirculation and exhaust contains water vapor, products of combustion, and a greater portion of air. Heat capacities of all vary so little that they have all been assumed to be air).

$S_{825} - S_{422} = 190 - 91 = 99$
$S_{825} - S_{70} = 190 - 8.6 = 181.4$

Substituting values of Y, H, etc. in Equation 6,

$(2\,X + 30)\,99 + 181.4\,X = 12{,}930$
$X = 26.3$ lb excess air.
$Y = 82.6$ lb recirculating air.

Total = 26.3 + 82.6 + 15 = 123 9 lb air and products of combustion circulated per pound fuel burned.

Heat in air exhausted from oven at 422 F per pound fuel burned = $0.333 \times 123.9 \times (S_{422} - S_{70}) = 41.3\,(91 - 8.6) = 3{,}400$ Btu.

Btu available for heating material = 16,005 − 3,400 = 12,605 Btu per pound fuel.

Fuel used in first hour = 2,180,470 ÷ 12,605 = 173 lb = 25.6 gal.

During the second hour the heater capacity will be much greater than required. If an automatic oven temperature control operates on the oil supply, the delivery temperature of the air entering the oven and the quantity of oil burned will decrease, the air supply being constant.

Heat in air exhausted = $41.3\,(S_{575} - S_{70}) = 41.3\,(127 - 8.6) = 4{,}880$ Btu per pound fuel.

Heat available for heating material = 16,005 − 4,880 = 11,125 Btu.

Fuel used in second hour = 1,072,008 ÷ 11,125 = 96.5 lb oil = 14.3 gal.

Total oil used per load = 25.6 + 14.3 = 39.9 gal.

HEATING LOAD FIRST HOUR

	HEATED TO			BTU
Sand	212 F	$\frac{142}{330} \times 120{,}120$	=	51,688
Binder[a]	212 F	$\frac{142}{330} \times 7{,}920$	=	3,408
Water	212 F		=	17,040
Evaporation	66.7%	$0.667 \times 116{,}520$	=	77,680
Superheat	66.7%	$0.667 \times 9{,}800$	=	6,530
Total Per Ton				156,346
For 6 ton		$6 \times 156{,}346$	=	938,076
Steel plates	390 F	$320 \times 30{,}000 \times 0.12$	=	1,152,000
Radiation[b]	422 F Avg	$352 \times 856 \times 0.30$	=	90,394
Total				2,180,470

HEATING LOAD SECOND HOUR

Sand	400 F	$\frac{188}{330} \times 120{,}120$	=	68,432
Binder[a]	400 F	$\frac{188}{330} \times 7{,}920$	=	4,512
Water				
Evaporation	33 3%	$0.333 \times 116{,}520$	=	38,840
Superheat	33 3%	$0.333 \times 9{,}800$	=	3.270
Total Per Ton				115,054
For 6 ton		$6 \times 115{,}054$	=	690,324
Steel plates	460	$70 \times 30{,}000 \times 0.12$	=	252,000
Radiation[b]	575	$505 \times 856 \times 0.30$	=	129,684
Total				1,072,008

[a]Binder oxidizes and liberates heat, which is neglected in this calculation

[b]Average value of coefficient is less than 0.3 because oven is not up to 575 F. This is neglected. 422 F is arrived at by taking area under curve as compared to area under 575 F ordinate

ESTIMATING METHODS

Values based on practical experience are available for rough estimating of drying problems. The temperature will drop approximately 8.5 F per grain of water evaporated per cubic foot of air (measured at 70 F) or approximately 0.62 F per pound of air at any temperature. Air will drop 55 F per cubic foot for each Btu extracted. Generally air will absorb from 2 grains to 5 grains per cubic foot of air in one passage through an air dryer, depending on the temperature and the degree of contact with the material. The amount of steam required to evaporate a pound of water will vary from 1.5 lb to a more usual figure of from 2.5 to 3 lb of steam per pound of water evaporated.

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CHAPTER 42

Natural Ventilation

Wind Forces, Stack Effect, Openings, Windows, Doors, Skylights, Roof Ventilators, Stacks, Principles of Control, General Rules, Measurements, Dairy Barn Ventilation, Garage Ventilation

VENTILATION by natural forces finds application in industrial plants, public buildings, schools, dwellings, garages, and in farm buildings.

The natural forces available for moving air into, through and out of buildings are: (*a*) wind forces, and (*b*) the difference in temperature between the air inside and outside a building. The air movement may be caused by either of these forces acting alone or by a combination of the two, depending upon atmospheric conditions, building design and location. The ventilating results obtained will vary, from time to time, due to variation in the velocity and direction of the wind and the heat generated in the building. The arrangement, location, and control of the ventilating openings should be such that the two forces act cooperatively rather than in opposition.

WIND FORCES

In considering the use of natural wind forces for producing ventilation, account must be taken of: (1) average wind velocity, (2) prevailing wind direction, (3) seasonal and daily variations in velocity and direction, and (4) local wind interference by nearby buildings, hills or other obstructions of similar nature.

Values are given in Table 1, Chapter 7 for the average summer wind velocities and the prevailing wind directions in various localities throughout the United States, while Table 1, Chapter 6, lists similar values for the winter. In almost all localities the summer wind velocities are lower than those in the winter, and in about two-thirds of the localities the prevailing direction is different during the summer and winter. While the tables give no average velocities below 5 mph, there will be times when the velocity is lower, even in localities where the seasonal average is considerably above 5 mph. There are relatively few places where the velocity falls below one half of the average for many hours per month. Consequently, if the natural ventilating system is designed for wind velocities of one-half of the average seasonal velocity, it should prove satisfactory in almost every case.

Equation 1 may be used for calculating the quantity of air forced through ventilation openings by the wind, or for determining the proper size of such openings to produce given results:

$$Q = EAV \tag{1}$$

where

Q = air flow, cubic feet per minute.

A = free area of inlet openings, square feet.

V = wind velocity, feet per minute, = miles per hour $\times$ 88.

E = effectiveness of openings. (E should be taken at 0.50 to 0.60 for perpendicular winds and 0.25 to 0.35 for diagonal winds[1].)

[1]Predetermining Airation of Industrial Buildings, by W. C. Randall and E. W. Conover (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p 605).

The accuracy of the results obtained by the use of Equation 1 depends upon the placing of the openings, as the formula assumes that ventilating openings have a flow coefficient slightly greater than that of a square-edged orifice. If the openings are not advantageously placed with respect to the wind, the flow per unit area of the openings will be less and, if unusually well placed, the flow will be slightly more than that given by the formula. Inlets should be placed to face directly into the prevailing wind, while outlets should be placed in one of the five places listed:

1. On the side of the building directly opposite the direction of the prevailing wind.

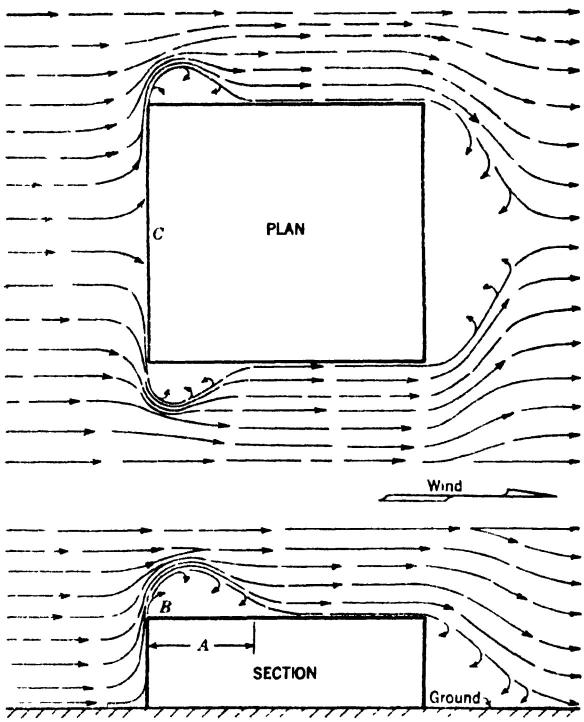

FIG. 1. THE JUMP OF WIND FROM WINDWARD FACE OF BUILDING. (*A*—LENGTH OF SUCTION AREA; *B*—POINT OF MAXIMUM INTENSITY OF SUCTION; *C*—POINT OF MAXIMUM PRESSURE)

2. On the roof in the low pressure area caused by the jump of the wind (see Fig. 1).
3. On the sides adjacent to the windward face where low pressure areas occur.
4. In a monitor on the side opposite from the wind.
5. In roof ventilators or stacks.

TEMPERATURE DIFFERENCE FORCES[2]

The stack effect produced within a building when the outdoor temperature is lower is due to the difference in weight of the warm column of air within the building and the cooler air outside. The flow due to stack effect is proportional to the square root of the draft head, or approximately:

$$Q = 9.4\,A\sqrt{H\,(t - t_o)} \tag{2}$$

[2]Neutral Zone in Ventilation, by J. E Emswiler (A.S.H.V.E TRANSACTIONS, Vol. 32, 1926, p 59).

where

Q = air flow, cubic feet per minute.
A = free area of inlets or outlets (assumed equal), square feet.
H = height from inlets to outlets, feet.
t = *average* temperature of indoor air in height H, degrees Fahrenheit.
t_o = temperature of outdoor air, degrees Fahrenheit.
9.4 = constant of proportionality, including a value of 65 per cent for effectiveness of openings. This should be reduced to 50 per cent (constant = 7.2) if conditions are not favorable.

HEAT REMOVAL

In problems of heat removal, knowing the amount of heat to be removed and having selected a desirable temperature difference, the amount

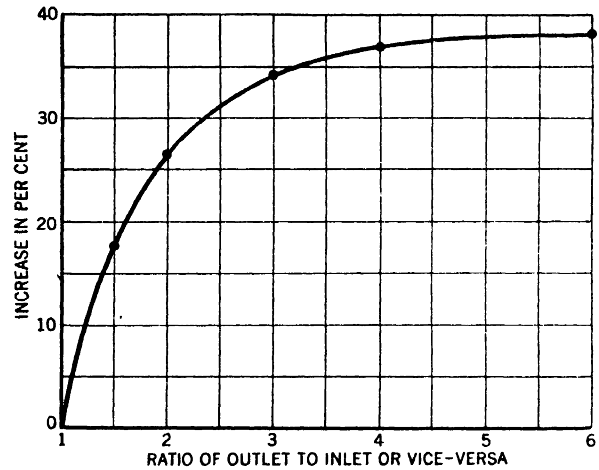

FIG. 2. INCREASE IN FLOW CAUSED BY EXCESS OF ONE OPENING OVER ANOTHER

of air to be passed through the building per minute to maintain this temperature difference can be determined by means of Equation 3.

$$H = 0.0175\, Q\, (t - t_o) \tag{3}$$

where

H = heat removed, Btu per minute.
Q = air flow, cubic feet per minute.
$t - t_o$ = inside-outside temperature difference, degrees Fahrenheit

EFFECT OF UNEQUAL OPENINGS

The largest flow per unit area of openings is obtained when inlets and outlets are equal, and the equations given previously are based on this condition. Increasing outlets over inlets, or vice-versa, will increase the air flow, but not in proportion to the added area. When solving problems having an unequal distribution of openings, use the smaller area, either inlet or outlet, in the equations and add the increase as determined from Fig. 2.

COMBINED FORCES OF WIND AND TEMPERATURE

Equations for determining the air flow due to temperature difference and wind have already been given. It must be remembered that when

both forces are acting together, even without interference, the resulting air flow is not equal to the sum of the two estimated quantities. The flow through any opening is proportional to the square root of the sum of the forces acting on that opening.

When the two forces are about equal in intensity and the ventilating openings are operated so as to coordinate them, the total air flow through the building is about 10 per cent greater than that produced by either force acting independently under conditions ideal to that force. This percentage decreases rapidly as one force increases over the other and the larger force will predominate.

The wind velocity and direction, the outdoor temperature, or the indoor distribution, cannot be predicted with certainty, and refinement in calculations is not justified; consequently, a simplified method can be

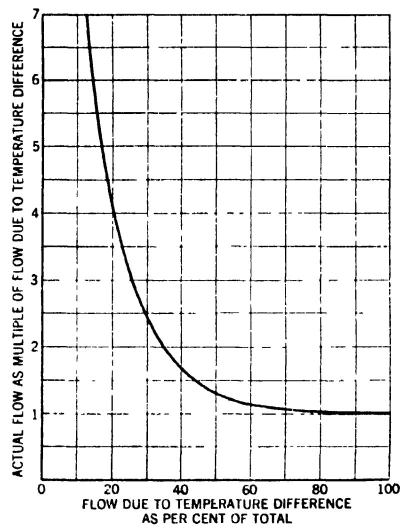

Fig 3. Determination of Flow Caused by Combined Forces of Wind and Temperature Difference

used. This may be done by using the equations and calculating the flows produced by each force separately under conditions of openings best suited for coordination of the forces. Then, by determining, as a percentage, the ratio of the flow produced by temperature difference to the sum of the two flows, the actual flow due to the combined forces can be approximated from Fig. 3.

Example 1. Assume a drop forge shop, 200 ft long, 100 ft wide, and 30 ft high. The cubical content is 600,000 cu ft, and the height of the air outlet over that of the inlet is 30 ft. Oil fuel of 18,000 Btu per pound is used in this shop at the rate of 15 gal per hour (7.75 lb per gal). Desired summer temperature difference is 10 deg and the prevailing wind is 8 mph perpendicular to the long dimension What is the necessary area for the inlets and outlets, and what is the rate of air flow through the building?

Solution for Temperature Difference Only. The heat $H = \dfrac{15 \times 7.75 \times 18{,}000}{60} =$ 34, 875 Btu per minute.

By Equation 3, the air flow required to remove this heat with an average temperature difference of 10 deg is:

$$Q = \frac{H}{0.0175\ (t - t_o)} = \frac{34{,}875}{0.0175 \times 10} = 199{,}286 \text{ cfm.}$$

This is equal to about 20 air changes per hour. From Equation 2 the inlet (or outlet) opening area should be:

$$A = \frac{Q}{9.4\sqrt{H\ (t - t_o)}} = \frac{199{,}286}{9.4\sqrt{30 \times 10}} = 1224 \text{ sq ft.}$$

The flow per square foot of inlet or outlet would be 199,286 ÷ 1224 = 163 cfm with all windows open.

Solution for Wind Only. With 1,224 sq ft of inlet openings distributed around the sidewalls, there would be about 410 sq ft in each long side and 202 sq ft in each end. The outlet area will be equally distributed on the two sides of the monitor, or 612 sq ft on each side. With the wind perpendicular to the long side, there will be 410 sq ft of opening in its path for inflow and 612 in the lee side of the monitor for outflow with the windward side closed. The air flow, as calculated by Equation 1, will be:

$$Q = 0.60 \times 410 \times 704 = 173{,}200 \text{ cfm.}$$

This gives 17.3 air changes per hour, which should be more than ample when there is no heat to be removed.

Solution for Combined Forces. Since the windward side of the monitor is closed when the wind is blowing, the flow due to temperature difference must be calculated for this condition, using Fig. 2. This chart shows that when inlets are twice the size of the outlets, in this case 1,224 sq ft in the sidewalls and 612 sq ft in the monitor, the flow will be increased 26.5 per cent over that produced by equal openings. Using the smaller opening and the flow per square foot obtained previously, the calculated amount for this condition will be:

$$612 \times 163 \times 1.265 = 126{,}200 \text{ cfm.}$$

Adding the two computed flows:

Temperature Difference	= 126,200 =	42 per cent.
Wind	= 173,200 =	58 per cent.
Total	299,400 =	100 per cent.

From Fig. 3, it is determined that when the flow, due to temperature difference, is 42 per cent of the total, the actual flow, due to the combined forces, will be about 1.6 times that calculated for temperature difference alone, or 201,920 cfm.

The original flow, due to temperature difference alone, was 199,286 cfm with all openings in use. The effect of the wind is to increase this to 201,920 cfm even though half of the outlets are closed.

A factor of judgment is necessary in the location of the openings in a building, especially those in the roof, where heat, smoke and fumes are to be removed. Usually windward monitor openings should be closed, but if the wind is low enough for the temperature head to overcome it, all windows may be opened.

TYPES OF OPENINGS

Types of openings may be classified as: (1) windows, doors, monitor openings and skylights, (2) roof ventilators, (3) stacks connecting to registers, and (4) specially designed inlet or outlet openings.

Windows, Doors and Skylights

Windows have the advantage of transmitting light, as well as providing ventilating area when open. Their movable parts are arranged to open in various ways; they may open by sliding either vertically or horizontally, by tilting on horizontal pivots at or near the center, or by swinging on pivots at the top, bottom or side. Regardless of their design, the air flow per square foot of opening will be the same under the same conditions. The type of pivoting should receive consideration from the

standpoint of weather protection, and certain types may be advantageous in controlling the distribution of incoming air. Deflectors are sometimes used for the same purpose, and these devices should be considered a part of the ventilation system.

Roof Ventilators

The function of a roof ventilator is to provide a storm and weather proof air outlet. These are actuated by the same forces of wind and temperature head, which create flow through other types of openings. The capacity of a ventilator depends upon four things: (1) its location on the roof, (2) the resistance it and the duct work offers to air flow, (3) the height of draft, and (4) the efficiency of the ventilator in utilizing the kinetic energy of the wind for inducing flow by centrifugal or ejector action.

For maximum flow induction, a ventilator should be located on that part of the roof where it will receive the full wind without interference. If ventilators are installed within the suction region created by the wind passing over the building, or in a light court, or on a low building between two high buildings, their performance will be the same there as for any other type of opening of the same area. Their normal ejector action, if any, will be of no value in such a location.

The base of the ventilator should be of a taper-cone design to produce the effect of a bell-mouth nozzle whose coefficient of flow is considerably higher than that of a square-entrance orifice. If a grille is provided at the base, additional resistance is introduced, and it should be increased in size accordingly.

Air inlet openings located at lower levels in the building should be at least equal to, and preferably larger than the combined throat areas of all roof ventilators. The air discharged by a roof ventilator depends on wind velocity and temperature difference, and, in general, their performance will be the same as any monitor opening located in the same place, but due to the four capacity factors already mentioned, no simple formula can be devised for expressing ventilator capacity.

Roof ventilators may be classified as stationary, pivoting or oscillating, and rotating. Generally, these have a round throat, but the continuous-ridge ventilator, or so-called heat valve, would fall in the stationary classification. When selecting roof ventilators, some attention should be given to ruggedness of construction, storm proofing features, dampers and damper operating mechanisms, possibility of noise, original cost and maintenance.

Natural ventilation units may be used to supplement power-driven supply fans, and under favorable weather conditions it may be possible to stop the power-driven units.

Controls

Gravity ventilators may have dampers controlled by (1) hand, (2) thermostat, and (3) wind velocity, in combination with a fan. The thermostat station may be located anywhere in the building, or it may be located within the ventilator itself. The purpose of wind velocity control is to obtain a definite volume of exhaust regardless of the natural forces, the fan motor being energized when the natural exhaust capacity falls below a certain minimum, and again shut off when the wind velocity rises to the point where this minimum volume can be supplied by natural forces.

Stacks

Stacks or vertical flues are really chimneys and utilize both the inductive effect of the wind and the force of temperature difference. Like the roof ventilator, the stack outlet should be located so that the wind may act upon it from any direction. With little or no wind, chimney effect depends on temperature difference to produce a removal of air from the rooms where the inlet openings are located.

GENERAL RULES

A few of the important requirements in addition to those already outlined are:

1. Inlet openings in the building should be well distributed, and should be located on the windward side near the bottom, while outlet openings are located on the leeward side near the top. Outside air will then be supplied to the zone to be ventilated.

2. Inlet openings should not be obstructed by buildings, trees, sign boards, etc., outside nor by partitions inside.

3. Greatest flow per square foot of total opening is obtained by using inlet and outlet openings of nearly equal areas.

4. In the design of window ventilated buildings, where the direction of the wind is quite constant and dependable, the orientation of the building together with amount and grouping of ventilation openings can be readily arranged to take full advantage of the force of the wind. Where the wind's direction is quite variable, the openings should be arranged in sidewalls and monitors so that, as far as possible, there will be approximately equal areas on all sides Thus, no matter what the wind's direction, there will always be some openings directly exposed to the pressure force and others to a suction force, and effective movement through the building will be assured.

5. Direct short circuits between openings on two sides at a high level may clear the air at that level without producing any appreciable ventilation at the level of occupancy.

6. In order that temperature difference may produce a motive force, there must be vertical distance between openings. That is, if there are a number of openings available in a building, but all are at the same level, there will be no motive head produced by temperature difference, no matter how great that difference might be.

7. In order that the force of temperature difference may operate to maximum advantage, the vertical distance between inlet and outlet openings should be as great as possible. Openings in the vicinity of the neutral zone are less effective for ventilation.

8. In the use of monitors, windows on the windward side should usually be kept closed, since, if they are open, the inflow tendency of the wind counteracts the outflow tendency of temperature difference. Openings on the leeward side of the monitor result in cooperation of wind and temperature difference.

9. In an industrial building where furnaces that give off heat and fumes are to be installed, it is better to locate them in the end of the building exposed to the prevailing wind. The strong suction effect of the wind at the roof near the windward end will then cooperate with temperature difference, to provide for the most active and satisfactory removal of the heat and gas laden air.

10. In case it is impossible to locate furnaces in the windward end, that part of the building in which they are to be located should be built higher than the rest, so that the wind, in splashing therefrom will create a suction. The additional height also increases the effect of temperature difference to cooperate with the wind.

11. The intensity of suction or the vacuum produced by the jump of the wind is greatest just back of the building face. The area of suction does not vary with the wind velocity, but the flow due to suction is directly proportional to wind velocity.

12. Openings much larger than the calculated areas are sometimes desirable, especially when changes in occupancy are possible, or to provide for extremely hot days. In the former case, free openings should be located at the level of occupancy for psychological reasons.

13. In single story industrial buildings, particularly those covering large areas, natural ventilation must be accomplished by taking air in and out of the roof openings. Openings in the pressure zones can be used for inflow and openings in the suction zone, or openings in zones of less pressure, can be used for outflow. The ventilation is accomplished by the manipulation of openings to get air flow through the zones to be ventilated

DAIRY BARN VENTILATION[3]

A successful barn ventilating system is one which continuously supplies the proper amount of air required by the stock, with proper distribution and without drafts, and one which removes the excessive heat, moisture, and odors, and maintains the air at a proper temperature, relative humidity, and degree of cleanliness.

Barn temperatures below freezing and above 80 F affect milk production. Milk producing stock should be kept in a barn temperature between 45 and 50 F. Dry stock, at reduced feeding, may be kept in a barn 5 to 10 deg higher. Calf barns are generally kept at 60 F, while hospital and maternity barns usually have a temperature of 60 F or somewhat higher.

The heat produced by a cow of an average weight of 1000 lb may be taken as 3000 Btu per hour. The average rate of moisture production by a cow giving 20 lb of milk per day is 15 lb of water per day, or 4375 grains per hour. To set a standard of permissible relative humidity for cow barns is difficult. For 45 F an average relative humidity of 80 per cent is satisfactory, with 85 per cent as a limit.

Where the barn volume is within the limit that can be heated by the stabled animals, the air supply need not be heated. The air should be supplied through or near the ceiling. It is better to have the exhaust openings near the floor as larger volumes of warm air are then held in the barn and there is better temperature control with less likelihood of sudden change in barn temperature.

If a cow weighs 1000 lb and produces 3000 Btu of heat per hour, and if a barn for the cow has 600 cu ft of air space with 130 sq ft of building exposure, one cow will require 2600 to 3550 cu ft per hour of ventilation, depending on the temperature zone in which the barn is located. The permissible heat losses through the structure, based on one cow and depending on the temperature zone, vary between 0.043 and 0.066 Btu per hour per cubic foot of barn space, and 0.197 to 0.305 Btu per hour per square foot of barn exposure.

GARAGE VENTILATION

On account of the hazards resulting from carbon monoxide and other physiologically harmful or combustible gases or vapors in garages, the importance of proper ventilation of these buildings cannot be overemphasized. During the warm months of the year, garages are usually ventilated adequately because the doors and windows are kept open. As cold weather sets in, more and more of the ventilation openings are closed and consequently on extremely cold days the carbon monoxide concentration runs high.

Many garages can be satisfactorily ventilated by natural means particularly during the mild weather when doors and windows can be kept open. However, the A.S.H.V.E. Code of Minimum Requirements for Heating and Ventilating Garages, adopted in 1935, states that natural ventilation may be employed for the ventilation of storage sections where it is practical to maintain open windows or other openings at all times. The code specifies that such openings shall be distributed as uniformly

[3]Dairy Barn Ventilation, by F L Fairbanks (A S H V E TRANSACTIONS Vol. 34, 1928, p 181). Cow Barn Ventilation, by Alfred J Offner (A.S H V E TRANSACTIONS, Vol 39, 1933, p 149). For additional information on this subject refer to *Technical Bulletin, U S Department of Agriculture* (1930) by M. A R. Kelley. Also see Air Conditioning of Farm Buildings, by F L Fairbanks (*Agricultural Engineering*, November, 1937, p 485).

as possible in at least two outside walls, and that the total area of such openings shall be equivalent to at least 5 per cent of the floor area. The code further states that where it is impractical to operate such a system of natural ventilation, a mechanical system shall be used which shall provide for either the supply of 1 cu ft of air per minute from out-of-doors for each square foot of floor area, or for removing the same amount and discharging it to the outside as a means of flushing the garage.[4]

Research

Research on garage ventilation, undertaken by the A.S.H.V.E. Committee on Research at Washington University, St. Louis, Mo., and at the University of Kansas, Lawrence, Kans., in cooperation with the A.S.H. V.E. Research Laboratory, and at the A.S.H.V.E. Research Laboratory, has resulted in authoritative papers on the subject.

Some of the conclusions from work at the Laboratory are listed in the following statements:

1. Upward ventilation results in a lower concentration of carbon monoxide at the breathing line and a lower temperature above the breathing line than does downward ventilation, for the same rate of carbon monoxide production, air change and the same temperature at the 30-in. level

2. A lower rate of air change and a smaller heating load are required with upward than with downward ventilation.

3. In the average case upward ventilation results in a lower concentration of carbon monoxide in the occupied portion of a garage than is had with complete mixing of the exhaust gases and the air supplied However, the variations in concentration from point to point, together with the possible failure of the advantages of upward ventilation to accrue, suggest the basing of garage ventilation on complete mixing and an air change sufficient to dilute the exhaust gases to the allowable concentration of carbon monoxide.

4. The rate of carbon monoxide production by an idling car is shown to vary from 25 to 50 cu ft per hour, with an average rate of 35 cu ft per hour

5. An air change of 350,000 cu ft per hour per idling car is required to keep the carbon monoxide concentration down to one part in 10,000 parts of air

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[4]Code of Minimum Requirements for Heating and Ventilating Garages (A S H V E TRANSACTIONS, Vol. 41, 1935, p. 30), (A S H V E. Reprint, January, 1935)

Airation Study of Garages by W. C. Randall and L W Leonhard (A.S H V E TRANSACTIONS, Vol. 36, 1930, p 233).

A.S H V.E. RESEARCH REPORT No. 874—Carbon Monoxide Concentration in Garages, by A S. Langsdorf and R. R. Tucker (A S H V E. TRANSACTIONS, Vol. 36, 1930, p 511)

A.S.H.V.E. RESEARCH REPORT No. 935—Carbon Monoxide Distribution in Relation to the Ventilation of an Underground Ramp Garage, by F. C. Houghten and Paul McDermott (A S H V E. TRANSACTIONS, Vol. 38, 1932, p. 439).

A.S.H.V.E. RESEARCH REPORT No. 934—Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A.S H V E TRANSACTIONS, Vol 38, 1932, p. 424).

A S.H.V.E. RESEARCH REPORT No. 967—Carbon Monoxide Distribution in Relation to the Heating and Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A.S.H V.E. TRANSACTIONS, Vol. 39, 1933, p. 395).

Carbon Monoxide Surveys of Two Garages, by A. H. Sluss, E K Campbell and Louis M. Farber (A.S.H V E. TRANSACTIONS, Vol. 40 1934, p. 263).

CHAPTER 43

District Heating

Steam Distribution Piping, Selection of Pipe Sizes, Provision for Expansion, Conduits for Piping, Pipe Tunnels, Inside Piping, Steam Requirements, Fluid Meters and Metering, Rates, Utilization, Automatic Temperature Control

THOSE phases of district heating which frequently fall within the province of the heating engineer are outlined here with data and information for solving incidental problems in connection with institutions and factories. Some data are included to cover the piping peculiar to heating systems which are to be supplied with purchased steam. A complete district heating installation should not be attempted without a thorough study of the entire problem by men competent and experienced in that industry.

STEAM DISTRIBUTION PIPING

The methods used in district heating work for the distribution of steam are applicable to any problem involving the supply of steam to a group of buildings. The first step is to establish the route of the pipes, and in this matter the local conditions so fully control the layout that little can be said regarding it.

Having established the route of the pipes, the next step is to calculate the pipe sizes. In district heating work it is common practice to design the piping system on the basis of pressure drop. The initial pressure and the minimum permissible terminal pressure are specified and the pipe sizes are so chosen that the required amount of steam, with suitable allowances for future increases, will be transmitted without exceeding this pressure drop. The steam velocity is therefore almost disregarded and may reach a very high figure. Velocities of 35,000 fpm are not considered high. By the use of this method the pipe sizes are kept to a minimum with consequent savings in investment.

The steam flowing through any section of the piping can be computed from a study of the requirements of the several buildings served. In general a condensation rate of 0.25 lb per hour per square foot of equivalent heating surface is a safe figure. This allows for line condensation which, however, is a small part of the total at times of maximum load. Miscellaneous steam requirements such as laundry, cooking, or process should be individually calculated.

The steam requirements for water heating should be taken into account, but in most types of buildings this load will be relatively small compared with the heating load and will seldom occur at the time of the heating peak. Unusual features such as large heaters for swimming pools should not be overlooked.

The pressure at which the steam is to be distributed will depend upon (1) boiler pressure, (2) whether exhaust or live steam, (3) pressure requirements of apparatus to be served. If steam has been passed through electrical generating units, the pressure will be considerably lower than if live steam, direct from the boilers, is used.

The advantages of low pressure distribution (2 to 30 psi) are (1) smaller heat loss per square foot of pipe surface, (2) less trouble with traps and valves, (3) simpler problems in pressure reduction at the buildings, and

(4) general reduction in maintenance costs. With distribution pressures not exceeding 40 psi there is little danger even if the full distribution pressure should build up in the radiators through the faulty operation of a reducing valve; but with pressures higher than 50 psi a second reducing valve or some form of emergency relief is usually desirable to prevent excessive pressures in the radiators.

The advantages of high pressure distribution are (1) smaller pipe sizes and (2) greater adaptability of the steam to various operations other than

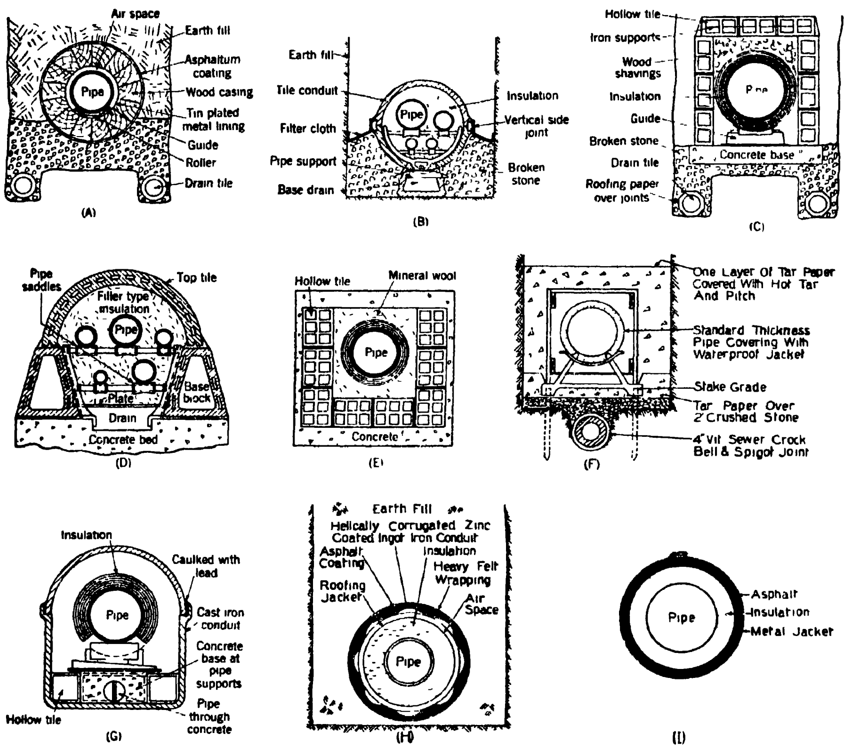


FIG. 1. CONSTRUCTION DETAILS OF CONDUITS COMMONLY USED

building heating, (3) wider flexibility as to allowance for maximum pressure drop.

The different kinds of apparatus which frequently must be served require various minimum pressures. Kitchen equipment requires from 5 to 15 psi, the higher pressures being necessary for apparatus in which water is boiled, such as stock kettles and coffee urns. An increased amount of heating surface, which is easily obtained in some kinds of apparatus, results in quicker and more satisfactory operation at low pressures. For laundry equipment, particularly the mangle, a pressure of 75 psi is usually demanded although 30 psi is sufficient if the flat work ironer is equipped with a large number of rolls and if a slower rate of operation is permissible. Pressing machines and hospital sterilizers require about 50 psi. Where pressures are not as high as desired higher pressures can be obtained by a steam compressor.

PIPE SIZES

The lengths of pipe, steam quantities, and initial and terminal pressures having been chosen, the pipe sizes can readily be calculated by means of Babcock's pressure drop formula:

$$P = 0.0000000367\left(1 + \frac{3.6}{D}\right)\frac{W^2L}{dD^5}$$

$$W = 5220\sqrt{\frac{PdD^5}{\left(1 + \frac{3.6}{D}\right)L}}$$

where

P = loss in pressure in pounds.
D = inside diameter of pipe in inches.
L = length of pipe in feet.
d = weight of 1 cu ft of steam.
W = pounds of steam per hour.

Numerical values of the various factors are given in Table 1, Chapter 14.

CONDUITS FOR PIPING

Conduits for steam pipes buried underground should be reasonably waterproof, able to withstand earth loads and to take care of the expansion and contraction of the piping without strain or stress on the couplings, or without affecting the insulation or conduit. Expansion of the piping must be carefully controlled by means of anchors and expansion joints or bends so that the pipes can never come in contact with the conduit. Anchors can be anchor fittings or U-shaped steel straps which partially encircle the pipes and are firmly bolted to a short length of structural or cast steel set in concrete. In general, cast steel is preferable to structural steel.

In laying out underground conduits the following points should be borne in mind:

1. The depth of the buried conduit should be kept at a minimum. Excavation costs are a large factor in the total cost.

2. An expansion joint, offset, or bend should be placed between each two anchors. Advantage should be taken of the flexibility of piping to absorb expansion wherever possible. Information on provisions for expansion will be found in Chapter 16.

3. A proper hydrostatic test should be made on the assembled line before the insulation and the top of the conduit are applied. The hydrostatic test pressure should be one and one-half times the maximum service pressure and it should be held for a period of at least two hours without evidence of leakage.

There are many types of conduits, some of which are manufactured products and some of which are built in the field. Some of the more common forms are illustrated in Fig. 1.

The conduit (A) is of a wood casing construction which has been widely used in the past. The wood casing is segmented, lined with tin, and bound with wire. The outside of the conduit is coated with asphaltum. It is not suitable for high temperatures or poorly drained soils.

In Fig. 1 (B), (C), (D), (H) and (I) are patented forms of conduits. The insulation is sometimes a loose filler packed into the conduit. Conduits (H) and (I) are prefabricated. Both of these conduits are enclosed in metal jackets.

At (C) and (E) are shown two tile conduits using sectional insulation. In these particular designs the space surrounding the pipe is filled par-

tially or wholly with a loose insulating material. The addition of this loose insulating material to the sectional insulation is, of course, optional and is justified only where high pressure steam is used.

(E) and (F) are conduits used by two district heating companies, and have the advantage of being constructed of common materials.

Conduit (G) is of cast-iron construction, assembled with lead joints and is water-tight, if properly laid. It is obviously expensive and is justified only in exceptional cases.

Since it is difficult to make a concrete or masonry conduit absolutely water-tight, provision should be made for some seepage. The pipe should

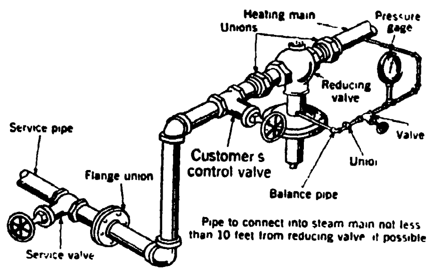


FIG. 2. CONNECTIONS FOR REDUCING VALVE WITHOUT BYPASS

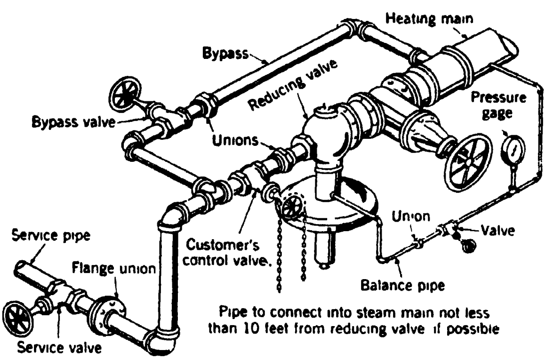


FIG. 3. CONNECTIONS FOR REDUCING VALVE WITH BYPASS

be protected by a waterproof jacket over the insulation and the seepage drained from the inside of the conduit.

Underdrainage of the conduit is generally provided for by a tile drain laid in crushed stone or gravel underneath the conduit. The tile underdrain should be carried to the sewer or some other drainage point.

Manholes are required at intervals for access to valves, traps, and some types of expansion joints.

Where steam and return piping are installed in the same conduit, the return piping usually follows the same grade as the steam piping. In general, the condensation is pumped back under pressure.

PIPE TUNNELS

Where steam heating lines are installed in tunnels large enough to provide walking space, the pipes are supported by means of hangers or

roller frames on brackets or frame racks at the side or sides of the tunnel. The pipes are insulated with sectional pipe insulation over which is placed a sewed-on, painted canvas jacket or a jacket of asphalt-saturated asbestos water-proofing felt. The tunnel itself is usually built of concrete or brick and water-proofed on the outside with membrane water-proofing.

Because of their relatively high first cost as compared with smaller conduits, walking tunnels are sometimes omitted along heating lines unless they are required to accommodate miscellaneous other services or provide underground passage between buildings.

OVERHEAD DISTRIBUTION

In some industrial and institutional applications, the distribution piping may be installed, entirely or in part, above ground. This method

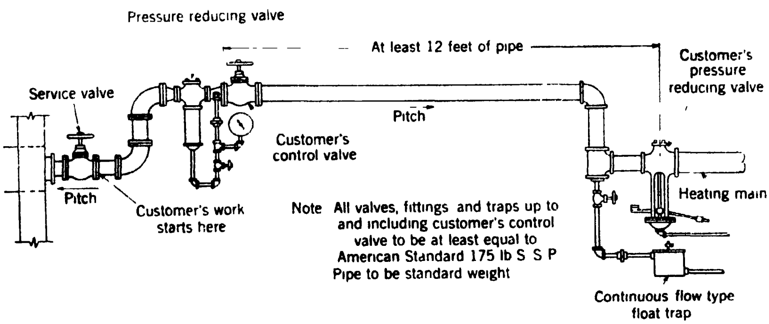


FIG. 4. STEAM SUPPLY CONNECTION WHEN USING TWO REDUCING VALVES

of construction has the advantage of requiring no excavation and being easily maintained.

INSIDE PIPING

Figs. 2 and 3 show typical service connections used for low pressure steam service.

Fig. 2 shows installation of a reducing valve without a bypass, which is usually omitted in the case of smaller size valves.

Fig. 3 illustrates the use of a reducing valve, with a bypass which is generally provided for larger installations. This latter construction permits the operation of the line in case of failure in the reducing valve. In the smaller sizes, the reducing valve can be removed, a filler installed, and the house valve used to throttle the flow of steam until repairs are made.

Fig. 4 shows a typical installation used for high pressure steam service[1]. The first reducing valve effects the initial pressure reduction. The second reducing valve reduces the steam pressure to that required.

Most district heating companies enforce certain regulations regarding the consumer's installation, partly to safeguard their own interests but principally to insure satisfactory and economical service to the consumer. There are certain fundamental principles that should be followed in the design of a building heating system which is to be supplied from street

[1]Code for Pressure Piping. B 31-1. 1942, *American Standards Association*, Paragraph 408, p 115.

mains. Although some of these apply to any building, they have been demonstrated to be especially important when steam is purchased.

1. Provision should be made for conveniently shutting off the steam supply at night and at other times when heat is not needed.

It has been thoroughly demonstrated that a considerable amount of heat can be saved by shutting off steam at night. Although there is, in some cases, an increased consumption of heat when steam is again turned on in the morning, there is a large net saving which may be explained by the fact that the lower inside temperature maintained during the night obviously results in lower heat loss from the building, and less heat need therefore be supplied.

Steam can be entirely shut off at night in most buildings even in very cold weather without endangering plumbing. It is necessary, however, to have an ample amount of heating surface so that the building can be quickly warmed in the morning. Where the hours of occupancy differ in various parts of the building, it is good practice to install

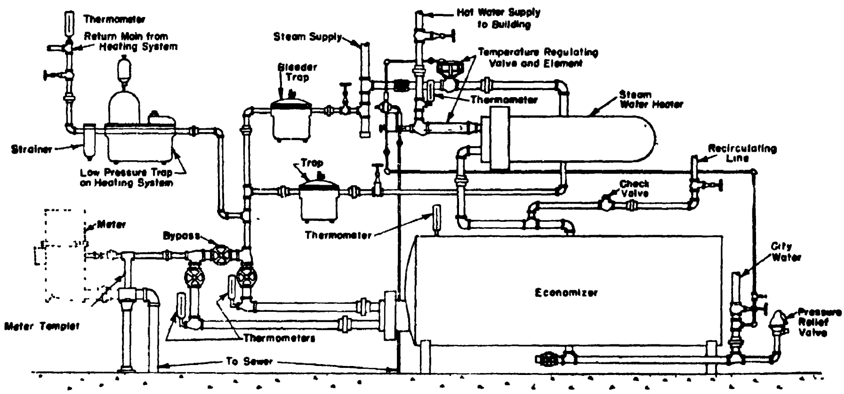


FIG. 5. METHOD OF INSTALLING A WATER HEATER AND ECONOMIZER IN A GRAVITY HEATING SYSTEM

separate supply pipes to the different parts. For example, in an office building with stores or restaurants on the first floor which are open in the evening, a separate main supplying the first floor will permit the steam to be shut off from the remainder of the building in the late afternoon. The division of the building into zones each with a separately controlled heat supply is sometimes desirable, as it permits the heat to be adjusted according to variations in sunshine and wind.

2. Residual heat in the condensate should be salvaged.

This heat may be salvaged by means of a cooling coil, or as is more frequently done, by a water heating economizer (see Fig. 5) which preheats the hot water supply to the building.

The condensation from the heating system, after leaving the trap, passes through the economizer. The supply to the hot water heater passes through the economizer, absorbing heat from the condensate. If the hot water system in the building is of the recirculating type, the recirculating connection should be tied in *between* the economizer and the water heater proper, not at the economizer inlet, because the recirculated hot water is itself at a high temperature.

Because of the lack of coincidence between the heating system load and the hot water demand, a greater amount of heat can be extracted from the condensate if storage capacity is provided for the preheated water. Frequently a type of economizer is used in which the coils are submerged in a storage tank.

3. Heat supply should be graduated according to variations in the outside temperature.

The maximum in economical operation and satisfactory heating can only be obtained by the use of some automatic temperature control system.

FLUID METERS

The perfection of fluid meters has contributed as much to the advancement of district heating as any other one thing. Meters are classified into two groups: *Quantity Meters* and *Rate of Flow Meters.*

Quantity Meters

The one type of quantity meter used is the condensation meter, which may be of the *tilting bucket* or *revolving drum* type.

The condensation meter is a popular type for use on small and medium sized installations, where all the condensate can be brought to a common point for metering purposes. Its simplicity of design, ease in testing, accuracy at all loads, low cost, and adaptability to low pressure distribution has made it standard equipment with many heating companies.

Condensation meters should not be operated under pressure; they are made for either gravity or vacuum installations. Where bucket traps

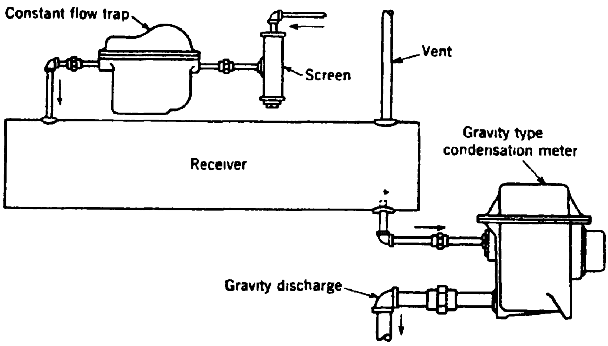


FIG 6 GRAVITY INSTALLATION FOR CONDENSATION METER USING VENTED RECEIVERS

are used, a vented receiver is essential ahead of the meter. Where continuous flow traps are used, a vented receiver is not necessary, but is desirable.

Fig. 6 illustrates a gravity condensation meter installation using a vented receiver.

Rate of Flow or Flow Meters

Flow meters used for district heating work are of three types: *Area Meters*, *Head Meters* and *Velocity Meters.*

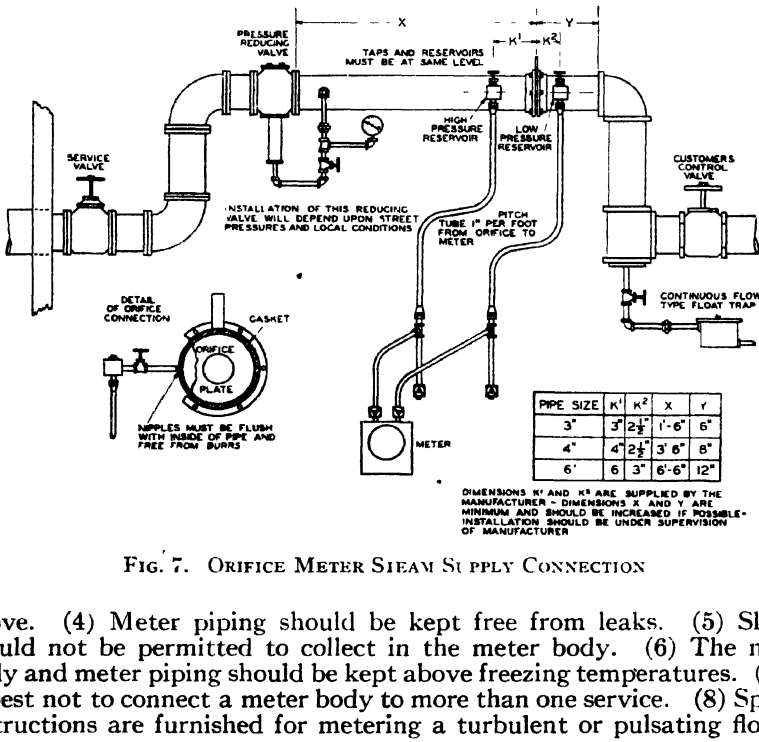
Area meters are those, in the operation of which, a variation in the cross-section of stream under constant head is used as an indication of the rate of flow. A tapered plug is suspended in an orifice and moves axially with the flow, which is vertically upward. The weight of the plug provides a definite pressure differential and the plug floats at such a height as will provide enough orifice area to pass the flow at the pressure difference. The position of the plug is transmitted by means of a lever and pencil and records the flow on a graduated strip chart.

Head meters are those in which the stream of fluid creates a difference of pressure, or differential head. This head is created by an orifice, Venturi tube, flow nozzle, or Pitot tube and will depend upon the velocity and density of the fluid.

The secondary element must contain a differential pressure gage, which

will translate the pressure difference into rate of flow or total flow. This mechanism may be either mechanical or electrical. The electric flow meter has the advantage of being able to locate the instruments as some distance from the primary element.

Fig. 7 is a typical example of an orifice-type meter installation. A few general points to be considered in installing a meter of this type are: (1) It is desirable to place the differential medium in a horizontal pipe in preference to a vertical one, where either location is available. (2) Reservoirs should always be on the same level and installed in accordance with the instructions of the meter company. (3) The meter body should be placed at a lower level than that of the pressure differential medium. Special instructions are furnished where the meter body is



PIPE SIZE	K^1	K^2	X	Y
3"	3"	2½"	1'-6"	6"
4"	4"	2½"	3' 6"	8"
6'	6	3"	6'-6"	12"

FIG. 7. ORIFICE METER STEAM SUPPLY CONNECTION

above. (4) Meter piping should be kept free from leaks. (5) Sludge should not be permitted to collect in the meter body. (6) The meter body and meter piping should be kept above freezing temperatures. (7) It is best not to connect a meter body to more than one service. (8) Special instructions are furnished for metering a turbulent or pulsating flow.

Velocity meters are those in which the primary element is some device that is kept in continual rotation by the linear motion of the stream. The secondary element is, essentially, a revolution counter. The primary and secondary elements are combined into one unit.

For steam metering, the shunt meter is an example of the velocity type. This unit is connected directly in 2, 3 and 4 in. pipe lines. Larger size mains are metered by installing a 2 in. meter in a bypass with a restricting orifice in the main line.

Selection of Meter

In selecting a meter for a particular installation, the number of different makes and types of meters suitable for the job is usually limited by one or more of the following considerations: (1) Its use in a new or an old

installation. (2) Method to be used in charging for the service. (3) Location of the meter. (4) Large or small quantity to be measured. (5) Temporary or permanent installation. (6) Cleanliness of the fluid to be measured. (7) Temperature of the fluid to be measured. (8) Accuracy expected. (9) Nature of flow: turbulent, pulsating, or steady. (10) Cost. *a.* purchase price, *b.* installation cost, *c.* calibration cost, *d.* maintenance cost. (11) Servicing facilities of the manufacturer. (12) Pressure at which fluid is to be metered. (13) Type of record desired as to indicating, recording or totalizing. (14) Stocking of repair parts. (15) Use of open jets where steam is to be metered. (16) Metering to be done by one meter or by a combination of meters. (17) Use as a check meter. (18) Its facilities for determining or recording information other than flow.

STEAM REQUIREMENTS

Methods of estimating steam requirements for heating various types of buildings are given in Chapter 11.

Table 7 in Chapter 11 represents information obtained from all sections of the United States, and the group of buildings from which the information was taken represents a cross section of all types of heating systems.

Steam requirements for water heating can be satisfactorily estimated by using a consumption of 0.0025 lb per day per cubic foot of heated space for office buildings, without restaurants, and 0.0065 lb per day per cubic foot of heated space for apartment buildings.

Complete information on water heating requirements is given in Chapter 46.

Additional data on steam requirements of various types of buildings in a number of cities may be found in the Handbook of the *National District Heating Association.*

RATES

Fundamentally, district heating rates are based upon the same principles as those recognized in the electric light and power industry, the main object being a reasonable return on the investment. However, there are other requirements to be met; the rate for each class of service should be based upon the cost to the utility company of the service supplied and upon the value of the service to the consumer, and it must be between these two limits. District heating rates should be designed to produce a sufficient return on the investment regardless of weather conditions, although existing rate schedules do not conform with this principle. Lastly, the rate schedule must be reasonably simple and understandable.

Glossary of Rate Terms

Load Factor. The ratio, in per cent, of the average hourly load to the maximum hourly load. This is usually based on a one year period but may be applied to any specified period.

Demand Factor. The relation between the connected radiator surface or required radiator surface and the demand of the particular installation. It varies from 0.25 to 0.3 lb per hour per square foot of surface.

Diversity Factor. The ratio of the sum of the individual demands of a number of buildings to the actual composite demand of the group.

Types of Rates

1. Flat Rates.
   a. Radiator surface charge. *Obsolescent.*

2. Meter Rates.
   a. Straight-line.
   b. Step. *Obsolescent.*
   c. Block.
      (a) Class rates.

Straight-Line Meter Rate. The price charged per unit is constant, and the consumer pays in direct proportion to his consumption without regard to the difference in costs of supplying the individual customers.

Block Meter Rate. The pounds of steam consumed by a customer are divided into blocks of thousands of pounds each, and lower rates are charged for each successive block consumed. This type of charge predominates in steam heating rate schedules for it has the advantage of proportioning the bill according to the consumption and the cost of service. It has the disadvantage of not discriminating between customers having a high load factor (relatively low demand) and those having a low load factor (relatively high demand). The utility company must maintain sufficient capacity to serve the high demand customers and the cost of the increased plant investment is divided equally among the users, so the high demand customers are benefited at the expense of the others.

3. Demand Rates.
   a. Flat demand.
   b. Wright.
   c. Hopkinson.
   d. Doherty (or Three charge).

Demand Rates These refer to any method of charge based on a measured maximum load during a specified period of time.

The *flat demand rate* is usually expressed in dollars per thousand pounds of demand per month or per annum. It is based on the size of a customer's installation, and is seldom used except where a meter is not practicable.

The *Wright demand rate* is similar in calculation to the block rate except that it is expressed in terms of hours' use of the maximum demand. It is seldom used but forms the basis for other forms of rates.

The *Hopkinson demand rate* is divided into two elements:

(a) A charge based upon the demand, either estimated or measured.

(b) A charge based upon the amount of steam consumed.

This rate may be modified by dividing the quantities of steam demanded and consumed into blocks charged for at different rates

The *Doherty rate* is divided into three elements.

(a) A charge based upon demand.

(b) A charge based upon steam consumed.

(c) A customer charge

In the *Hopkinson rate*, the last two elements are combined into one element.

Demand rates are comparatively new and are not yet widely used; though they are equitable and competitive they are difficult for the average layman to understand. They are of benefit to utility companies and to consumers because the investment and operating costs can be divided to suit the particular circumstances into demand, customer, and consumption groups through the use of some modification of the Hopkinson rate. Demand rates are an advantage to the customer in that the use of such a rate reduces the rate per thousand pounds to the long-hour user.

Fuel Price Surcharge. It is usually desirable to establish a rate upon a specified basic cost of fuel to the utility company. Where there are wide variations in the price of fuel, it is also desirable to add a definite charge per thousand pounds of steam sold for each increment of increase in the price of fuel. This surcharge automatically compensates for the

variations without necessitating frequent changing of the whole rate structure.

Some utility companies include a labor surcharge as well as a coal surcharge.

UTILIZATION

Considerable savings can be made by the proper and intelligent operation of heating systems. It should be borne in mind that a heating system is designed to heat a building to 70 F inside when the outside temperature is at its lowest point for that particular locality. There is a tendency to overheat the building at any time the outside temperature is above the design temperature unless some method of regulation is used, either automatic or manual.

The general rules for economical operation[2] are as follows:

1. *Reduce the heat losses from the building to a minimum.*
   a. Weatherstrip all windows, and caulk all window frames.
   b. Provide revolving or vestibule doors on all entrances. Separate shipping and receiving rooms from the remainder of the building by partitions so that the large doors will not ventilate the entire building.
   c. Eliminate all unnecessary ventilation. Ventilating equipment is usually sized to meet extreme requirements. In a theater or auditorium, do not supply enough ventilation for an audience of 2000 when there are only 200 present.

2. *Limit the hours of heating to those in which the required temperature is necessary.*
   a. Determine the hours that heating is required and see that steam is shut off for the maximum time when not required, such as nights, Sundays, and holidays.
   b. Shut steam off entirely in unoccupied sections of the building, taking care to avoid freezing plumbing.
   c. Install separate lines for those parts of the building that require long-hour or 24-hour heating. This is much cheaper than heating the entire building
   d. Control the heat supplied to water storage tanks located on or above the roof. Such tanks require heat to prevent freezing when the outdoor temperature is below 32 F.

3. *Regulate the amount of heat so as to prevent overheating and to maintain uniform temperatures during the hours of occupancy.*
   a. Determine the temperature required for the occupancy of a building Do not heat a storage garage or a furniture warehouse to the temperature required in a hospital ward.
   b. Shut off steam during the day whenever possible. An automatic control will do this, but it can be done by hand, with good results
   c. Provide some good means of temperature control.

4. *See that the heat input is properly balanced throughout the building.*
   a. See that the entire heating system responds rapidly when steam is turned on. Locate and eliminate the cause of any sluggish circulation. Balance the radiation, provide adequate air elimination, and correct any trapped run-outs to provide quick system drainage.
   b. Place the radiation near the outside walls under the windows or where the exposure occurs, if possible.
   c. Do not obstruct radiators or prevent the free circulation of air around them; to do so seriously reduces the heating capacity of a radiator.

5. *Keep all heating equipment in first class condition*
   a. Keep the system in good repair. This applies to all traps, valves, vents, steam and return piping, vacuum pumps, and temperature control apparatus.
   b. In a vacuum system, maintain the degree of vacuum recommended by the control manufacturer. If this is not possible, locate and eliminate all leaks.
   c. Insulate all steam pipes not used as heating surface.

[2]Principles of Economical Heating, *National Association of Building Owners and Managers*

6. *Arrange the heating system to obtain from it the highest possible efficiency.*
   a. Locate all valves and controls so as to be convenient and accessible. It is only human nature to delay or avoid doing that which is unnecessarily inconvenient.
   b. Investigate every complaint of "No Heat;" find the cause and correct it. Do not overheat an entire building to correct a local condition.
   c. Extract the heat in the condensate for heating water or for some other useful purpose.

7. *Make a study of the heating system and heating requirements.*
   a. Provide thermometers and recording pressure gages so that the heating system may be operated with full knowledge of what is being accomplished.
   b. Keep daily consumption records and check against the theoretical requirements.
   c. Study the system and understand its functions and its operations.

AUTOMATIC TEMPERATURE CONTROL

As stated in Chapter 33, Automatic Control, properly applied to heating, ventilating and air conditioning systems, makes possible the maintenance of desired conditions with maximum operating economy. The use of adequate temperature control provides more healthful, comfortable, and efficient working conditions in buildings.

There are three general means of obtaining centralized control of heat output of radiators.

1. *Controlling the rate of steam flow* into the radiators. This is accomplished by equipping the radiator inlets with orifices and controlling the flow of steam through them into the radiator by controlling the difference in absolute pressure between the supply and return.

2. *Controlling the temperature of steam* in the radiators by varying its pressure. This involves the use of high vacuums to obtain low steam temperatures. This must be supplemented by some other type of control for low heat output.

3. *Controlling the length of time steam flows* into the radiators by admitting steam to a heating system intermittently and varying the length of the *on* and *off* periods. Two types of controls are used. (1) A clock control providing *on* and *off* settings of various lengths, which can be changed in accordance with outside temperatures. In most cases these changes are made automatically by means of a thermostatic bulb, placed outdoors. (2) A control, having an outdoor bulb and a bulb attached to the radiator, which varies the length and frequency of the *on* intervals in such a way that the radiator temperature is varied according to the outside temperature. In some cases heat supply is controlled by combinations of the three methods described

Before installing any type of modern temperature control equipment, it is necessary to see that the heating system is put in good operating condition. In general, the heating system in a building is not given the attention that other mechanical equipment is given because it will continue to function, after a fashion, even though changes in piping, location of radiation, settlement of piping, and the normal wear and tear or other changes have taken place. Because of this depreciation of the system, operation becomes more and more costly and parts of the building have to be greatly overheated in order to prevent underheating in other parts. Vents, traps, vacuum pumps, and valves should be given a careful inspection and replaced or repaired if required. The piping should be of adequate size and graded properly. The return piping should be inspected, and any pockets or lifts removed and properly vented. These inspections and repairs are not costly and may prevent a much greater outlay in future years. In most cities district heating companies will be willing to make a survey of heating systems and offer recommendations in regard to operation and changes in piping layout.

The selection of control equipment depends upon the type and size of building and the degree of saving which may be obtainable.

CHAPTER 44

Electric Heating

Resistors, Heating Elements, Electric Heaters, Unit Heaters, Central Fan Heating, Electric Boilers, Electric Hot Water Heating, Heating Domestic Water Supply, Radiant Drying, Reversed Cycle Refrigeration, Auxiliary Electric Heating, Induction and Electrostatic Heating, Control, Calculating Capacities, Power Problems

ELECTRIC heating is steadily assuming a more important place in heating, ventilating and air conditioning installations because it is flexible, clean, safe, convenient and easy to control. It has many basic principles in common with fuel heating, but there are also important differences. When heat is delivered by wire, no combustion process is necessary, either at a central plant or at the individual room units. The output of an electric heater is a fixed constant, unaffected by the temperature of the surrounding air and it follows that the total load on an electric heating system is the total wattage of connected electric heaters, regardless of weather conditions. The main obstacle to the more general adoption of electric heating for buildings is the cost of the electricity itself.

All heat is a form of energy. Fuels hold stored chemical energy which is released into heat by combustion. Electrical power is a form of energy which can be released into heat by passing it through a resisting material. Both fuel and electric heating have two divisions: *first*, the conversion of energy into heat; *second*, the distribution and practical use of the heat after it is produced.

In converting the chemical energy of fuels into heat by combustion, there is necessarily a considerable variation in thermal efficiency. This is not true, however, when converting electric power into heat, as 100 per cent of the energy applied to the resistor is always transformed into heat. In electric heating practice no concern need be given to efficiencies of heat production, but rather to efficiencies of heat utilization. The problem is to distribute the electrically produced heat units in such manner as to obtain conditions of maximum comfort with the minimum consumption of electricity.

DEFINITIONS

Definitions of general terms used in fuel heating are given in Chapter 47. Terms which apply particularly to electric heating are as follows:

Electric Resistor: A material used to produce heat by passing an electric current through it.

Electric Heating Element: A unit assembly consisting of a resistor, insulated supports, and terminals for connecting the resistor to electric power.

Electric Heater: A complete assembly of heating elements with their enclosure, ready for installation in service.

RESISTORS AND HEATING ELEMENTS

Solids, liquids, and gases may be used as resistors, but most commercial electric heating elements have solid resistors, such as metal alloys, and non-metallic compounds containing carbon. In some types of electric boilers, water forms the resistor which is heated by an alternating current of electricity passing through it. One of the more common

resistors is nickel-chromium wire or ribbon which, in order to avoid oxidation, contains practically no iron.

Commercial electric heating elements are made in many types. Some have resistors exposed to the air being heated. The resistors may be coils of wire or metal ribbon, supported by refractory insulation, or they may be non-metallic rods, mounted on insulators. This type of element is used extensively for operation at high temperatures when radiant heat is desired, also at low temperatures for convection and fan circulation heating, especially in large installations.

Some elements have metallic resistors embedded in a refractory insulating material, encased in a protective sheath of metal. Fins or extended surfaces may be used to add heat-dissipating area. Elements are made in many forms, such as strips, rings, plates and tubes. Strip elements are used for clamping to surfaces requiring heat by conduction, and in some types of convection air heaters. Ring and plate elements are used in electric ranges, waffle irons, and in many small air heaters. Tubular elements may be immersed in liquids, cast into metal, and, when formed into coils, used in electric ranges and air heaters. Cloth fabrics woven from flexible resistor wires and asbestos thread are used for many low temperature purposes such as heating pads and aviators' clothing.

Special incandescent lamps are used as heating elements in certain applications where radiant heat is desired. These use carbon or tungsten filaments as resistors, and are designed to produce maximum energy in the infra-red portion of the spectrum.

ELECTRIC HEATERS

Electric heaters may be divided into four groups: conduction, radiant, convection, and induction.

Conduction electric heaters, which deliver most of their heat by actual contact with the object to be heated, are used in such applications as aviators' clothing, hot pads, foot warmers, soil heaters, ice melters, and water heaters. Conduction heaters are useful in conserving and localizing heat delivery at definite points. They are not suitable for general air heating.

Radiant electric heaters, which deliver most of their heat by radiation, have high temperature heating elements and reflectors to concentrate the heat rays in the desired directions. The immediate and pleasant sensation of warmth which is caused by radiant heat makes this type desirable for temporary use where the heat rays can fall directly upon the body. They are not satisfactory for general air heating, as radiant heat rays do not warm the air through which they pass. They must first be absorbed by walls, furniture, or other solid objects which then give up the heat to the air. For a discussion of electrically heated panels as applied to radiant heating, see Chapter 45.

Gravity convection electric heaters, designed to induce thermal air circulation, deliver heat largely by convection, and should be located and used in much the same manner as steam and hot water radiators or convectors. They generally have heating elements of large area, with moderate surface temperature, enclosed to give proper stack effect to draw cold air from the floor line. The flexibility possible with electric heating elements should discourage the use of secondary mediums for heat transfer. Water and steam add nothing to the efficiency of an electric heater and entail expensive construction and maintenance.

Induction Heaters are described in the section on Electronic Heating By Induction and Electronic Means.

UNIT HEATERS

Electric unit heaters include a built-in fan unit which circulates room air over the heating elements. Heaters of this type are manufactured in many designs and sizes, and can be located in the same manner as steam unit heaters.

Electric unit heaters are used in industrial plants, sub-stations, power houses, pumping stations, etc., where the power rate for electric heating is found to be favorable. In many large plants, such as flour mills, grain elevators, etc., in which there are a number of small offices, locker rooms, etc., scattered over wide areas, electric unit heaters are frequently economical in such locations. In small unattended stations, where freezing temperatures cannot be permitted, thermostatically-controlled electric unit heaters are frequently used to maintain a temperature above freezing. The best location for the heaters depends upon local circumstances as they can be mounted on the ceiling to direct the air downward,

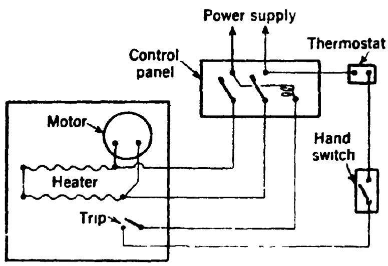


FIG. 1. WIRING DIAGRAM FOR UNIT HEATER

on the side wall about 7 ft from the floor, or near the floor line. Variations in design are necessary for different locations, as with steam unit heaters, see Chapter 21.

The arrangement of the wiring circuits is very important for electric unit heaters. In principle they are all the same and include as essential elements an automatic control panel, a thermostat, and a master hand switch. All heaters should be designed with a safety thermal trip wired in series with the magnet coil of the control panel and with the hand switch and thermostat. A typical wiring diagram is shown in Fig. 1. This applies to a single phase power supply, but for 3-phase the only difference is to have a 3-pole panel and a heater arrangement for 3-phase connection.

Portable unit heaters are useful for temporary work, such as drying out damp rooms, or for warming rooms during construction.

CENTRAL FAN HEATING

Electric heating elements can be used for the prime source of heat in a central fan electric heating system or in the heating phase of a complete air conditioning system. They can be used in the same manner as steam heating units for tempering, preheating or reheating the air at the main supply fan location and as booster heaters at the delivery terminals of the duct system. In the humidification phase of air conditioning electric heating elements can be used to provide moisture by the evaporation

of water, or for controlling air washer dew-point temperatures when mounted as preheating units on the intake side of the air washer. (See Chapter 20.)

In coordinating the input of heat energy and the volume of air circulation, a basic difference between electric heating and steam heating enters into the problem. Steam is approximately a constant-temperature source of heat for any given pressure and a change in air volume flowing over steam coils does not greatly affect the temperatues of the delivered air. The amount of steam condensed (heat input) varies in proportion to the air volume, but the surface temperature of the steam coils remains about the same. Electric heat is quite different, having a constant input of energy. If the volume of air flow over electric heating elements is changed, and no change is made in the electrical power connections, there will be a corresponding change in the temperature of the air delivered. This occurs because the electrical energy input remains constant and the surface temperature of the heating elements will vary as is necessary to force the air to accept all the heat. With electric heat the total heat is constant unless some compensating action is performed by control. Automatic variation of the electrical heat input synchronized properly with the air flow can be successfully accomplished by various special methods of control. By-pass dampers as used in steam units will not control electric heat.

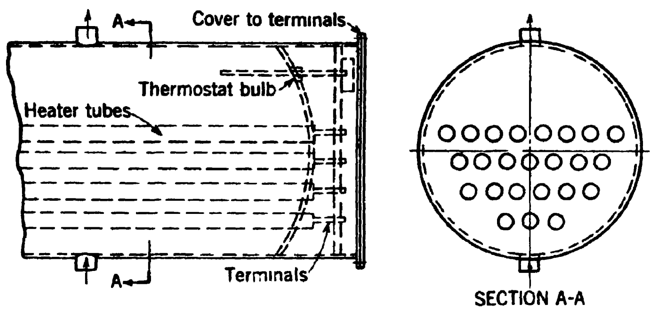


FIG. 2. RESISTANCE TYPE BOILER FOR STEAM OR HOT WATER

Electric heaters are useful in balancing the heat distribution in central fan heating systems. Even in those instances where steam is the principal heat source, the temperature of individual rooms can be controlled locally by separate electric booster heaters. These heaters can be installed in branch ducts or behind the air outlet grilles in each room. With this arrangement, the central heating unit distributes air at an average temperature, controlled from a thermostat centrally located, such as in the main return duct. The electric booster heaters may be controlled by thermostats mounted in each individual room which permit the occupant to maintain any desired temperature independent of the rest of the building.

ELECTRIC BOILERS

Steam or hot water generating boilers using electric energy are entirely automatic and are well adapted to intermittent operation. Small electric boilers usually have heating elements of the enclosed metal resistor type immersed in the water. Boilers of this construction may be used either with direct or alternating current since the heat is delivered to the water

by contact with the hot surfaces. To lessen the likelihood of the heating elements burning out, they should be of substantial construction, with a low heat density per unit of surface area and provision should be made for cleaning off deposits of scale which restrict the heat flow. A typical resistance type of steam or hot water boiler is shown in Fig. 2.

Large electric boilers are usually of the type employing water as the resistor, using immersed electrodes. With this type only alternating current can be used, as direct current would cause electrolytic deterioration. Such a type of electrode boiler is shown in Fig. 3.

Electric steam boilers are useful in industrial plants which require limited amounts of steam for local processes, and also for sterilizers, jacketed vessels and pressing machines which need a ready supply of steam. It sometimes is economical to shut down the main plant fuel

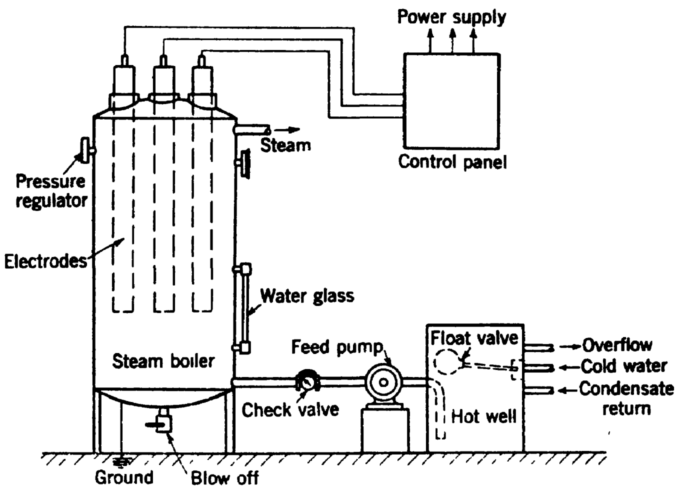


FIG. 3. DIAGRAMMATIC ARRANGEMENT OF AN ELECTRODE BOILER

burning boilers when the heating season ends, and to supply steam for summer needs with small electric steam boilers located close to the operation. In general, electric steam heating is confined to auxiliary or other limited applications. If the heating system is designed to use electricity exclusively, steam generating or distributing equipment is superfluous.

ELECTRIC HOT WATER HEATING

Electric water heating, using an electric boiler in place of a fuel burning boiler, like electric steam heating, is generally confined to auxiliary or other limited applications. The use of insulated water storage tanks, in which to store heat generated by electricity during off-peak hours at extremely low rates, is a development which has some special applications.

In this system of heating, the primary storage tank is simply a large, well-insulated, pressure type steel tank, equipped with electric heating elements and automatic time switches, which also have automatic limit controls for temperature and pressure. The heating system installed in the building may be of any standard individual radiator or fan-served indirect type or with provisions for the heating and humidification phases

of an air conditioning system. A system of this kind requires very careful design to avoid excessive over-all radiation losses during periods of low heat demand. It is also important to provide for sudden changes in heat demand. A typical hot water heating boiler is illustrated in Fig. 2.

HEATING DOMESTIC WATER BY ELECTRICITY [1]

Electric water heaters of the automatic storage type for domestic hot water supply are simple and reliable. In many sections of the country low electric rates have been established by the electric utilities to secure this load. In many localities, electric rate schedules divide the current used for water heating into two classifications, regular and off-peak. A time switch automatically limits use of the off-peak heating element to the hours of off-peak load, while the regular heating element is a stand-by at all times. Storage of this two-element type of water heater is larger than average to carry over the periods when the off-peak element is timed

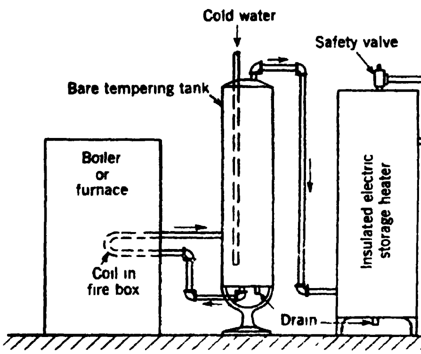


Fig. 4. Piping Arrangement for Connecting Electric Water Heater to Fire-Box Coil

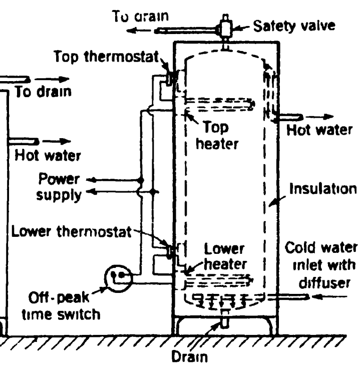


Fig. 5. Domestic Hot Water Heater for Off-Peak Service

out, without too frequent demands on the regular heating element which takes the higher domestic lighting service rate. Some utilities now offer a schedule which, beyond a stipulated minimum, lowers the rate for all electric service if an electric water heater is installed.

Competition with other fuels, especially gas, seems to be the major controlling factor in the use of electricity. The first cost of electric storage heaters is greater than for gas, owing to the need for larger tank storage due to off-peak service and slower recuperating capacity.

In residential work, to effect a saving in the cost of operation, it is sometimes desirable to use a furnace coil or indirect heater in connection with an electric water heater. In this case it is important to make the proper connections in order to benefit by any heat obtained from the furnace and at the same time to prevent dangerous overheating. The proper piping connections are shown in Fig. 4, and in this case the electric heater will only furnish heat when insufficient heat is supplied from the furnace. This arrangement has a further advantage in the summertime in that the bare tank through which the cold water passes on its way to the electric heater serves as a tempering tank, absorbing heat from the

[1]Application of Electric Water Heaters To Domestic Service, by C. G. Hillier (A S H V E Journal Section, *Heating, Piping and Air Conditioning*, November, 1936, p. 632) Fourteenth Range and Water Heater Survey (*Electric Light and Power*, August, 1940).

basement air and requiring the use of less energy in the electric heater.

A typical domestic hot water heater as shown in Fig. 5 is arranged with upper and lower heating elements for the usual type of off-peak heating service. The lower heating element is under the control of the off-peak time switch. However, the upper heating element is usually connected to the line so that, in case the supply of hot water in the tank becomes exhausted, the top thermostat can turn on the top heater and heat a small supply of water. The top heater will not heat the water in the tank below its location, but when the off-peak period arrives the lower heater is turned on and the entire tank becomes heated.

HEATING BY REVERSED CYCLE REFRIGERATION

Reversed refrigeration is frequently referred to as a *heat pump* since the electric motor driving the refrigerating compressor furnishes the motive power to transfer heat from one temperature to a higher temperature level. The compressor acts as a reversible refrigerating unit to extract heat from the outdoor air in winter and deliver it indoors for heating purposes, and, by a reversal, to extract heat from the indoor air in summer and discharge it outdoors.

In normal use a refrigerating machine is arranged to remove heat and the heat removed is dissipated to the condenser cooling water. The driving energy is converted into heat, most of which is added to the heat removed and extracted. In so-called reversed refrigeration the heat removed together with the heat converted from the driving energy is utilized to heat the building. This conservation of the heat converted from the driving energy enables the reversed refrigeration to show a better performance in heating service than straight refrigeration can show in cooling service. In order to overcome the drop in capacity and in efficiency with lower outside temperatures, it is often desirable to use well-water instead of air as the source of heat. For a detailed description of this cycle see Chapter 24.

AUXILIARY ELECTRIC HEATING

In conjunction with heating systems of other types, an auxiliary electric heating arrangement is a convenient means of caring for mild days in the spring and fall which require little heat to make a building comfortable. Likewise, such electric heating might be used on abnormally cold days to help out the main heating system and by this means reduce the necessary size of the system.

A few installations have been made using electric heating cable buried in the floors of bathrooms, etc., to provide auxiliary electric heating. At least one airplane hangar is heated in this manner.

Because of the feeling of comfort that a radiant type heater gives, bathrooms may be heated electrically with this type of heater while the rest of the house is cared for by some other system. Offices and rooms which require heat at periods when the main heating plant is shut down can be conveniently heated electrically.

CONTROL

Because the efficiency of electric heat production is the same for small and large units, it is possible to reduce heat waste to a minimum by applying local heating, locally controlled. Heaters are often controlled manually but thermostatic control is essential for economical operation. For duct systems having a variable volume of air flow the electric heater

control must automatically vary the heat input in coordination with the changes in air volume and demand for heat.

CALCULATING CAPACITIES

In calculating electric heating capacity one kilowatt is equal to 3413 Btu per hour or 14.2 sq ft equivalent direct steam radiation.

INDUSTRIAL USES FOR ELECTRIC HEAT

Electric heating is valuable for many industrial processes in both low and high temperature ranges. It is easily controlled and can be justified where savings in labor or improved quality of product outweigh the inherent higher cost per unit of heat as compared with other fuels. Important examples are metal melting and heat treating furnaces, ovens, dryers, laboratory equipment, cooking vessels, oil preheaters, catalysts, and countless other special uses.

RADIANT DRYING

Lacquers and similar surface films can be very effectively dried by radiation. Special electric lamp bulbs have been developed which give off a high percentage of infra-red and similar heat rays[2]. These are mounted in very efficient reflectors. For continuous manufacturing processes these reflectors are mounted in tunnels through which conveyors pass. For local applications, as for example paint drying in automobile repair shops, they may be mounted on portable racks.

In the application of this type of drying the composition of the paint or lacquer is important. In general, lacquers and those enamels using synthetic resins react most favorably. Other applications include the drying of ink, glue, and water, the softening of celluloid and bakelite for punching or shearing, and a wide variety of other uses[3].

Objects of relatively large surface area in proportion to their weight, and fabricated materials having a rather high heat absorption, may be satisfactorily heated by such a source.

ELECTRONIC HEATING BY INDUCTION AND ELECTROSTATIC MEANS

These methods differ radically from resistance heating as they employ high frequency radio waves to apply the energy which produces heat. High frequency heating has many important industrial uses and opens up a whole new field of special applications where extreme accuracy and speed are vital. Some spectacular results are being attained with this modern industrial tool. Skillful engineering design and experience are necessary to produce safe and satisfactory performance, but this technical assistance is now available from many sources.

Metals can be heated by induction. When the work is placed in a magnetic field within a high frequency coil, eddy currents immediately produce heat in the body of the metallic piece. The speed and intensity of this heating can be regulated by controlling the high frequency currents producing the magnetic field and the location of the spot heated by the position of the work piece within the coil. Induction heating is

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[2]Infra-Red Lamps Speed Up Drying Operations (*Automotive Industries* 82:376-7; April 15, 1940). Invisible Rays Build Visible Profits, by H. M. Archer (*Electric Light and Power*, May, 1940). Radiant Energy Drying and Baking for Organic Finishing (*Metal Industry* 38:294-6; May, 1940).

[3]Infra-Red Heating, Section IV, Power Sales Manual (*Edison Electric Institute*).

very useful in special processes such as melting metals, brazing, forging, heat treating, etc. It is possible to apply localized heat so rapidly that conduction cannot draw the heat away before it has time to accomplish the desired purpose at a particular spot. One example is the rapid hardening of a tool edge or tip too quickly for scale to form.

Dielectric materials can be heated internally by introducing them into an electrostatic field between high frequency electrode plates. Foods can be sterilized, plywoods bonded, plastics heated, granular or crystalline materials dehydrated, and countless other products heated quickly and uniformly, although they are poor thermal conductors and resist heat applied to their exteriors. Electrostatic heating is ideally suited for continuous production processes as the materials can pass through the heating field quickly with very short exposures to the high frequency radio waves.

POWER PROBLEMS

The cost of electric energy varies because of several factors. Distribution costs differ for large and small users. The fact that electricity cannot be economically stored, but must be used as fast as it is generated, makes it impossible to operate electric plants at uniform loads; hence, even the time of use may affect the cost of electricity. Special low rates are sometimes available during certain prescribed hours of use.

Since the cost of production and distribution depends not only upon the quantity of energy used but also upon the maximum rate at which it is used, electric energy is often sold on a demand rate basis. In some cases, the demand charge is based upon the rated connected load, in other cases, upon the maximum demand as indicated by a demand meter.

Homes are almost universally supplied with lighting current of 115 volts, which can only be used economically for small heaters. Usually the service lines will not permit more than plug-in devices. The Underwriters permit approved heaters of 1320 watts or less to be plugged into approved baseboard receptacles, but such heaters cannot be served on a circuit supplying much other load without overloading the fuses. There is an increasing trend toward supplying homes with three wire 115-230 volt service. Where homes have such service, larger heaters can be installed. For industrial purposes, heaters should be designed to use polyphase power, which is usually supplied at 208, 220, 440 or 550 volts. All polyphase heaters should be balanced between phases. In ordering electric heaters the proper voltage must be specified as the heat produced will vary as the square of any variation in voltage.

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CHAPTER 45

Panel Heating and Radiant Heating

Influence of Heat Radiation on Human Comfort, Objectives of Radiant Heating, Practical Problems of Radiant Heating from a Physiological Standpoint, Fundamental Computations, Application Methods, Calculation Principles, Measurement and Control

IT has been pointed out in Chapter 2 that the human body loses heat to its environment in three ways; by convection, radiation, and evaporation. The Effective Temperature Chart takes account of convection and evaporation, but does not provide for such radiative effects as occur when room air and its surrounding surfaces differ widely in temperature.

INFLUENCE OF HEAT RADIATION ON HUMAN COMFORT

When, however, the body is exposed to radiation from a hot surface or is radiating to a cold surface, the factor of radiative heat gain or heat loss may be important. This phenomenon is most marked in the case of exposure to the sun's radiative heat. On a cold day, with no wind blowing, while standing in the sunshine, one may feel perfectly comfortable but, when a cloud passes over the sun, one may instantly feel much cooler. The cloud acts as a shield to interrupt the radiant heat from the sun. The change in feeling of comfort is due to the instant change in rate of heat loss from the body caused by the shielding effect of the cloud. A shielded thermometer under the same condition would register no change in temperature.

The rate of heat loss by convection depends upon the average temperature difference between the surface of the body and the surrounding air, the shape and size of the body, and the rate of air motion over the body.

The rate of heat loss by radiation depends upon the exposed surface area of the body, and upon the difference between the mean surface temperature of the body and the mean surface temperature of the surrounding walls or other objects. This latter temperature is called the Mean Radiant Temperature (MRT)

Because these two types of heat loss supplement each other, a required rate of total heat loss can result either from a relatively low air temperature and a relatively high MRT, or vice versa.

At the temperature which produces comfort (and at all lower temperatures) the production of sweat is low and the heat loss by evaporation is relatively low and relatively constant, irrespective of the relative humidity of the atmosphere. Under such conditions the heat loss from the body is chiefly related to the combined effect of convection and radiation. The heat demand of the environment, so far as these two factors are concerned, may be measured by Operative Temperature, which is defined by the following formula, modified from that of Gagge[1] by the expression of air velocity in feet per minute and temperature in degrees Fahrenheit.

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[1]Standard Operative Temperature, A Generalized Temperature Scale, Applicable to Direct and Partitional Calorimetry, by A P. Gagge (*American Journal Physiology*, 1940, Vol. CXXXI, p. 93).

$$t_O = 0.81\, t_W + 0.135 \left[\sqrt{V}\, t_A - \left(\sqrt{V} - 1.40 \right) t_S \right],$$

where

t_O = operative temperature, degrees Fahrenheit.
t_W = mean radiant temperature, degrees Fahrenheit.
t_A = air temperature, degrees Fahrenheit.
t_S = mean skin temperature, degrees Fahrenheit.
V = air velocity in feet per minute.

At high environmental temperatures, a given Operative Temperature with cold air and hot walls produces a slightly greater cooling effect on the body than the same Operative Temperature with air and wall temperatures equal, probably on account of local cooling of the nose and throat[2]. This phenomenon is not important in the comfort zone.

Under ordinary conditions, with normally clothed human beings in still air, the mean skin temperature of the body is about 90 F (with lower values for the extremities); and the mean temperature of the surface of the clothing is about 86 F.

The normal rate of heat production in an average sized sedentary individual is about 400 Btu per hour. The heat production for persons subjected to various rates of activity is given in Chapter 2. The human body is of complicated shape, and radiation takes place freely only from the exposed outer surfaces; there are considerable portions of the body such as the legs, arms, lower part of the head, etc., which radiate most of their heat to other portions. It is necessary to determine the equivalent surface of the body from which heat is radiated and a similar value for convection. The total may be assumed to be about 19.5 sq ft for convection and 15.5 sq ft for radiation, in an average sized individual.

The loss by respiration and by evaporation from the nose and throat depends on the temperature and area of the moist surfaces (respiratory) of the body, the air temperature, air movement, and humidity. In air at a temperature of 70 F, this loss, for a sedentary individual of average size, will be approximately 90 Btu per hour; and at 60 F about 70 Btu per hour. These values are relative, because the total will vary materially with change of position, bodily activity, age, sex, race, etc.

The balance of the heat generated in the average human body, approximately 300 to 320 Btu per hour at about 70 F room temperature, is the approximate amount of heat given off by radiation and by convection from the external body surfaces. Under normal conditions (in still air), the radiation loss will be about 190 Btu per hour; and the convection loss about 120 Btu per hour. With an air velocity of 520 fpm, comfort will require an increase in Operative Temperature of nearly 12 F; under such conditions the convection loss will rise to 250 Btu per hour but comfort may be attained if the subject is surrounded by heated walls which keep the radiation loss at about 50 Btu[3].

It is neither feasible nor desirable to change the relationships of convection and radiation very greatly in actual heating practice. In the laboratory, where the laws of radiative heat loss have been deduced, it is necessary to produce wide differences between radiative and convective heat loss. This can only be accomplished, however, by elaborate and

[2]Physiological Reactions and Sensations of Pleasantness Under Varying Atmospheric Conditions, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (A.S.H.V.E. TRANSACTIONS, 1938, Vol. 44, p 190).

[3]The Influence of Air Movement Upon Heat Losses from the Clothed Human Body, by C.-E. A. Winslow, A. P. Gagge and L. P. Herrington (*American Journal Physiology*, 1939, Vol. CXXVII, p 505).

powerful conditioning apparatus which simultaneously heats walls and cools air, or vice versa. Such a process would be very costly in practice and would not be justified unless marked improvement in comfort resulted from such a condition—an assumption which has not been demonstrated. In practice, where radiant heat is introduced into a room, that heat is absorbed by surfaces, furniture, and the like, and then transformed into convective heat so that air and surfaces tend to attain a generally uniform temperature.

OBJECTIVES OF RADIANT HEATING

Under ordinary circumstances the human being, indoors, is not subjected to marked variations between the factors affecting convection and radiation. Air and walls are not commonly very far apart in temperature; air movement and relative humidity are usually low. Where such conditions obtain, the ordinary air thermometer is a good measure of comfort—which is the reason why it has enjoyed such universal use. Where considerable window surfaces create heavy radiation loss, or where stoves or open fires, or very hot ceilings contribute to large radiation gain, the picture is changed and the air temperature productive of comfort must be correspondingly modified.

In general, however, radiant heating of occupied spaces is not a procedure designed to create differences between air and walls, but is merely one method of introducing heat into that space. The engineering factors used in determining desirable heat input will be essentially the same as if the heat were introduced by convection, or in any other way.

PRACTICAL PROBLEMS OF RADIANT HEATING FROM A PHYSIOLOGICAL STANDPOINT

It is convenient to distinguish two different methods of introducing radiant heat into an enclosed space. The first, which may be called High-Temperature Radiation, involves direct exposure of the occupied parts of the room to radiation emitted from relatively small heating units of very high temperatures (perhaps 1,000 F); the second, Panel Heating, involves exposure to relatively large surfaces at not over 130 F.

High-Temperature Radiant Heating may be useful for temporary purposes, as in the use of a bathroom heater. It is, however, generally an undesirable process (except in rooms of great height) on account of the marked unevenness of the effect produced on the human body. Studies at the John B. Pierce Laboratory of Hygiene have shown that this type of heating produces uncomfortable differences in the temperature of different parts of the body (an over-heated head, for example, if the heat comes from the ceiling).

Panel Heating, on the other hand, is advantageous from the standpoint of temperature differentials. In actual practice, a well-designed system of this sort produces very uniform conditions, the air throughout the room differing at various points by only 5 deg. This is desirable from the comfort standpoint and may also be a factor in heat economy, since high temperatures in the upper part of the room favor excessive heat loss. The esthetic value of such a system is also considerable, since it avoids the presence of registers or free-standing radiators in the room.

In the design of Panel Heating, however, careful thought must be given to the location of the panels from the standpoint of comfort. The English commonly use the ceiling for their panels, but English rooms are generally high-studded, and outdoor winter temperatures moderate. With low

ceilings even panels may produce an excessive directional heating effect if all the heat necessary in a cold climate is introduced from above. Similarly, if the floor alone is used, it may—in very cold weather—be necessary to make the floor too hot for comfort. Wall panels, or a combination of ceiling and floor panels, will perhaps produce the best results.

FUNDAMENTAL COMPUTATIONS

The mean surface temperature of an inert body, which will cause given rates of heat loss by radiation and by convection in a uniform environment, having a given air temperature and a given mean wall temperature, may be calculated from fundamental equations[4] for radiation and natural convection, with substitution of comparable cylinders for the irregular human body.

$$q_r = 0.1730\, e \left[\left(\frac{T_s}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4 \right] \tag{1}$$

$$q_c = 1.235 \left(\frac{1}{D}\right)^{0.2} \times \left(\frac{1}{T_m}\right)^{0.181} \times \left(T_s - T_a\right)^{1.266} \tag{2}$$

where

q_r = heat loss by radiation, Btu per square foot per hour
q_c = heat loss by convection, Btu per square foot per hour.
T_s = absolute temperature of the body surface, degrees Fahrenheit
T_w = absolute temperature of the walls, degrees Fahrenheit
T_a = absolute temperature of the air, degrees Fahrenheit.
$T_m = \frac{T_s + T_a}{2}$,
D = diameter of cylinder, inches.
e = the ratio of actual emission to black body emission.

If it is assumed that an average adult has a height of 5 ft 8 in., a body surface of 19.5 sq ft for convection, and 15.5 sq ft for radiation, an equivalent effect can be worked out for two cylinders, 5 ft 8 in. high by 13.15 in. diameter and 10.45 in. diameter, respectively. However, while the effects on a cylinder, of a particular size and shape may be used to estimate average similar effects on the human body, it should be remembered that the heat loss from the body varies greatly. Every movement alters not only its shape, but also the heat generated by the body and the velocity of the air passing over it and the surface exposed to radiation. This fact renders the results of any such computation only approximate.

APPLICATION METHODS

The several methods of applying panel and radiant heating to a structure are:

1. *By warming the interior wall and ceiling surface of the building.* Pipe coils are imbedded in the concrete or plaster of the walls or ceilings, the heating medium being hot water circulating through the pipe coils. These coils are generally constructed of small pipe ½ or ¾ in. I.D. and spaced about 6 to 9 in. apart. See Fig. 1. This has the effect of warming the entire concrete or plaster surface in which the pipes are imbedded. Since the temperature of the heating medium should never exceed about 130 F, due to the possibility of cracking the plaster the area of the warmed surface must be sufficient to supply the requisite quantity of heat at this low temperature. When carefully designed, this method produces very comfortable results and great operating economy, but offers some slight obstacles when alterations or additions to the building are desirable. Normally the hot water circulation is maintained by means of a circulating pump and facilities have to be provided to eliminate all air at the top of the system. All coils and

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[4]Surface Heat Transmission, by R. H. Heilman (*A.S.M.E. Transactions, Fuels and Steam Power Section*, Vol. 51, No. 22, September-December, 1929).

circulating pipes are welded together and tested after erection to a hydraulic pressure of 300 psi.

2. *By circulating warm air through shallow ducts under the floor.* In this design the entire floor surface of a room is heated as in Fig. 2. This method was used 2000 years ago in many parts of the Roman Empire. While this method is more expensive in construction, it is effective and quite suitable for cathedrals and large public buildings.

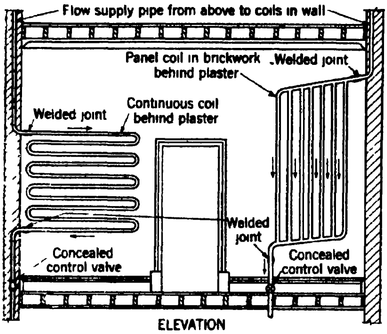

FIG. 1. COILS IN WALL SURFACES

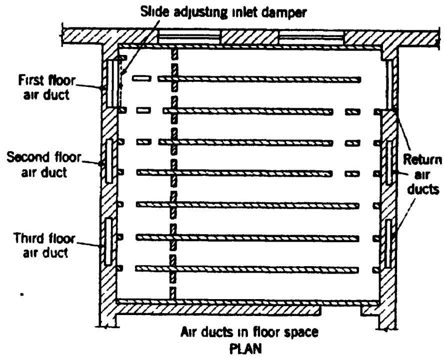

FIG. 2. AIR DUCTS FOR FLOOR HEATING

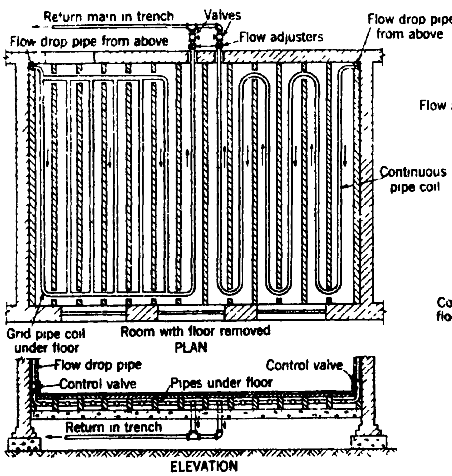

FIG. 3. CONTINUOUS COIL IN FLOOR

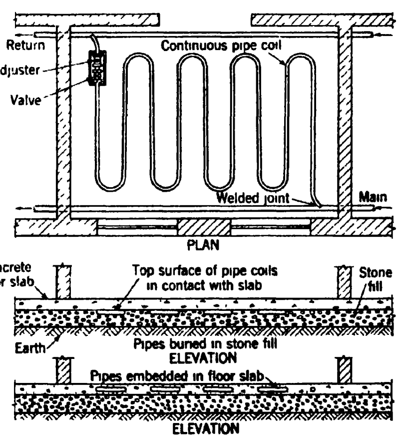

FIG. 4. COILS EMBEDDED IN FLOORS

To provide a uniform floor temperature, one should give special consideration to the design of the air ducts so that equal heat distribution is obtained.

3. *By placing hot water pipes in or under the floor.* With this arrangement the whole floor surface of a room is raised to a temperature sufficient to give comfortable conditions. Floor heating is recommended for schools and hospitals where large quantities of outside air are desirable. The floor surface may be of concrete, wood blocks, marble or any other material unaffected by heat, and while it is true that heat will be conducted through all materials used in floor construction, it is important that due consideration be given to the emissivity of the floor. In some cases where pipe coils are installed in the air space under the floor, special floors are constructed in sections so that the whole floor can be lifted to examine the coils. See Fig. 3. Pipes supported thus may be larger and the heating medium maintained at a higher temperature than when pipes are actually imbedded in the floor. Pipes may be $1\frac{1}{2}$ or 2 in. in the former, but for the latter $\frac{3}{4}$ or 1 in pipes are recommended. See Fig. 4. Where the heat losses from a room are ex-

ceptionally high it may be necessary to supplement the warm floor by either adding some coils in the ceiling or forming heated panels in the side walls.

4. *By attaching separate heated metal plates or panels to the interior surfaces.* These plates or panels are placed either in an insulated recess so that the surface of the panel is flush with the surface of the walls or ceilings, or they may be secured to the face of the wall. They may be covered with wood veneers and decorated to harmonize with other parts of the room, or they may be cast into panels to imitate oak or other wood designs. With flat plate panels it is common practice to use a frame of plaster, wood, metal or composition to allow for expansion. These plates may be heated with either hot water or steam and connected as in an ordinary radiator system. See Figs. 5 and 6.

5. *By electric heated metal plates or panels.* These plates or panels are either placed in insulated recesses of walls or ceilings or fastened to the construction, as found desirable.

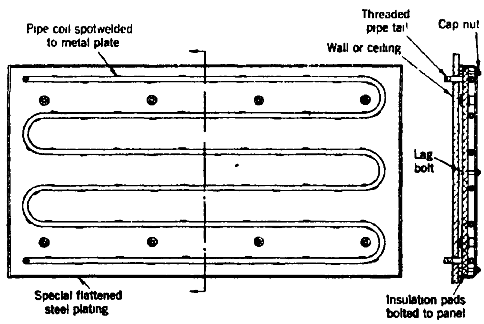

FIG. 5. PILLAR TYPE RADIANT HEAT PANEL

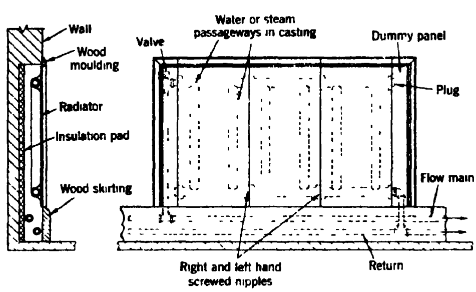

FIG 6. FLAT TYPE PANEL INSTALLED IN WALL RECESS

They should not have a surface temperature much above 200 F. Some have a much higher surface temperature but a lower temperature gives a more comfortable condition and is more efficient.

6. *By electrically heated tapestry mounted on screens and on the wall.* For this purpose the screen is woven with an electric continuous conductor. Such screens are useful to plug in at any position for emergency local heating without taking care of a large room or office.

Note. If all of a heating panel is installed at one end of a large room there may be a marked difference between the equivalent temperature on the two sides of the body. It is usually desirable, therefore, that the heat be distributed at different parts of the walls and ceilings so that no uncomfortable effect will be felt from unequal heating.

CALCULATION PRINCIPLES

Part I—Panel Heating

The term *panel heating*, involving both radiation and convection, is applied in this chapter to a system in which the heat is transmitted from

panel surfaces to both air and surrounding surfaces, as is the case under indoor conditions.

Panel heating systems for buildings may be designed as illustrated and described in the following design of a panel heating system for the room shown in Fig. 7. The design is based on continuous heating. Panel heating systems should, in general, be operated continuously since the panels have large thermal capacities. Where panels are heated by high temperature radiation from a heat source directed toward them, this qualification does not apply.

1. *Assume the location and the approximate size of the heating panel.*

Heating panels may be located in ceilings or floors or walls. Ceiling panels have the advantage that their heat emission is not affected by tapestry or furniture and that they can be used, to a limited extent, as cooling panels during the summer months. If used in low rooms, however,

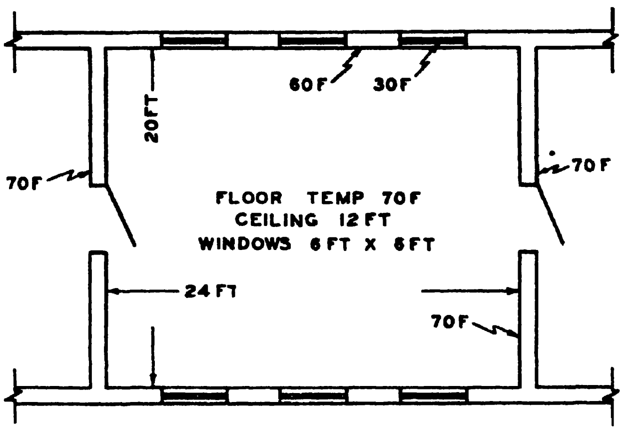

FIG. 7. ROOM PLAN USED FOR ILLUSTRATION OF METHOD OF DESIGNING A PANEL HEATING SYSTEM

they may produce an undesirable heating effect upon the head. Floor panels have the advantage that they can be easily installed and that much of the radiated heat is delivered to the lower portions of the walls; they have the disadvantage that their heat emission is rather uncertain, since it may be affected by rugs, carpets, furniture, machinery, etc.

It is best to make the panels as large as practicable; for example, if the floor or ceiling is used as the heating panel, it is best to use the entire floor or the entire ceiling, or both. Heating a room by means of panels is very similar to lighting a room. Heat radiation is exactly like light radiation, except that it has a longer wave length. If a room is lighted by means of a large number of small units distributed uniformly over the ceiling, the room is lighted more uniformly than if it is lighted by means of a single unit of equal capacity. Similarly, if the entire ceiling is the heating panel, the room is heated more uniformly than if only a fractional part of the ceiling is used as the heating panel.

In the following example, *the entire ceiling will be used as the heating panel.*

2. *Select the desired mean temperature of the air in the room.*

In a panel-heated room, the mean air temperature is a few degrees

lower than the mean temperature of the surfaces of the enclosing walls, floor, and ceiling. In a room heated by introduction of warm air or by means of radiators, convectors, or other similar heating appliances, located within the room, the mean temperature of the air is a few degrees higher than the mean temperature of the surfaces of the enclosing walls, floor, and ceiling. Under ordinary conditions, in a panel-heated room, the mean temperature of the air ranges from approximately 65 F to 72 F.

In the following example, *68 F is selected as the mean air temperature.*

3. *Determine as accurately as practicable, the mean temperature of the inside surfaces of the enclosing walls, floor, and ceiling.*

The two inside walls are assumed to separate rooms, which are filled with 68 F air, so that both surfaces of each wall are in close contact with

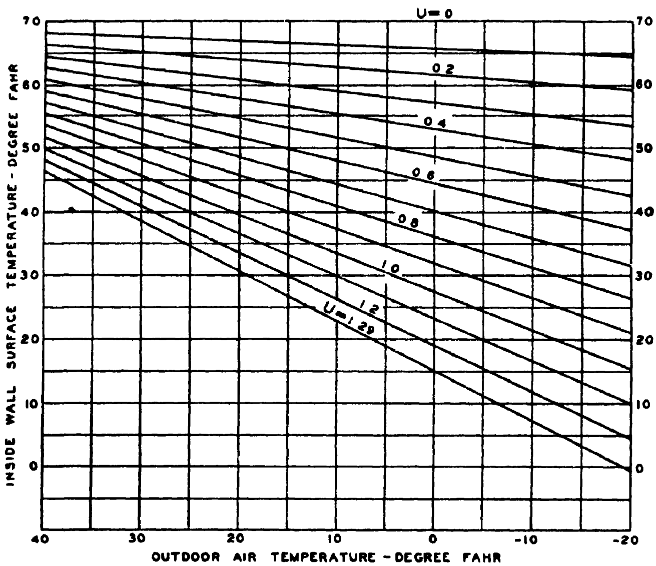

FIG. 8. CHART FOR ESTIMATING INSIDE SURFACE TEMPERATURES OF OUTSIDE WALLS[a]

[a]NOTE: The value of U, the over-all coefficient of heat transmission, cannot exceed 1 29 if the inside and outside film coefficients are 1.65 and 6 0 respectively (e g. $\frac{1}{U} = \frac{1}{1.65} + \frac{1}{6} = \frac{1}{0\ 773}$. Therefore U = 1 29 maximum)

68 F air. The surface temperatures of these walls, therefore, can not be lower than 68 F and must actually be higher than 68 F because, in addition to their contact with 68 F air and their contact with the heated ceiling, they are exposed to the heat radiation from the ceiling.

In the following example, *70 F will be selected as the mean surface temperature of the inside walls.*

For the two outside walls, the mean inside surface temperature can be calculated with fair accuracy. For a heat transmission coefficient of 0.25 and a temperature difference of 68 deg, heat flows through the wall at the rate of 17 Btuh per square foot. If the indoor film coefficient is 1.65, the temperature difference, indoor air to inside wall surface, is 17/1.65 or 10 deg and the wall surface temperature is 68 − 10, or 58 F. This value may be taken directly from Figure 8. However, the film coefficient 1.65 was determined to represent the sum of the heat flow into the wall, by

TABLE 1. CALCULATED HEAT LOSS OF ROOM

SURFACE	AREA SQ FT	U	CALCULATION	HEAT LOSS BTUH
Outside Walls	360	0.25	360 x 0.25 x 68	6,120
Glass	216	1.13	216 x 1.13 x 68	16,597
Inside Walls	480	....	No heat loss	....
Ceiling	480	....	Heating Panel	....
Floor	480	0.10	480 x 0.10 x 38	1,824
Infiltration	......	....	5,760 cu ft x 1.50 x 68 x 0.018	10,576
			Total	35,117

conduction from the air in contact with the wall, and by radiation from the warmer surfaces *seen* by the wall surface. In a panel-heated room, the rate of heat flow into the outside wall by radiation is greater than it is in a radiator-heated room; consequently, the film coefficient is higher, and the temperature difference, air to wall surface, is smaller, and therefore, the wall surface temperature is higher than the calculated 58 F. It is impossible to determine accurately how much higher than 58 F the temperature of the wall surface will be until the corresponding indoor air film coefficient has been determined accurately.

In the following example, *60 F will be selected as the probable mean inside surface temperature of the outside walls.*

The probable mean inside surface temperatures of the floor and the glass may be determined by calculations and by reasoning similar to that employed to determine the inside surface temperature of the outside walls.

In the following example, *30 F and 70 F will be selected as the probable inside surface temperatures of the glass and floor*, respectively.

4. *Determine the heat loss of the room.*

In the following example, the heat loss calculation will be based on an outdoor air temperature of 0 F. Since the functioning of a panel-heating system differs very little from that of a radiator-type heating system, the heat loss may be calculated according to Chapter 6 as shown in Table 1.

The heat loss through the outside walls and through the glass is probably a little greater than calculated because the calculation is based on an indoor air film coefficient of 1.65 Btuh, whereas, for a panel-heated room, this coefficient is a little higher, but the difference is probably not sufficiently large to be considered in design calculations for a heating system.

5. *Estimate the Mean Radiant Temperature.*

The Mean Radiant Temperature of the surfaces enclosing the room but not including the heating panels may be estimated as follows:

SURFACE	AREA	DEG FAHR	PRODUCT
Interior Walls	480	70	33,600
Exterior Walls	360	60	21,600
Glass	216	30	6,480
Floor	480	70	33,600
	1,536		95,280

The sum of these products divided by the sum of the surface areas is: 95,280/1,536 or 62.03 F, the required mean surface temperature.

In the following example, *62 F will be selected as the MRT of walls, glass and floor.*

6. *Determine the temperature of the ceiling panel.*

Determine the temperature of the ceiling so that the ceiling panel will deliver heat to the room at a rate equal to the rate at which the room is calculated to lose heat, namely, 35,117 Btuh.

When a room is heated by means of a panel, air convection currents are developed in the room similar to those which are developed when the room is heated by means of a free-standing radiator. Consequently, the heating panel delivers heat to the room partly by radiation and partly by convection. The proportion of the total heat flow delivered by convection varies with the location of the heating panel, with the height of the ceiling, and with the size, number, and location of pieces of furniture and other articles which interfere with the free flow of air along the floor and along the walls. It is generally sufficiently accurate to assume that a ceiling panel will deliver 70 per cent of its heat by radiation and 30 per cent by convection; a floor panel 55 per cent by radiation and 45 per cent by convection; and a wall panel 65 per cent by radiation and 35 per cent by convection.

In the following example, *it will be assumed that the ceiling panel must deliver 70 per cent of its heat or 24,582 Btuh by radiation,* since the total calculated heat loss is 35,117.

When two plane surfaces of infinite size are parallel to each other and their surfaces are at different temperatures, the exchange of heat between the two is proportional to the difference between the fourth powers of their absolute temperatures. This is also true when one surface is completely surrounded by another surface; for example, if one sphere is placed within another sphere, the flow of heat between the outer surface of the smaller sphere and the inner surface of the larger sphere is proportional to the fourth power of the absolute temperatures of the two surfaces.

In a panel-heated room, the heated panel may be considered to be completely enclosed by the remaining surfaces, because all heat radiated by the heated panel is intercepted by those surfaces. Consequently, the flow of heat from the heated ceiling to the room, by radiation, is proportional to the difference between the fourth powers of the absolute temperature of the ceiling and the absolute mean radiant temperature of the remaining surfaces.

The rate at which a surface emits heat varies with the temperature of the surface and with other characteristics of the surface. For ordinary heat flow calculations it is sufficiently accurate to assume that the materials which are commonly used in building construction emit heat at a rate of:

$$0.156 \left(\frac{T}{100}\right)^4 \text{ Btuh per square foot}$$

where

T is the absolute temperature of the surface in degrees Fahrenheit.

On this basis the flow of heat from the ceiling to its surrounding surfaces is at the rate of:

$$480 \times 0.156 \left[\left(\frac{T}{100} \right)^4 - \left(\frac{522}{100} \right)^4 \right] \text{ Btuh.}$$

In order that this rate may be equal to 24,582 Btuh, T must be 572 and the ceiling temperature about 112 F.

Instead of calculating this temperature it may be taken from Fig. 9, as follows: The ceiling must deliver heat to the room, by radiation, at the rate of 24,582/480 or 51 Btuh per square foot. Find 51 on the left margin and move horizontally to the intersection with a 62 MRT line, and from the point of intersection to the lower margin and read about 112 F.

With a ceiling temperature of 112 F, the MRT of the room will be $480 \times 112 + 1{,}536 \times 62$; the sum divided by 2,016, or 74 F.

If an air temperature of 68 F and an MRT of 74 F should not produce satisfactory conditions, the ceiling temperature can easily be changed as necessary by changing the temperature of the circulating water.

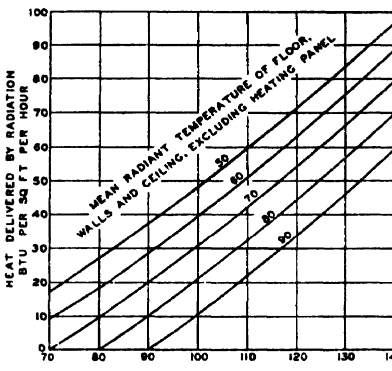

FIG. 9. HEAT DELIVERED TO ROOM BY RADIATION FROM PANEL

Calculations like the preceding may also be made with the aid of Table 2. The rate at which the ceiling must radiate heat exceeds by 51 Btuh per square foot the rate at which the ceiling receives radiant heat from its surroundings. Assuming the emissivity of the walls, floor, and ceiling to be 90 per cent of that of a *black* body, the heat radiated to the ceiling, from the surfaces whose MRT is 62 F, is (Table 2) at the rate of 115.3 Btuh per square foot; the ceiling must therefore radiate heat at the rate of 115.3 plus 51, or 166.3; its temperature must be (Table 2) between 110 F and 120 F, and, by interpolation, 112 F, as calculated.

7. *Select the medium for heating the ceiling panel.*

The medium may be electricity, steam, air, or water, but usually is air or water. If air is used it is generally heated in the basement, passed up through hollow inside walls or through ducts in those walls, allowed to flow between the ceiling and the floor above, and returned to the basement through hollow outside walls or through ducts in those walls.

If the walls and floors are constructed of hollow tile, the cells in the tile can be placed so that they will form continuous ducts through which the warm air can flow up the inside walls, then between the ceiling and

the floor above, and down the outside walls. In this way the walls and ceiling become heating panels.

If water is used as the medium, the pipes through which the water circulates—almost always under forced circulation—are placed in the floor, walls, or ceiling in such a manner that as much as possible of the heat emitted by the pipes will be delivered to the space to be heated.

TABLE 2. TOTAL HEAT EMISSION BY RADIATION[a]

Body or Mean Radiant Temperature Deg F	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e				Body or Mean Radiant Temperature Deg F	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e			
	e 1.00	e 0.95	e 0.90	e 0.80		e 1.00	e 0.95	e 0.90	e 0.80
30	99.7	94.7	89.8	79.7	71	137.0	130.1	123.3	109.7
35	103.9	98.7	93.6	83.1	72	137.9	131.0	124.0	110.3
40	108.0	102.8	97.2	86.4	73	138.9	132.0	124.9	111.0
45	112.5	106.9	101.3	89.0	74	140.2	133.1	126.1	112.1
46	113.3	107.7	102.0	90.8	75	141.6	134.4	127.4	113.2
47	114.3	108.6	102.9	91.5	80	147.2	140.0	132.5	117.9
48	115.2	109.5	103.8	92.3	85	152.9	145.2	137.6	122.4
49	116.0	110.3	104.5	92.8	90	158.5	150.5	142.7	126.9
50	116.9	111.0	105.3	93.6	100	170.3	161.7	153.2	136.2
51	117.9	112.0	106.4	94.4	110	182.3	173.2	164.2	146.0
52	118.8	112.9	106.9	95.1	120	195.6	185.7	176.1	156.5
53	119.8	113.8	107.8	95.9	130	210.9	200.4	189.8	168.8
54	120.6	114.6	108.6	96.6	140	224.1	212.9	201.8	179.2
55	121.7	115.5	109.4	97.3	150	238.0	226.1	214.4	190.5
56	122.6	116.4	110.3	98.1	160	252.1	239.8	226.9	201.8
57	123.5	117.4	111.3	98.9	170	271.6	258.0	244.5	217.2
58	124.4	118.2	112.0	99.6	180	289.1	274.9	260.1	231.3
59	125.3	119.0	112.8	100.3	190	307.7	292.1	276.9	246.1
60	126.3	119.9	113.8	101.1	200	326.5	310.2	293.9	261.3
61	127.1	120.7	114.4	101.8	210	349.3	331.9	314.3	279.5
62	128.2	121.8	115.3	102.6	220	373.0	354.4	335.7	298.2
63	129.1	122.6	116.2	103.3	250	439.5	417.6	395.6	351.6
64	130.1	123.5	117.1	104.1	300	577.3	548.2	519.6	461.8
65	131.0	124.4	117.9	104.8	350	743.0	705.8	668.6	594.3
66	132.1	125.5	118.8	105.8	400	945.8	898.5	850.8	756.5
67	133.0	126.4	119.7	106.4	450	1181.0	1121.0	1063.0	944.0
68	134.0	127.3	120.5	107.2	500	1470.0	1396.0	1323.0	1176.0
69	135.0	128.3	121.5	108.0	550	1798.0	1708.0	1619.0	1439.0
70	136.0	129.3	122.3	108.8	600	2181.0	2072.0	1962.0	1745.0

[a]These factors are calculated from the formula

$$q_r = e\left(\frac{0.173 \times T^4}{100{,}000{,}000}\right)$$

where

q_r = total radiation, Btu per (sq ft) (hr)
e = emissivity.
T = absolute temperature, degrees Fahrenheit.

Practical limits for surface temperature of heating panels are given in Table 3.

In this example water will be selected as the medium.

8. *Determine the size, length, and location of the pipe coils in the panels.*

When hot-water pipes are imbedded in concrete slabs or attached to plastered surfaces, their rate of heat emission varies with many factors. If the pipes are imbedded in dense concrete slabs, it may be assumed that the rate of heat emission of ½-in. pipe, spaced 6 in. on centers; ¾-in. pipe, spaced 9 in. on centers; and 1-in. pipe spaced 12 in. on centers; per foot of length of pipe and per degree difference between the temperature of the water in the pipe and that of the air in the space to be heated, is

0.8, 1.0, and 1.2 Btuh, respectively. If the distance between the pipes is increased, the rate of heat emission, per foot of pipe, is also increased; if the distance is doubled, the rate of heat emission is increased about 15 per cent. If the pipes are attached to plastered ceilings, the rate of heat emission is slightly less, probably about 10 per cent less, than when the pipes are imbedded in concrete slabs. The data given regarding heat emission of panels are intended as general guides for the designer. Additional experience and research are needed to develop definite and complete data. However, after a heating panel has been designed and installed, any small error can easily be corrected by modifying the temperature of the water circulating through the coils.

When the heating pipes are attached to a plastered ceiling, a portion of the heat emitted by the pipes is delivered to the space below the ceiling and a portion to the space above the ceiling. The relative quantities depend on the degree of insulation applied above the heating coils.

When the heating pipes are imbedded in a concrete floor slab a portion

TABLE 3 HIGHEST SAFE SURFACE TEMPERATURES FOR HEATING PANEL

TYPE OF PANEL	SURFACE TEMPERATURE DEG F
Plastered Ceiling (Pipes Imbedded)	115
Plastered Walls (Pipes Imbedded)	120
Floor, Any Method	90
Floor, Border and Aisles	120
Iron, Hot Water Medium[a]	160
Iron, Steam Vapor[a]	180
Electrically Heated Panels[a]	200

[a]Low surface temperature radiation is recommended regardless of the heating medium employed.

of the heat emitted by the pipes will flow upward into the space to be heated, and the remainder will flow downward into the ground.

When the heating pipes are placed below the concrete floor slab instead of being imbedded in the slab, a larger portion of the heat will flow into the ground, and a smaller portion into the space to be heated.

In the following example *it is assumed that the insulation above the pipe coils is such that 90 per cent of the heat emitted by the pipe coils will flow into the room and 10 per cent into the space above.*

Since the room is to receive 35,117 Btuh, and since the room is assumed to receive only 90 per cent of the heat emitted by the coils attached to the plastered ceiling, the coils must emit 35,117/0.9 or 39,000 Btuh. If ¾-in. pipe and a mean water temperature of 140 F are selected, the heat emitted, per foot of pipe, will be 0.9 (140 − 68) or 65 Btuh. The quantity of pipe required will therefore be 39,000/65 = 600 lineal feet.

The pipe coils can be arranged in any convenient manner, but should be arranged so that the temperature of the water in the pipe will vary only slightly; otherwise, the temperature distribution over the ceiling will not be uniform. Generally, it is best to arrange the pipes so as to form two-pipe reversed-return systems, as suggested by the two sketches in Fig. 10. By using 33 runs of ¾-in. pipe, welded to two 1¼ in. mains, sufficient pipe surface is secured; the ¾-in. pipes will then be spaced about 8½ in. on centers, which is satisfactory.

The friction heads of water flowing in pipes and fittings are so well known that the pipe coils can be designed so that each will receive its

proper share of the hot water without the use of balancing valves. However, it is desirable to divide large heating systems into sections and to install valves so that individual sections can be disconnected without interfering with the operation of the system as a whole.

Part II—Radiant Heating

The term *radiant heating* is applied in this chapter to a system in which only the heat radiated from the panel is effective as in outdoor and semi-outdoor conditions.

The outstanding example of radiant heating is the transfer of heat from the sun to the earth. The sun radiates large quantities of energy

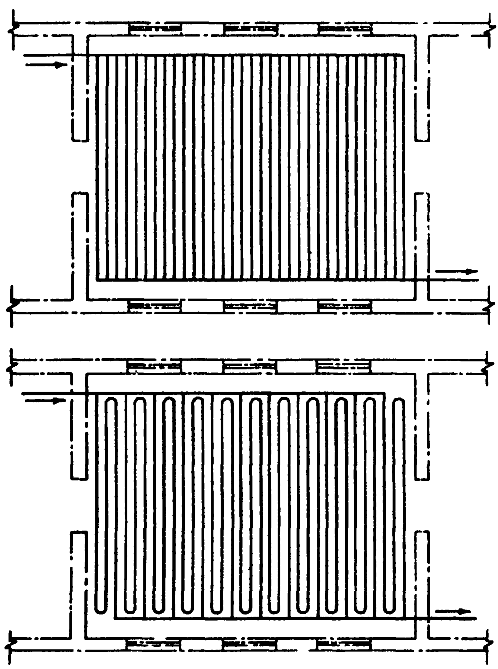

FIG. 10. ARRANGEMENT OF PIPE COIL FOR EVEN DISTRIBUTION OF PANEL TEMPERATURE

of which a very small portion is intercepted by the earth. A part of the intercepted radiation is transformed into heat when it strikes the earth's surface. In this manner *heat* is received by the earth from the sun by *radiation.*

In industry, radiant heating is employed in manufacturing processes, particularly in drying, baking, and dehydrating operations; in agriculture, it is employed to improve living and growing conditions for young plants and young animals.

The heating engineer employs radiant heat primarily in the heating of open-air schools and open-air hospitals. When a surface radiates heat, and every surface does unless its temperature is absolute zero, every point of the surface radiates heat in all directions. The total quantity of heat radiated by a point or by an elementary area is π times the quantity of heat radiated at right angles to the surface.

Thus, if in an elementary cube the upper face is the heating panel, the lower face would receive only about 32 per cent of the radiated energy and the four sides would receive each about 17 per cent.

For larger surfaces the conditions are different. If two parallel plane surfaces of considerable size are near each other, the rate of heat exchange between the two can be determined fairly accurately by means of the chart of Fig. 9. This is possible because the larger part of the heat radiated by one of the surfaces is intercepted by the other surface and only a small portion is radiated in such directions that it will not impinge upon the opposite surface.

As the distance between the two surfaces is increased, the proportion of the heat radiated by one of the parallel plane surfaces and intercepted by the other decreases almost as the square of the distance between the surfaces increases, because the intensity of heat radiation, like the intensity of light radiation, varies inversely as the square of the distance from the source of radiation.

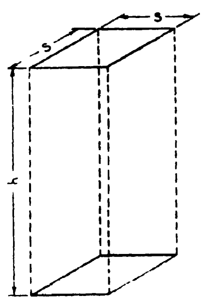

Fig. 11. Effect of Height Upon Radiation Received from a Panel

For the purpose of designing radiant heating systems in which the heating panel is practically square and is radiating heat toward a parallel surface of equal size and shape, as shown in Fig. 11, the rate of heat exchange between the two surfaces will be equal to that shown in Fig. 9, multiplied by a factor, p, which depends upon the ratio of h to s (Fig. 11) as shown in the following table:

$h/s =$	1	2	3	4	5
$p =$	0.200	0.070	0.034	0.020	0.013

For example, if a panel 3 ft square is located parallel to, and 9 ft above, a bed in an open-air hospital, and if the temperature of the panel is 112 F and that of the bed is 70 F, the rate of heat transfer from the panel to a 9 sq ft section of the bed directly beneath the panel, will be 3.4 per cent of the rate shown in Fig. 9, or $0.034 \times 9 \times 44$, or 14 Btuh, approximately. The rate of heat transfer from the panel to a section of the bed other than the 9 sq ft directly beneath the panel will be lower than 14/9 Btuh per square foot.

This is a crude way of designing a radiant heating system for an open-air hospital, but it is sufficiently accurate, because the required temperature of the bed and the required rate of heat flow into it will vary with the temperature of the outdoor air, with the air movement over the bed, with the thickness and the character of the bedding, and with the physical condition of the patient.

A radiant heating system for an open-air school may be designed as described for the open-air hospital. The heating panel in such a case should be almost as large as the ceiling and, in order to keep the heat loss by radiation at a minimum, should be placed so that a maximum portion of the heat radiated by the panel will be directed toward the pupils and a minimum toward the outside walls and particularly the windows.

MEASUREMENT OF RADIANT HEATING

Radiant heating is intended to control the rate of radiant heat loss from the human body and should be measured by calorimetric methods.

The apparatus for this purpose consists essentially of a cylinder, maintained at the accepted mean surface temperature of the human body, together with an accurate (usually electrical) measuring of the varying rate of heat supply required to maintain this exact temperature. This instrument, the *eupatheoscope*, is readily adapted to function like a thermostat so as to turn heat on or off, when the desired temperature of 80 F, or any other predetermined surface temperature of the cylinder, decreases or increases as a result of changes in the Operative Temperature.

For testing work, the *globe thermometer* is a useful instrument. It consists of an ordinary mercury thermometer, with its bulb placed in the center of a sphere from 6 to 9 in. in diameter, usually made of thin copper and painted black and sometimes covered with cloth. The temperature recorded by thermometer with its bulb in the center of the sphere is termed the *radiation-convection temperature*. See Chapter 34.

CONTROL OF PANEL AND RADIANT HEATING

The effectiveness of any type of control will depend largely on the time lag of the system. With warm air passing through floor ducts the time lag is usually too long for any kind of room thermostat, in fact a thermostat will not prove suitable with any system if the building is constructed with massive brickwork and masonry, unless it operates in conjunction with a time control responsive to changes in outside conditions.

The heat emitted by hot water pipes imbedded in the plaster of the ceiling and walls or in the concrete base of a floor can be effectively controlled by an instrument designed to modulate the temperature of the water circulating in the system according to the outside conditions. Metal panels which can be installed in the ceiling or side walls may be either controlled by an instrument responsive to outside weather conditions or by a specially designed instrument responsive to both air temperature and radiation. Any purely *on or off* control system is not recommended for panel heating.

A typical control system operated from an outside thermostat and supplemented with a room heat control instrument is illustrated in Fig. 12. The outside thermostat modulates the temperature of the circulating water in the coils by mixing some of the hot water leaving the boiler with a proportionate amount of return water which is diverted to the three-way valve.

One type of room instrument consists of a blackened copper sphere of 6 or 8 in. in diameter, in which a cylindrical sump contains a volatile liquid. A small electric heating coil creates in the sphere a vapor pressure which remains constant as long as the total heat loss from the sphere is at the desired rate. If the Operative Temperature becomes too high for comfort, a greater vapor pressure results from the smaller heat loss from the sphere. This acts on a diaphragm and reduces the supply of heat to

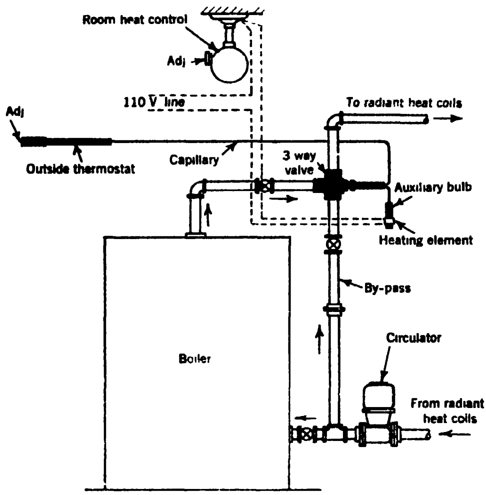

FIG. 12. TYPICAL PANEL AND RADIANT HEAT CONTROL SYSTEM

the room. With too low an Operative Temperature the reverse action occurs. A similar instrument which has an electric heating element for warming the air inside the sphere and the thermostat operated switch is also used for controlling room conditions.

In addition to a thermostatically controlled device for modulating the temperature of the circulating water, it is advantageous to insert in each coil a locked flow control or adjustable resistance to give uniform conditions throughout all rooms. Owing to unforeseen difficulties with varying frictional losses in pipes, emission factor, and exposures, it is an advantage to be able to regulate permanently the flow through each circuit by means of a key operated valve as indicated in Fig. 4.

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CHAPTER 46

Hot Water Supply

Hot Water Supply Piping, Storage Capacity and Heating Load, Methods of Heating Water, Computing Grate and Coil Surface Area, Controls, Solar Water Heaters, Friction Loss

IN computing the total heating load for a building, it is important to allow ample boiler capacity for heating the hot water supply. The amount of warm water used in any large building is variable, depending on the type of structure, usage, occupancy and time of day. It is necessary to provide the piping, water heating and storage facilities of sufficient capacity to meet the peak demand without wasteful excess in equipment cost.

The determination of the amount of water used in residences has been well established over a period of time and as a result, reliable factors for water consumption are available. Tests have been made of the amount of water required by standard fixtures in normal use with water at ordinary pressures so that this information permits a fairly correct basis of design.

HOT WATER SUPPLY PIPING

As a result of investigations conducted at the *National Bureau of Standards*, basic design principles have been outlined for the design of the hot and cold water supply piping requirements in a plumbing system[1].

It is common practice to provide circulating piping in all hot water supply systems in which it is desirable to have hot water available continuously at the fixtures. In average sized and small residences and systems, in which the piping from the heater to the fixtures is short, return circulating piping is generally omitted in order to reduce installation cost and to reduce heat loss from the piping, particularly during periods of no water demand.

The hot water supply may be distributed by either an up-feed or down-feed piping system. Three common methods of arranging the circulating lines are shown in Fig. 1. Although the diagrams apply to multi-story buildings the arrangements (*a*) and (*b*) are sometimes used in residential designs.

A check valve should be provided in the run-out from each return riser to prevent temporary reversal of flow in the piping when a faucet is open. Proper air venting of a circulating system is extremely important, particularly if gravity circulation is employed. In Fig. 1 (*a*) and (*b*) this is accomplished by connecting the circulating line below the top fixture supply. With this arrangement, air is eliminated from the system each time the top fixture is opened.

Where an overhead supply main is located above the highest fixture as in Fig. 1 (*c*), an automatic float type air vent is installed at the highest point of the system or a fixture branch is connected to the top of the main where air venting is desired and then dropped to the fixture outlet.

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[1]Methods of Estimating Loads in Plumbing Systems, by R B Hunter (*National Bureau of Standards*, Report BMS65, 1940). Plumbing Manual, Report of the Subcommittee on Plumbing, Central Housing Committee on Research, Design and Construction (*National Bureau of Standards*, Report BMS66, 1940). Water-Distributing Systems for Buildings, by R B Hunter (*National Bureau of Standards*, Report BMS79, 1941)

It is sometimes necessary to make an allowance for pressure drop through the heater when sizing hot water lines. This is particularly true where instantaneous hot water heaters are used and the available pressure is low.

STORAGE CAPACITY AND HEATING LOAD

In estimating the size of hot water storage tank required and the heating capacity to be provided either from the boiler or from an independent domestic hot water heater, it is necessary to know the total quantity of water to be heated per day, and the maximum amount which will be used in any one hour, as well as the duration of the peak load.

In cases where the requirements for hot water are reasonably uniform, as in residences, apartment buildings, hotels, and the like, smaller storage capacity is required than in the case of factories, schools, office buildings, etc., where practically the entire day's usage of hot water occurs during a

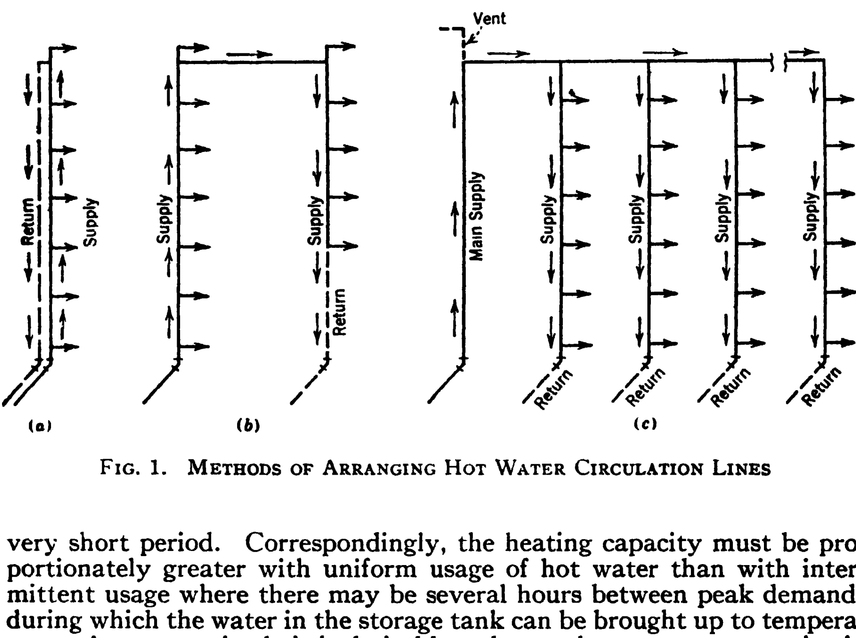

FIG. 1. METHODS OF ARRANGING HOT WATER CIRCULATION LINES

very short period. Correspondingly, the heating capacity must be proportionately greater with uniform usage of hot water than with intermittent usage where there may be several hours between peak demands during which the water in the storage tank can be brought up to temperature. As a general rule it is desirable to have a large storage capacity in order that the heating capacity and consequently the size of the heater, or the load on the heating boiler, may be as small as possible.

In estimating the hot water which can be drawn from a storage tank it should be borne in mind that only about 75 per cent of the volume of the tank is available, as by the time this quantity has been drawn off the incoming cold water has cooled the remainder down to a point where it can no longer be considered hot water.

Where steam from the heating boiler is used to heat domestic hot water, the computed load on the boiler should be increased by 4 sq ft EDR (equivalent direct radiation) for every gallon of water per hour heated through a 100 F rise. The actual requirement is $\frac{100 \times 8.33}{240} = 3.48$ sq ft

per gallon of water heated 100 F. The value of 4 allows for transmission losses, etc.

There are two ways in common use of estimating the hot water requirements of a building; first, by the number of people and second, by the number of plumbing fixtures installed. Where the number of people to be served is known or can be reasonably estimated, the data in Table 1 may be used.

Example 1. From Table 1, a residence housing five people would have a daily requirement of $5 \times 40 = 200$ gal per day, and a maximum hourly demand of $200 \times 1/7 = 28.5$ gal. The heater should have a storage capacity of $200 \times 1/5 = 40$ gal and a heating capacity of $200 \times 1/7 = 28.5$ gal per hour.

The conditions given in *Example 1* may be cited as average. It is possible to vary the storage and heating capacity by increasing and

TABLE 1. ESTIMATED HOT WATER DEMAND PER PERSON FOR VARIOUS TYPES OF BUILDINGS

Type of Building	Hot Water Required at 140 F	Max Hourly Demand in Relation to Day's Use	Duration of Peak Load Hours	Storage Capacity in Relation to Day's Use	Heating Capacity in Relation to Day's Use
Res., apts., hotels, etc.	40 gal per person per day	1/7	4	1/5	1/7
Office buildings	2 gal per person per day	1/5	2	1/5	1/6
Factory buildings	5 gal per person per day	1/3	1	2/5	1/8
Restaurants $0.50 meals $1.00 meals $1.50 meals	 1.5 gal per meal 2 5 gal per meal 4.5 gal per meal			1/10	1/10
Restaurants 3 meals per day		1/10	8	1/5	1/10
Restaurants 1 meal per day		1/5	2	2/5	1/6

decreasing one over the other. Such a condition is illustrated in *Example 2.*

Example 2. Assume an apartment house housing 200 people. From the data in Table 1: Daily requirements $= 200 \times 40 = 8000$ gal. Maximum hours demand $= 8000 \times 1/7 = 1140$ gal. Duration of peak load = 4 hours Water required for 4-hour peak $= 4 \times 1140 = 4560$.

If a 1000 gal storage tank is used, hot water available from the tank $= 1000 \times 0.75 = 750$. Water to be heated in 4 hours $= 4560 - 750 = 3710$ gal. Heating capacity per hour $= \frac{3710}{4} = 930$ gal.

If instead of a 1000 gal tank, a 2500 gal tank had been installed, the required heating capacity per hour would be $\frac{4560 - (2500 \times 0.75)}{4} = 671$ gal.

Table 2 may be used to determine the size of water heating equipment from the number of fixtures. To obtain the *probable maximum demand* multiply the total quantity for the fixtures by the Demand Factor in line 11. The heater or coil should have a water heating capacity equal

to this *probable maximum demand.* The storage tank should have a capacity equal to the *probable maximum demand* multiplied by the storage capacity factor in line 12. *Example 3* will illustrate the procedure.

Example 3. Determination of heater and storage tank size for an apartment building from number of fixtures.

60 lavatories	× 2 =	120 gal per hour
30 bath tubs	× 20 =	600 gal per hour
30 showers	× 75 =	2250 gal per hour
60 kitchen sinks	× 10 =	600 gal per hour
15 laundry tubs	× 20 =	300 gal per hour
Possible maximum demand		= 3870 gal per hour
Probable maximum demand	= 3870 × 0.30 =	1161 gal per hour
Heater or coil capacity		= 1161 gal per hour
Storage tank capacity	= 1161 × 1.25 =	1450 gal.

TABLE 2. HOT WATER DEMAND PER FIXTURE FOR VARIOUS TYPES OF BUILDINGS

Gallons of water per hour per fixture, calculated at a final temperature of 140 F

	Apartment House	Club	Gymnasium	Hospital	Hotel	Industrial Plant	Office Building	Private Residence	School	Y.M. C.A.
1 Basins, private lavatory	2	2	2	2	2	2	2	2	2	2
2 Basins, public lavatory	4	6	8	6	8	12	6		15	8
3 Bathtubs	20	20	30	20	20	30		20		30
4 Dishwashers	15	50–150		50–150	50–200	20–100		15	20–100	20–100
5 Foot basins	3	3	12	3	3	12		3	3	12
6 Kitchen sink	10	20		20	20	20		10	10	20
7 Laundry, stationary tubs	20	28		28	28			20		28
8 Pantry sink	5	10		10	10			5	10	10
9 Showers	75	150	225	75	75	225		75	225	225
10 Slop sink	20	20		20	30	20	15	15	20	20
11 Demand factor	0.30	0.30	0.40	0.25	0.25	0.40	0.30	0.30	0.40	0.40
12 Storage capacity factor[a]	1.25	0.90	1.00	0.60	0.80	1.00	2.00	0.70	1.00	1.00

[a]Ratio of storage tank capacity to probable maximum demand per hour

METHODS OF HEATING WATER

Hot water may be heated either by the direct combustion of fuel, by an intermediate carrier such as steam or hot water, or by electrically heated surfaces. The simplest method is to have the fire on one side of a metal barrier and water on the other. In such a method if the water surfaces of heat transfer are small, and if the water carries a heavy proportion of precipitable salts, the water passages may soon clog and then burn out. A familiar example of such trouble is the water back of the firebox in the kitchen stove or the pipe coil inserted into the firebox of a warm air furnace or small boiler. The critical water temperature at which the lime, magnesia, etc. collect on hot surfaces, varies with the character and proportions of the solids, but generally such deposits are not a serious trouble with water temperatures lower than 140 F.

Coal-burning direct-fired water heaters may be constructed of cored cast-iron sections or of steel. In some cases the external appearance of

the cast-iron sections is the same as in heating boilers, but internally the cores are changed to enable the sections to withstand the city water pressure. In small capacity water heaters, efficiency is not considered so important as low first cost and ability to maintain a fire at a low rate of combustion, and consequently such heaters are generally built with a *dry* section or fire-brick lining at the base of the fire-pot to prevent too much chilling of the fuel. While mud and scale will eventually clog the water ways of any direct-fired heater, increased life may be obtained by providing a three-way cock in the return line between the heater and the

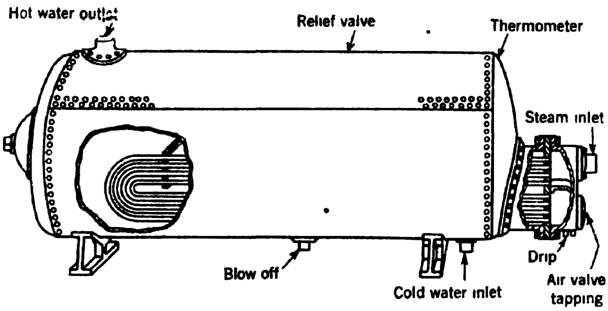

FIG. 2. INDIRECT WATER HEATER

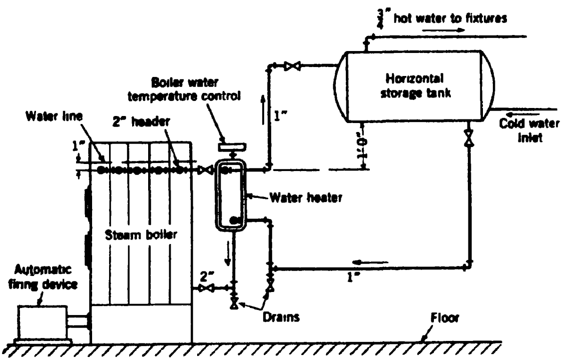

FIG. 3. INDIRECT WATER HEATER MOUNTED ON SIDE OF BOILER

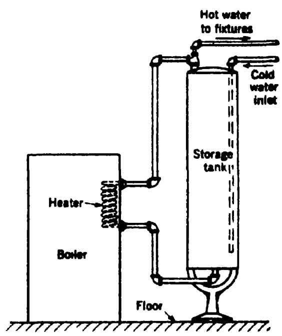

FIG. 4. INDIRECT WATER HEATER PLACED IN BOILER

bottom of the storage tank, so that water can be blown through the heater or the tank separately at full line pressure to clean out loose sediment. Clean-out openings in the bottom of the heater are advantageous if used by operators of water heaters for periodic cleaning out of sediment.

Oil-burning direct-fired water heaters usually are of steel and operate with higher flame temperature and better efficiency then commensurate sized coal-burning heaters. They have the same tendency as coal boilers to *lime up*, and the water passages should be large in cross-section and accessible for periodic cleaning.

Gas-burning direct-fired water heaters may be of the instantaneous or storage type. Instantaneous heaters are generally constructed of spiral water tubes of copper around which the products of combustion circulate

upward from high capacity burners. Storage-type heaters may include in one unit an insulated storage tank, a combustion chamber, flues, burner equipment and controls, or may consist of a separate storage tank and external direct-fired water heater, which may be a so-called *side-arm* heater for small capacity or a gas-fired boiler for larger capacity. Gas boilers used for direct hot water supply must be able to withstand the city water operating pressure. While direct-fired gas heaters are used generally for residences and small installations of 100 gal storage capacity or less, indirect heaters are recommended for larger installations.

In the indirect method either steam or hot water is used for heating the water. With steam the water to be heated is preferably circulated around the outside of the steam tubes which are submerged within a tank. A typical indirect heater using steam is shown in Fig. 2. The coils usually are of copper and are *U* shaped to permit expansion and contraction. The shell may be of steel, copper, or with a special inside protective lining. Where straight heating tubes are used, one end of the tube is usually expanded into a *floating* head to take care of expansion. The coils should be capable of easy withdrawal for inspection and for removal of scale. Instead of steam the heating medium may also be hot water inside the tubes.

Another method of transferring heat from a heating boiler to the domestic water is illustrated in Fig. 3. The water heater is generally a cast-iron shell within which there is located a spiral copper coil. Hot water from the boiler circulates inside the shell and around the coil and returns to the boiler, while domestic water from the storage tank circulates inside the coil. The storage tank should be installed with the bottom of the tank as far above the boiler as possible. Horizontal storage tanks smaller than 18 or 20 in. diameter are not recommended because of the difficulty of preventing the hot and cold water from mixing, and especially is this an important consideration when large quantities of water are withdrawn. In Fig. 4 the heat transfer surface is placed inside the boiler instead of in a separate vessel, but otherwise the operation is similar to that of Fig. 3. This arrangement with vertical tank is commonly used for small domestic installations.

Sometimes the heating element is located inside of the larger type fire tube boilers and small residential boilers. In this case the heat transfer surface is in the form of a number of straight copper tubes with rear *U* bends or a floating head, inserted through a special opening in the boiler. While the coil may be located in the steam space above the water line of a steam boiler, it operates more satisfactorily when below the water line since clogging of the water tubes may thereby be delayed. This method is widely used without storage tanks since the intimate contact and efficient circulation of the water in this arrangement permit the utilization of the heat stored in the water of the boiler. A thermostatic three-way mixing valve is frequently used to maintain a uniform temperature of the hot water supply to the plumbing fixtures.

In order to reduce clogging by precipitated solids, water heating plants sometimes develop steam in a closed circuit, transferring the heat through a tubular heater to the domestic water. The water in the primary heater, exposed to the high temperature of the fire, is repeatedly used and hence has no appreciable tendency to deposit scale, while the domestic water, heated by steam at a much lower temperature than that of the fire, also exhibits a much reduced tendency to precipitate its dissolved salts.

COMPUTING AREA OF HEAT TRANSMITTING SURFACE

The area of the inside surface of a heating coil may be determined from Equation 1.

$$A = \frac{Q \times 8.33\ (t_2 - t_1)}{k_o \times t_m} \tag{1}$$

where

A = surface area of coil, square feet.
Q = quantity of water heated, gallons per hour.
t_2 = hot water outlet temperature, degrees Fahrenheit.
t_1 = cold water inlet temperature, degrees Fahrenheit.
k_o = coefficient of heat transmission, Btu per hour per square foot surface. For copper or brass coils k_o = 240 (steam) and 100 (hot water). For iron coils k_o = 160 (steam) and 67 (hot water).
t_m = logarithmic mean of the difference between the temperature of the heating medium and the average water temperature and is *approximately:*

$$t_s - \left[\frac{(t_2 + t_1)}{2}\right]$$

t_s = temperature of the heating medium, degrees Fahrenheit.

Equation 1 may be used to check the heating coil ratings under temperature conditions differing from those stated in the manufacturer's published ratings.

TABLE 3. COEFFICIENT OF HEAT TRANSFER OF INSTANTANEOUS WATER HEATERS
k = Btu per hr per sq ft per degree Fahrenheit logarithmic mean temperature difference

BOILER WATER TEMPERATURE	k
210	225
200	175
180	150

Example 4. What area of copper transfer surface will be required to heat 70 gal per hour from 40 to 180 F with boiler water at 220 F?

$$t_m = \left[220 - \frac{(180 + 40)}{2}\right] = 110 \qquad A = \frac{70 \times 8.33\ (180 - 40)}{100 \times 110} = 7.39 \text{ sq ft.}$$

For instantaneous submerged heaters the surface required will depend upon (1) the velocity of water in the tubes, (2) the boiler water temperature, (3) the inlet water temperature, (4) the outlet water temperature, (5) the cleanliness of the coil surface, and (6) the condition of the boiler water surrounding the coil. If the heater is located in the water of an actively steaming part of a boiler the heat transfer may be twice as great as would be obtained if the water surrounding the coil were circulating slowly. Ratings of instantaneous water heating coils will therefore vary greatly depending upon the assumptions made regarding the conditions of operation. The values of the coefficient of heat transmission for instantaneous heaters shown in Table 3 are conservative.

For a coil in which heat is transferred from steam to water the value of $k = 300\sqrt{v}$ may be safely used (v = velocity of water in feet per second).

The rate of heat transfer between steam or water as the carrier and the domestic water is influenced by the rate of movement of both the carrier

and the water which receives the heat. For this reason, where the transfer occurs from heating system water to domestic water, it is good practice to install a circulating pump to insure rapid movement of the boiler water.

In view of the high condensation rates obtained when steam is used with gravity circulation from the boiler, as when there is a sudden demand followed by an inflow of cold water, the bottom of a steam heating transfer element always should be at least 30 in. above the boiler water line, and the steam and condensate return pipes should be of liberal size. Otherwise water hammer and reduced capacity may result due to imperfect drainage of condensate.

When connecting a transfer-type hot water heater below the water line of a cast-iron steam boiler having vertical sections, there should be a separate tapping for water circulation into every section of the boiler, as shown in Fig. 3, unless the boiler has large top nipple ports providing inter-sectional circulation. If the top nipples are entirely within the boiler steam space, no internal circulation occurs between sections and steaming may occur in any section not connected to the indirect heater and further the unconnected sections will not deliver any heat to the water heater.

COMPUTING GRATE AREA FOR COAL-FIRED HEATER

The grate area required for a small coal-fired water heater may be calculated by Equation 2.

$$G = \frac{W\,(t_2 - t_1) \times 100}{H \times E \times C} \tag{2}$$

where

G = grate area, square feet
W = weight of water, pounds per hour.
$t_2 - t_1$ = temperature difference between entering and leaving water, degrees Fahrenheit.
H = heating value of coal, Btu per pound.
C = weight of coal burned, pounds per hour per square foot of grate.
E = efficiency, per cent.

In a small heater 4.5 lb is a conservative value for C, and an efficiency of 60 per cent would represent excellent performance.

Example 5. What grate area is required for a coal-burning water heater warming 100 gal per hour of water from 50 to 180 F, when the combustion rate is 4.5 lb per hour per square foot of grate, if the heating value of the fuel is 12,500 Btu per pound, and the efficiency is 60 per cent?

$$\text{Substituting: } G = \frac{100 \times 8.3 \times (180 - 50) \times 100}{12{,}500 \times 60 \times 4.5} = 3.2 \text{ sq ft.}$$

The quantity of gas, oil, or other fuel required per hour for water heating may be calculated by Equation 3.

$$F = \frac{W\,(t_2 - t_1) \times 100}{H \times E} \tag{3}$$

where

F = units of fuel (lb, cu ft, gal, etc).
H = heating value of fuel, Btu per unit.
W = weight of water, pounds per hour.
$t_2 - t_1$ = temperature difference between entering and leaving water, degrees Fahrenheit.
E = efficiency, per cent

Efficiencies for oil and gas may be taken as 75 and 80 per cent respec-

tively. The heating value of the fuel and the temperature rise should be determined to suit local conditions.

CONTROL OF SERVICE WATER TEMPERATURE

Coal-fired boilers are usually controlled by an immersion thermostat located in the heated water, which opens or closes draft dampers at the boiler to adjust the rate of fuel combustion. With oil- or gas-fired boilers the immersion thermostat controls the oil burner motor or the automatic gas valve. The gas pilot flame usually burns continuously. With electric heaters the immersion thermostat operates a switch on the source of energy.

When steam or hot water is the medium for heating the water in the tank, immersion thermostat controls a valve in the steam or hot water supply line. In small residence installations, using water as the carrier, a combined immersion thermostat and butterfly valve all in one simple fitting may be installed in the transmitting circuit to prevent overheating of the service water.

In residences heated by pump circulated hot water, the house temperature is controlled by operating the circulating pump intermittently, while domestic hot water is warmed by transfer from the house boiler, independent of the pump operation. The domestic water is heated from the heating boiler the year 'round. Under such an arrangement, to prevent overheating the house by thermal circulation when the pump is not running, it is usual to insert a weighted check-valve in the house heating main, so that no circulation to the house heating system can occur unless the pump operates. In summer the fire may be controlled to maintain a boiler water temperature lower than when heating and generally about 20 F warmer than that desired in the domestic hot water system.

In buildings which have restaurants it is generally desirable to install two separate service hot water systems so that water at about 180 F minimum may be available for dish washing, while water at 140 F maximum may be used for lavatory and bath purposes.

The immersion thermostat in a hot water storage tank should be located no higher than the center of the tank, and possibly should be even closer to the bottom since water in a tank stratifies proportionally to the temperature. When hot water is removed, the cold water entering to replace it quickly reduces the temperature in the lower parts of the tank.

SOLAR WATER HEATERS

Solar heaters utilize the energy of the sun for heating hot water. The successful operation of such heaters requires the availability of sunshine practically every day in the year, which has limited their use to Florida and the southern portions of California. When supplemented with some other means of gas, coal or oil water heating, solar heaters may be used in climates where sunshine may be more or less intermittent. They have been used in summer homes as far north as Chicago. When properly installed and proportioned, solar water heaters render satisfactory service especially in climates where the outside temperatures are high and extremely hot water is not necessarily desirable. Such installations consist essentially of a storage tank, heating coil and hot box. The coil is installed in the hot box and is arranged to circulate water to and from the storage tank. The advantage in the use of this type of heater is the fact that it requires no fuel. The same materials should be used for the coil,

circulation lines and tank. A copper coil is more efficient in absorbing heat in the box but galvanized iron or steel may be substituted depending on the local water conditions, cost and other considerations.

The storage tank must be able to store sufficient heated water for the night period of about 16 hours when the coil is not functioning or is operating under such poor sun conditions as to make its heating effect negligible. Due to the fact that the *no sun* period includes the night period when little or no hot water is used, an available storage of 50 per cent of the average daily usage is considered adequate. Since about 25 per cent of stored hot water cannot be drawn out of a storage tank before the incoming cold water reduces the temperature of all of the

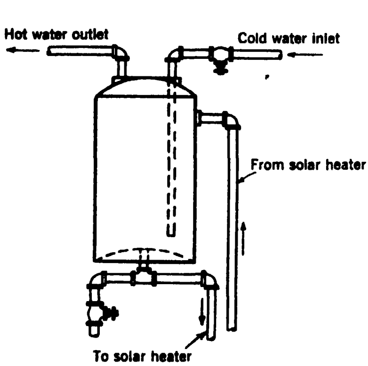

FIG. 5. SOLAR HEATER TANK CONNECTIONS

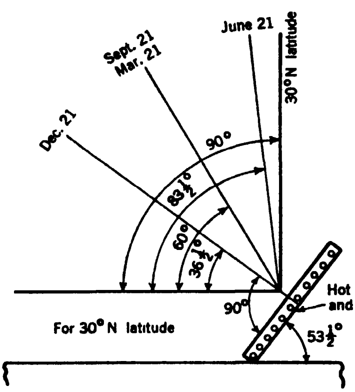

FIG. 6. SOLAR HEATING COIL INCLINATION

water in the tank to an unsatisfactory point for usage, the equation for calculating the storage capacity of the tank becomes:

$$S = \frac{Q \times 0.50}{0.75} = 0.666Q \qquad (4)$$

where

S = storage capacity of tank, gallons.
Q = average daily usage, gallons.

Thus for a family of four persons using an average of 40 gal of hot water per person per day the size of the tank would be 4 persons x 40 gal x 0.666 or 106 gal, and the nearest standard size of tank to this theoretical capacity would be used. The tank should be well insulated to prevent undue loss of heat during the 16-hour period when the coil is inoperative, and it should be located as high as possible in the building (under the peak of the roof if such exists) so as to secure a maximum circulation head from the coil. The hot water supply to the house, as shown in Fig. 5, is located at the top of the tank, which serves to air vent the tank by blowing air out through the hot water faucets in small bubbles as fast as it accumulates.

The coil should be of the return-bend type, square or slightly rectangular in form, and should have the pipes running east and west, with the coil on the south side of the building where it can receive the full sun effect all day long without shadows from the building itself or from adjacent obstructions such as trees or other structures. The coil should

be placed as low as possible in relation to the storage tank level, such as on a porch roof, the roof of a one-story extension or, if necessary, even on the ground. Both the coil and the circulation lines should be designed to facilitate the circulation flow as much as possible using long radius copper fittings or recessed galvanized iron fittings to match the materials of the coil, circulation lines and tank. The coil should be inclined as shown in Fig. 6 so that the north end is raised above the south end to secure an angle with the horizontal of about 53 deg. This will result in the inlet end of the coil being on the south side (or bottom) and the outlet end being on the north side (or top). This will satisfy conditions along the 30 deg N latitude which includes the portions of Florida and Southern California where these heaters are most frequently used.

The hot box is usually constructed of wood on the four sides and

TABLE 4. SUGGESTED SOLAR HEATER DESIGN DATA[a]

Design Item	Based on Rate of 30 Gal per Day per Person								Based on Rate of 40 Gal per Day per Person							
No. of Occupants in Residence	1	2	3	4	5	6	7	8	1	2	3	4	5	6	7	8
Hot Water Used at Night, gal per person	15	15	15	15	15	15	15	15	20	20	20	20	20	20	20	20
Hot Water Used at Night, gal total	15	30	45	60	75	90	105	120	20	40	60	80	100	120	140	160
Retained in Tank, 25 per cent, gal	4	8	11	15	19	23	27	30	5	10	15	20	25	30	35	40
Tank Capacity Required, gal	20	40	59	75	94	113	130	150	25	50	75	100	125	150	175	200
Hot Water Used During Day, gal	15	30	45	60	75	90	105	120	20	40	60	80	100	120	140	160
Total Water to be Heated:																
Gal per 8 hr period	35	70	104	135	169	203	235	270	45	90	135	180	225	270	315	360
Gal per hour	4.5	9	13	17	21	26	29	34	6	12	17	23	28	34	39	45
Copper Coil Required:																
Surface area, sq ft	25	50	75	100	121	145	168	192	32	64	96	128	160	192	224	256
Equivalent length 1 in coil, ft	100	200	300	400	484	580	664	768	128	256	384	512	640	768	896	1024
Box Size:																
Area, sq ft	25	50	75	100	121	145	168	192	32	64	96	128	160	192	224	256
Width, ft	4	6	7	8	9	10	10	11	4	6	8	9	10	11	12	12
Length, ft	6	8	11	12 5	13.5	14 5	16.5	17.5	8	10	12	14	16	18	19	21

[a]Sun Effect and the Design of Solar Heaters, by H L. Alt (A S.H V.E. TRANSACTIONS, Vol. 41, 1935, p. 131).

bottom, and is insulated. Over the top of the box glass sash are placed and the box should be constructed as near air-tight as possible. The interior surfaces should be painted white to reflect the heat while the coil should be painted black to absorb the heat. The box need not be deeper than necessary to house the coil and to protect it from the weather.

The addition of a light gage copper plate on the bottom of the box to which the pipe of the coil is soldered, for good metallic contact, will add to the amount of heat received by the coil due to the fact that this plate will receive all of the sun rays which fail to directly strike the coil. The heat from this source is transmitted to the coil through the plate instead of by heating the air surrounding the coil and from which only part of the heat enters the coil, the balance being transmitted through the glass sash.

Design data given in Table 4 may be used with some judgment in selecting the size of solar heater coil and box for a particular application. These data are based on consumptions of 30 and 40 gal of hot water per day per person.

FRICTION LOSS IN WATER SUPPLY PIPING

Fig. 7, showing friction loss in *fairly rough* pipe, has been reproduced from Fig. 4 of Building Materials and Structures Report BMS 79, Water Distributing Systems for Buildings, by Roy B. Hunter, *National Bureau of Standards*. For friction loss in tubing, as well as in smooth

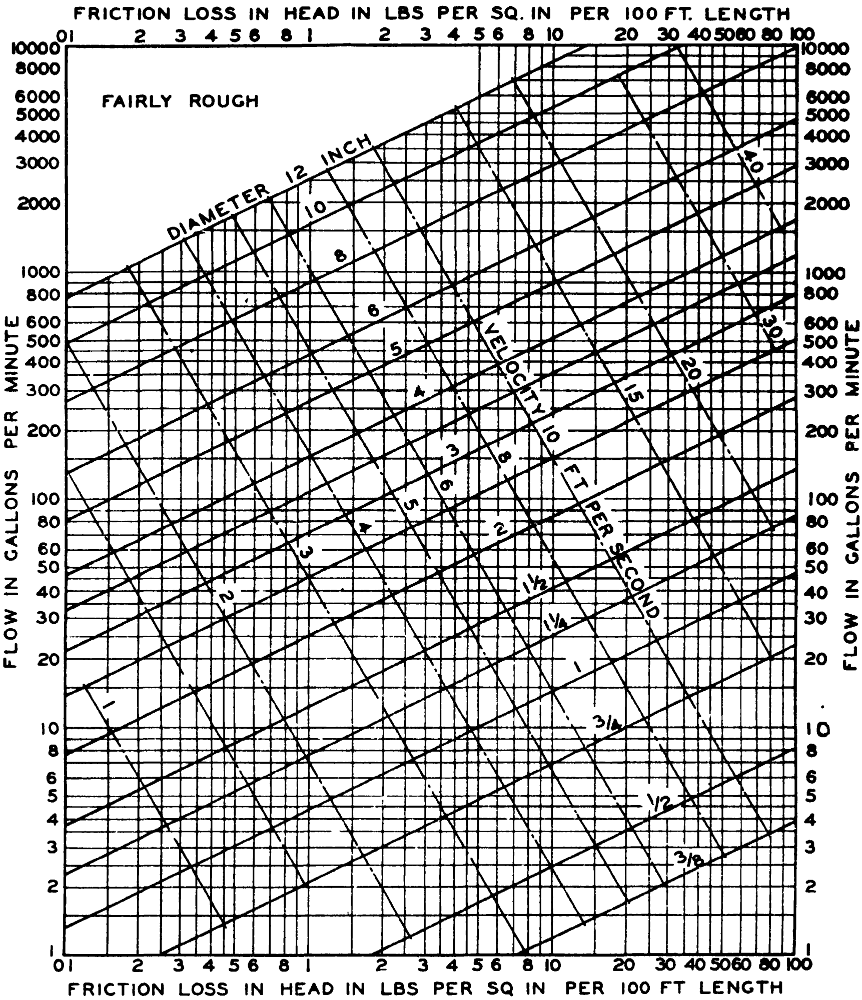

FIG. 7. FRICTION LOSS IN FAIRLY ROUGH PIPE

and rough pipe the reader is referred to Figs. 2, 3 and 5 in the same publication.

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CHAPTER 47

Terminology

Glossary of Physical and Heating, Ventilating, Refrigerating and Air Conditioning Terms Used in the Text

Absolute Humidity: *See Humidity.*

Absolute Pressure: The pressure referred to that of a perfect vacuum. It is the sum of gage pressure and barometric pressure.

Absolute Temperature: A reading on the absolute temperature scale. Absolute temperature is obtained by adding 459.70 degrees to the Fahrenheit temperature.

Absolute Zero: The zero point on the absolute scale 459.70 F below the zero of the Fahrenheit scale.

Acceleration: The rate of change of velocity. In the fps system this is expressed in units of one foot per second. $a = V \div t$.

Acceleration Due to Gravity: The rate of gain in velocity of a freely falling body, the value of which varies with latitude and elevation. The international gravity standard has the value of 980.665 cm per second per second or 32.174 ft per second per second, which is the actual value of this acceleration at sea level and about 45 deg latitude.

Adiabatic: An adjective descriptive of a process in which no heat is added to or extracted from the system executing the process.

Air Cleaner: A device designed for the purpose of removing air-borne impurities such as dusts, fumes and smokes. (Air cleaners include air washers and air filters.)

Air Conditioning: The simultaneous control of all or at least the first three of those factors affecting both the physical and chemical conditions of the atmosphere within any structure. These factors include temperature, humidity, motion, distribution, dust, bacteria, odors and toxic gases, most of which affect in greater or lesser degree human health or comfort. (See *Comfort Air Conditioning.*)

Air Washer: An enclosure in which air is forced through a spray of water in order to cleanse, humidify, or dehumidify the air.

Anemometer: An instrument for measuring the velocity of moving air.

Atmospheric Pressure: The pressure indicated by a barometer. *Standard atmospheric pressure* is a pressure of 76 cm mercury (density 13.5951 grams per cubic centimeter, gravity 980.665 cm per second per second). It is equivalent to 14 696 lb per square inch or 29.921 in. of mercury at 32 F.

Baffle: A plate or wall for deflecting gases or fluids.

Blast Heater: A set of heat transfer coils or sections used to heat air which is drawn or forced through it by a fan.

Boiler Heating Surface: That portion of the surface of the heat-transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other, in which the fluid being heated forms part of the circulating system; this surface shall be measured on the side receiving heat. This includes the boiler, water walls, water screens, and water floor. (*A.S.M.E.* Power Test Codes, Series 1929.)

Boiler Horsepower: The equivalent evaporation of 34.5 lb of water per hour from and at 212 F. This is equal to a heat output of $970.3 \times 34.5 = 33{,}475$ Btu per hour.

British Thermal Unit: A unit of energy defined in terms of the international steam-table calorie through the convenient relation 1 Btu per pound per degree Fahrenheit = 1 cal per gram per degree Centigrade. It is approximately the quantity of heat required to raise the temperature of 1 lb of liquid water from 63 to 64 F.

By-pass: A pipe or duct, usually controlled by valve or damper, for conveying a fluid around a main control valve or damper.

Calorie: (Large calorie or I. T. kilocalorie) is equal to 1000 international steam-table calories = 1/860 international kilowatthour. For practical purposes it may be con-

sidered as 1/100 of the heat required to raise the temperature of 1 kilogram of water from 0 to 100 C.

Central Fan System: A mechanical indirect system of heating, ventilating, or air conditioning, in which the air is treated or handled by equipment located outside the rooms served, usually at a central location, and is conveyed to and from the rooms by means of a fan and a system of distributing ducts. (See Chapter 20.)

Chimney Effect: The tendency in a duct or other vertical air passage for air to rise when heated, owing to its decrease in density.

Coefficient of Heat Transmission: The amount of heat (Btu) transmitted *from air to air* in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 F *between the air on the inside and that on the outside of the wall, floor, roof or ceiling.*

Comfort Air Conditioning: The process by which simultaneously the temperature, moisture content, movement and quality of the air in enclosed spaces intended for human occupancy may be maintained within required limits. (See *Air Conditioning.*)

Comfort Line: The effective temperature at which the largest percentage of adults feels comfortable.

Comfort Zone (*Average*): The range of effective temperatures over which the majority (50 per cent or more) of adults feel comfortable. *Comfort Zone* (*Extreme*): The range of effective temperatures over which one or more adults feel comfortable. (See Chapter 2.)

Concealed Radiator: A heating device located within, adjacent to, or exterior to the room being heated but so covered or enclosed or concealed that the heat transfer surface of the device, which may be either a radiator or a convector, is not visible from the room. Such a device transfers its heat to the room largely by convection air currents.

Conductance: The amount of heat (Btu) transmitted from surface to surface in one hour through one square foot of a material or construction, whatever its thickness, when the temperature difference is 1 F between the two surfaces.

Conduction: The transmission of heat through and by means of matter unaccompanied by any obvious motion of the matter.

Conductivity: The amount of heat (Btu) transmitted in one hour through one square foot of a homogeneous material *1 in. thick* (*or 1 ft*) for a difference in temperature of 1 F between the two surfaces of the material.

Conductor (*Heat*): A material capable of readily conducting heat. The opposite of an insulator or insulation.

Constant Relative Humidity Line: Any line on the psychrometric chart representing a series of conditions which may be evaluated by one percentage of relative humidity; there are also *constant* dry-bulb lines, wet-bulb lines, effective temperature lines, vapor pressure lines, and lines showing other physical properties of air mixed with water vapor.

Convection: The transmission of heat by the circulation of a liquid or a gas such as air. Convection may be *natural* or *forced.*

Convector: A heat transfer surface designed to transfer its heat to surrounding air largely or wholly by convection. Such a surface may or may not be enclosed or concealed. When concealed and enclosed the resulting device is sometimes referred to as a concealed radiator. (See also definition of *Radiator.*) (See also Chapter 13.)

Decibel: A unit commonly used for expressing sound or noise intensities referred to an arbitrary reference level. It is defined by the relation $db = 10 \log_{10} \frac{P_1}{P_0}$, where P_1 is the unknown intensity, and P_0 is the reference level which is commonly taken as 10^{-16} watts per square centimeter.

Degree-Day: A unit, based upon temperature difference and time, used in specifying the nominal heating load in winter. For any one day there exists as many degree-days as there are degrees Fahrenheit difference in temperature between the mean temperature for the day and 65 F.

Degree of Saturation or Per Cent Saturation: The ratio of actual humidity ratio W to the saturation humidity ratio W_s corresponding to the actual temperature and the

observed pressure. $\mu = \frac{W}{W_s}$ (Approximately the same as but not identical with *relative humidity*. See Chapter 1).

Dehumidify: To reduce, by any process, the density of water vapor within a given space.

Dehydrate: To separate or to remove water in all forms from matter. Liquid water, hygroscopic water, and water of crystallization or water of hydration are included.

Density: The weight of a unit volume, expressed in pounds per cubic foot. $d = W \div V$.

Dew-Point Temperature: The temperature corresponding to saturation (100 per cent relative humidity) for a given moisture content.

Direct-Indirect Heating Unit: A heating unit located in the room or space to be heated and partially enclosed, the enclosed portion being used to heat air which enters from outside the room.

Direct Radiator: Same as *Radiator*.

Direct-Return System (*Hot Water*): A hot water system in which the water, after it has passed through a heating unit, is returned to the boiler along a direct path so that the total distance traveled by the water is the shortest feasible, and so that there are considerable differences in the lengths of the several circuits composing the system.

Down-Feed One-Pipe Riser (*Steam*): A pipe which carries steam downward to the heating units and into which the condensation from the heating units drains.

Down-Feed System (*Steam*): A steam heating system in which the supply mains are above the level of the heating units which they serve.

Draft Head (*Side Outlet Enclosure*): The height of a gravity convector between the bottom of the heating unit and the bottom of the air outlet opening. (*Top Outlet Enclosure*): The height of a gravity convector between the bottom of the heating unit and the top of the enclosure.

Drip: A pipe, or a steam trap and a pipe, considered as a unit, which conducts condensation from the steam side of a piping system to the water or return side of the system.

Dry: (*a*) To separate or to remove liquid water or water vapor from matter. (*b*) To separate or to remove any readily condensible vapor from other matter

Dry Air: In psychrometric work, *dry air* is defined as air without water vapor. This state, though not obtained practically, is used as the basis of calculations.

Dry-Bulb Temperature: The temperature indicated by a standardized thermometer after correction for radiation, etc.

Dry Return: A return pipe in a steam heating system which carries both water of condensation and air. The dry return is above the level of the water line in the boiler in a gravity system. (See *Wet Return*.)

Dust: Solid material in a finely divided state, the particles of which are large and heavy enough to fall with increasing velocity, due to gravity in still air. For instance, particles of fine sand or grit, the average diameter of which is approximately 0.01 centimeter, such as are blown on a windy day, may be called dust.

Dynamic Head or Pressure: Same as *Total Pressure*.

Effective Temperature: An arbitrary index which combines into a single value the effect of temperature, humidity, and movement of air on the degree of warmth or cold felt by the human body. The numerical value is that of the temperature of still, saturated air which would induce an identical sensation of warmth.

Enthalpy: Enthalpy was formerly called *heat content*, sometimes *total heat*. It is a thermodynamic property which serves as a measure of the quantity of thermal energy *convected* by a fluid in steady flow. In a non-flow process the increase of enthalpy equals the quantity of heat absorbed *provided pressure is constant*. Specific enthalpy is the ratio of total enthalpy to total weight, that is, enthalpy per unit weight of substance, Btu per pound.

Entropy: Entropy is the ratio of the heat added to a substance to the absolute temperature at which it is added. It is a thermodynamic property which, for practical

purposes, is best defined by stating its principal functions: (1) during a reversible *adiabatic* change of state, entropy is constant; (2) during a reversible *isothermal* change of state, the heat absorbed is equal to absolute temperature times change of entropy. *Specific entropy* is the ratio of total entropy to total weight, that is, entropy per unit weight, Btu per degree Fahrenheit per pound.

Equivalent Evaporation: The amount of water a boiler would evaporate, in pounds per hour, if it received feed water at 212 F and vaporized it at the same temperature and atmospheric pressure.

Estimated Design Load: The sum of the heat emission of the equivalent direct radiation to be installed plus the allowance for heat loss of the connecting piping plus the heat requirements of any auxiliary apparatus connected with the system.

Estimated Maximum Load: The load stated in Btu per hour or equivalent direct radiation that has been estimated to be the greatest or maximum load that the boiler will be called upon to carry. (See Chapter 12).

Extended Heating Surface: See *Heating Surface.*

Extended Surface Heating Unit: A heating unit having a relatively large amount of extended surface which may be integral with the core containing the heating medium or assembled over such a core, making good thermal contact by pressure or by being soldered to the core or by both pressure and soldering. An extended surface heating unit is usually placed within an enclosure and therefore functions as a convector.

Fan Furnace System: See *Warm Air Heating System*

Force: The action on a body which tends to change its relative condition as to rest or motion. $F = (WV) \div (gt)$.

Free Enthalpy: A thermodynamic property which serves as a measure of the available energy of a system with respect to surroundings at the same temperature and same pressure as that of the system. No process involving an increase in available energy can occur spontaneously. (See example on *Free Enthalpy* in Chapter 1.)

Fumes: Particles of solid matter resulting from such chemical processes as combustion, explosion, and distillation, ranging from 0.1 to 1.0 micron in size.

Furnace: That part of a boiler or warm air heating plant in which combustion takes place. Also a complete heating unit for transferring heat from fuel being burned to the air supplied to a heating system

Furnace Volume (*Total*)*:* The total furnace volume for horizontal-return tubular boilers and water-tube boilers is the cubical contents of the furnace between the grate and the first plane of entry into or between tubes. It therefore includes the volume behind the bridge wall as in ordinary horizontal-return tubular boiler settings, unless manifestly ineffective (*i e.*, no gas flow taking place through it), as in the case of waste-heat boilers with auxiliary coal furnaces, where one part of the furnace is out of action when the other is being used. For Scotch or other internally fired boilers it is the cubical contents of the furnace, flues and combustion chamber, up to the plane of first entry into the tubes. (*A.S.M.E.* Power Test Codes, Series 1929.)

Gage Pressure: Pressure measured from atmospheric pressure as a base. Gage pressure may be indicated by a manometer which has one leg connected to the pressure source and the other exposed to atmospheric pressure.

Grate Area: The area of the grate surface, measured in square feet, to be used in estimating the rate of burning fuel. This area is construed to mean the area measured in the plane of the top surface of the grate, except that with special furnaces, such as those having magazine feed, or special shapes, the grate area shall be the mean area of the active part of the fuel bed taken perpendicular to the path of the gases through it. For furnaces having a secondary grate, such as those in double-grate down draft boilers, the effective area shall be taken as the area of the upper grate plus one-eighth of the area of the lower grate, both areas being estimated as previously defined.

Gravity Warm Air Heating System: See *Warm Air Heating System.*

Heat: That form of energy which flows from one body to another by virtue of a difference in temperature between the bodies.

Heating Medium: A substance such as water, steam, air, or furnace gas used to convey heat from the boiler, furnace or other source of heat or energy to the heating unit from which the heat is dissipated.

Heating Surface: The exterior surface of a heating unit. *Extended heating surface (or extended surface):* Heating surface consisting of fins, pins or ribs which receive heat by conduction from the prime surface. *Prime Surface:* Heating surface having the heating medium on one side and air (or extended surface) on the other. (See also *Boiler Heating Surface.*)

Heat of the Liquid: This can usually be interpreted as the specific enthalpy of saturated liquid.

Hot Water Heating System: A heating system in which water is used as the medium by which heat is carried through pipes from the boiler to the heating units.

Humid Heat: Ratio of increase of enthalpy per pound of dry air to rise of temperature under conditions of constant pressure and constant humidity ratio.

Humidify: To increase, by any process, the density of water vapor within a given space.

Humidistat: A regulatory device, actuated by changes in humidity, used for the automatic control of relative humidity.

Humidity: Water vapor when mixed with dry air or other diluent gases. *Absolute humidity* is the weight of water vapor per unit volume of moist air, pounds per cubic foot. It can be calculated by dividing the humidity ratio, weight of water vapor per pound of dry air, by the volume of the mixture per pound of dry air. *Relative humidity* is the ratio of the partial pressure of the water vapor in the air to the saturation pressure of pure water corresponding to the actual temperature. (See Chapter 1.)

Humidity Ratio: Weight of water vapor per pound of dry air. (Formerly called specific humidity.)

Hygrostat: Same as *Humidistat.*

Inch of Water: The pressure due to a column of liquid water one inch high at a temperature of 60 F.

Insulation (*Heat*)*:* A material having a relatively high heat-resistance per unit of thickness.

Isobaric: An adjective used to indicate a change taking place at constant pressure.

Isothermal: An adjective used to indicate a change taking place at constant temperature.

Latent Heat: Latent heat is the heat required to produce a change of state at constant temperature. The most general interpretation is heat absorbed at constant temperature. More specifically the *latent heat of vaporization* is the difference between the specific enthalpies of saturated vapor and saturated liquid at the same temperature (and, for a pure substance, the same pressure). *Latent heat of sublimation* is the difference between the specific enthalpies of saturated vapor and saturated solid at the same temperature. *Latent heat of fusion* is the difference between the specific enthalpies of saturated liquid and saturated solid at the same temperature.

Laws of Thermodynamics: The Law of Conservation of Energy states that energy, in any of its forms, can neither be created nor destroyed. As a corollary to this, the *First Law* of Thermodynamics states that in any power cycle or refrigeration cycle the net heat absorbed by the working substance is exactly equal to the net work done. The *Second Law* of Thermodynamics states that a power cycle which absorbs heat at a single temperature and converts it wholly into work, as required by the First Law, is impossible; hence it is absolutely necessary to reject heat at some lower temperature if any work is to be done. The Second Law further prescribes the least possible quantity of heat that must be so rejected depending on the two temperatures involved.

Manometer: An instrument for measuring pressures; essentially a U-tube partially filled with a liquid, usually water, mercury, or a light oil, so the amount of displacement of the liquid indicates the pressure being exerted on the instrument.

Mass: The quantity of matter, in pounds, to which the unit of force (one pound) will give an acceleration of one foot per second per second. $m = W \div g$.

Mb, Mbh: Symbols which represent, respectively, 1000 Btu and 1000 Btu per hour.

Mechanical Equivalent of Heat: The conversion factor from Btu to foot pounds; $J = 778.3$ foot pounds per Btu. This is also referred to as Joule's Equivalent.

Micron: A unit of length, the thousandth part of one millimeter or the millionth of a meter.

Mol (*Pound Mol*)*:* A weight in pounds numerically equal to the molecular weight of a substance. In the case of gases, and at not too high pressures, the volume of 1 mol is approximately the same for any gas at the same temperature and pressure. At 32 F and standard atmospheric pressure this volume is 358.65 cu ft.

One-Pipe Supply Riser (*Steam*)*:* A pipe which carries steam upward to a heating unit and which also carries the condensation from the heating unit in a direction opposite to the steam flow.

One-Pipe System (*Hot Water*)*:* A hot water system in which the cooled water from .he heating units is returned to the supply main. Consequently, the heating units 'arthest from the boiler are supplied with cooler water than those near the boiler in the ıame circuit.

One-Pipe System (*Steam*)*:* A steam heating system consisting of a main circuit in vhich the steam and condensate flow in the same pipe, usually in opposite directions.)rdinarily to each heating unit there is but one connection which must serve as both he supply and the return, although separate supply and return connections may be used.

Overhead System: Any steam or hot water system in which the supply main is above he heating units. With a steam system the return must be below the heating units; vith a water system, the return *may* be above the heating units.

Panel Radiator: A heating unit placed on or flush with a flat wall surface and inended to function essentially as a radiator.

Plenum Chamber: An air compartment maintained under pressure and connected to ıne or more distributing ducts.

Potentiometer: An instrument for measuring or comparing small electromotive forces.

Power: The rate of performing work; usually expressed in units of horsepower, Btu ıer hour, or watts.

Prime Surface: See *Heating Surface.*

Psychrometer: An instrument for ascertaining the humidity or hygrometric state of he atmosphere. *Psychrometric:* Pertaining to psychrometry or the state of the atmophere as to moisture. *Psychrometry:* The branch of physics that treats of the measureıent of degree of moisture, especially the moisture mixed with the air.

Pyrometer: An instrument for measuring high temperatures.

Radiant Heating (Panel Heating): A method of heating involving the installation of ıe heating units (pipe coils) within the wall, floor or ceiling of the room, so that the eating process takes place mainly by radiation from the wall, floor or ceiling surfaces ɔ the objects in the room. As used in Chapter 45, *panel heating* refers to a system ·ansmitting heat by both radiation and convection to both air and surrounding surıces, while *radiant heating* refers to a system dependent on radiation alone in which ıse only the heat radiated from the panel is effective for heating purpose.

Radiation: The transmission of heat through space by wave motion.

Radiator: A heating unit exposed to view within the room or space to be heated. A ıdiator transfers heat by radiation to objects within visible range and by conduction › the surrounding air which in turn is circulated by natural convection; a so-called *ıdiator* is also a *convector* but the single term *radiator* has been established by long usage.

Recessed Radiator: A heating unit set back into a wall recess but not enclosed.

Refrigerant: A substance which produces a refrigerating effect by its absorption of ɛat while expanding or vaporizing.

Relative Humidity: See *Humidity;* also discussion relative humidity, Chapter 1.

Return Mains: The pipes which return the heating medium from the heating units to ıe source of heat supply.

Reversed-Return System (*Hot Water*)*:* A hot water heating system in which the water om several heating units is returned along paths arranged so that all circuits composing ıe system or composing a major sub-division of the system are practically of equal ngth.

Saturated Air: A mixture of dry air and saturated water vapor, all at the same dry-bulb temperature. It may also be considered as air containing the maximum possible amount of water vapor at a given temperature without becoming supersaturated.

Saturation: The condition for coexistence in stable equilibrium of two or more distinct phases, such as steam over the water from which it is being generated.

Saturation Pressure: The saturation pressure for a pure substance for any given temperature is that pressure at which vapor and liquid or vapor and solid can coexist in stable equilibrium.

Sensible Heat: Heat which manifests itself by temperature change.

Smoke: Carbon or soot particles less than 0.1 micron in size which result from the incomplete combustion of carbonaceous materials such as coal, oil, tar, and tobacco.

Smokeless Arch: An inverted baffle placed in an up-draft furnace toward the rear to aid in mixing the gases of combustion and thereby to reduce the smoke produced.

Specific Enthalpy: The ratio of total enthalpy to total weight. The specific enthalpy of air is its enthalpy, Btu per pound, measured above 0 F and 29.921 in. Hg as a reference point. The specific enthalpy of water is its enthalpy, Btu per pound, measured from the reference point of saturated liquid at 32 F. (See *Enthalpy*.)

Specific Gravity: The ratio of the weight of a body to the weight of an equal volume of water at some standard temperature, usually 39.2 F.

Specific Heat: The ratio of heat absorbed per unit weight of substance to temperature rise. For gases, both specific heat at constant pressure, c_p, and specific heat at constant volume, c_v, are frequently given. In air conditioning, c_p is usually used.

Specific Volume: The volume, expressed in cubic feet, of one pound of a substance. $v = 1 \div d = V \div W$.

Split System: A system in which the heating and ventilating are accomplished by means of radiators or convectors supplemented by mechanical circulation of air (heated or unheated) from a central point.

Square Foot of Heating Surface (*Equivalent*): Equivalent Direct Radiation (EDR). That amount of heating surface which will give off 240 Btu per hour. The *equivalent* square feet of heating surface may have no direct relation to the actual surface area.

Stack Height: The height of a gravity convector between the bottom of the heating unit and the top of the outlet opening.

Standard Air: Air weighing 0.075 lb per cubic foot. (The density of air at 29.921 in. Hg barometric pressure, 68 F dry-bulb and 50 per cent relative humidity is 0.07497; and dry air at 70 F dry-bulb is 0.07496.)

Static Pressure: The normal force per unit area that would be exerted by a moving fluid on a small body immersed in it if the body were carried along with the fluid. Practically, it is the normal force per unit area at a small hole in a wall of the duct through which the fluid flows (piezometer) or on the surface of a stationary tube at a point where the disturbances created by inserting the tube cancel. It is supposed that the thermodynamic properties of a moving fluid depend on static pressure in exactly the same manner as those of the same fluid at rest depend upon its uniform hydrostatic pressure.

Steam: Water in the vapor phase. *Dry Saturated Steam* is steam at the saturation temperature corresponding to the pressure, and containing no water in suspension. *Wet Saturated Steam* is steam at the saturation temperature corresponding to the pressure, and containing water particles in suspension. *Superheated Steam* is steam at a temperature higher than the saturation temperature corresponding to the pressure.

Steam Heating System: A heating system in which heat is transferred from the boiler or other source of steam to the heating units by means of steam at, above, or below atmospheric pressure.

Steam Trap: A device for allowing the passage of condensate and preventing the passage of steam, or for allowing the passage of air as well as condensate.

Superheated Steam: See *Steam*.

Supply Mains (*Steam*): The pipes through which the steam flows from the boiler or source of supply to the run-outs and risers leading to the heating units.

Surface Conductance: The amount of heat (Btu) transmitted by radiation, conduction, and convection *from a surface to the air or liquid surrounding it,* or vice versa, in one hour per square foot of surface for a difference in temperature of 1 deg between the surface and the surrounding air or liquid.

Therm: 100,000 Btu. (Used in the gas industry.)

Thermal Resistance: The reciprocal of *conductance.*

Thermal Resistivity: The reciprocal of *conductivity.*

Thermostat: An instrument which responds to changes in temperature and which directly or indirectly controls the source of heat supply.

Ton of Refrigeration: The removal of 12,000 Btu of heat per hour at a low temperature.

Ton Day of Refrigeration: The removal of 288,000 Btu of heat at a low temperature.

Total Heat: This can usually be interpreted as increase of enthalpy at constant pressure. It is often regarded as synonymous with enthalpy.

Total Pressure: In the theory of the flow of fluids; the sum of the static pressure and the velocity pressure at the point of measurement.

Tube (or Tubular) Radiator: A cast-iron heating unit used as a radiator and having small vertical tubes.

Two-Pipe System (*Steam or Water*): A heating system in which one pipe is used for the supply of the heating medium to the heating unit and another for the return of the heating medium to the source of heat supply. The essential feature of a two-pipe system is that each heating unit receives a direct supply of the heating medium which medium cannot have served a preceding heating unit.

Underfeed Distribution System (*Hot Water*): A hot water heating system in which the main flow pipe is below the heating unit.

Underfeed Stoker: A stoker which feeds the coal underneath the fuel bed.

Unit: As applied to heating, ventilating and air conditioning equipment this word means factory-built and assembled equipment with apparatus for accomplishing some specified function or combination of functions. (See Chapters 21 and 22.)

It is loosely applied to a great variety of equipment. Usually the function is included in the name, and hence come terms like Unit Heater, Unit Ventilator, Humidifying Unit, and Air Conditioning Unit.

Units are said to be *direct* or *room*, when intended for location, or located in, the treated space; *indirect* or *remote*, when outside or adjacent to the treated space. They are *ceiling* units when suspended from above, and *floor* when supported from below. Other descriptive words include *free delivery* when the unit is not intended to be attached to ducts or similar resistance-producing devices, and *pressure* when for use with such ducts. Complete description requires the use of several of these qualifying words or phrases. (See Chapter 22.)

Up-Feed System (*Steam*): A steam heating system in which the supply mains are below the level of the heating units which they serve.

Vacuum Heating System: A two-pipe steam heating system equipped with the necessary accessory apparatus which will permit operating the system below atmospheric pressure when desired.

Vapor: Any substance in the gaseous state.

Vapor Heating System: A steam heating system which operates under pressures at or near atmospheric and which returns the condensation to the boiler or receiver by gravity. Vapor systems have thermostatic traps or other means of resistance on the return ends of the heating units for preventing steam from entering the return mains; they also have a pressure-equalizing and air-eliminating device at the end of the dry return. *Direct Vent Vapor System:* A vapor heating system with air valves which do not permit re-entry of air.

Vapor Pressure: Synonymous with saturation pressure in the case of a pure substance.

Velocity: The time rate of motion of a body in a fixed direction. In the fps system it is expressed in units of one foot per second. $V = \frac{s}{t}$.

Velocity Pressure: The difference due to velocity between total pressure and static pressure. It is supposed to equal the kinetic energy per unit volume of the fluid at the point of measurement.

Ventilation: The process of supplying or removing air by natural or mechanical means, to or from any space. Such air may or may not have been conditioned. (See *Air Conditioning.*)

Warm Air Heating System: A warm air heating plant consists of a heating unit (fuel-burning furnace) enclosed in a casing, from which the heated air is distributed to the various rooms of the building through ducts. If the motive head producing flow depends on the difference in weight between the heated air leaving the casing and the cooler air entering the bottom of the casing, it is termed a *gravity* system. A booster fan may, however, be used in conjunction with a gravity-designed system. If a fan is used to produce circulation and the system is designed especially for fan circulation, it is termed a *fan furnace* system or a *central fan furnace* system. A fan furnace system may include air washers and filters.

Wet-Bulb Temperature: *Thermodynamic wet-bulb temperature* is the temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature. *Wet-bulb temperature* (without qualification) is the temperature indicated by a wet-bulb psychrometer constructed and used according to specifications. (*A S.M.E.* Power Test Codes, Series 1932, Instruments and Apparatus, Part 18.)

Wet Return: That part of a return main of a steam heating system which is filled with water of condensation. The wet return usually is below the level of the water line in the boiler, although not necessarily so. (See *Dry Return.*)

CHAPTER 48

Abbreviations, Symbols, Standards

Standard Abbreviations; Standard Symbols; Conversion Equations; Graphical Symbols for Piping, Ductwork, Heating and Ventilating, Refrigerating; Specific Heat Table; State Codes or Standards

THIS chapter contains information regarding abbreviations, symbols, conversion equations, and State codes, which are of particular interest to the engineer engaged in heating, ventilating, and air conditioning.

ABBREVIATIONS

Abbreviations are shortened forms of names and expressions employed in texts and tabulations and should not generally be used as symbols in equations. Most of the following abbreviations have been compiled from a list of approved standards[1]. In general the period has been omitted in all abbreviations except where the omission results in the formation of an English word.

Absolute	abs
Air horsepower	air hp
Alternating-current (as adjective)	a-c
Ampere	amp
Ampere-hour	amp-hr
Atmosphere	atm
Average	avg
Avoirdupois	avdp
Barometer	bar.
Boiling point	bp
Brake horsepower	bhp
Brake horsepower-hour	bhp-hr
British thermal unit	Btu
British thermal units per hour	Btuh
Calorie	cal
Centigram	cg
Centimeter	cm
Centimeter-gram-second (system)	cgs
Cubic	cu
Cubic foot	cu ft
Cubic feet per minute	cfm
Cubic feet per second	cfs
Decibel	db
Degree[2]	deg or °
Degree Centigrade	C
Degree Fahrenheit	F
Degree Kelvin	K
Degree Réaumur	R
Diameter	diam
Direct-current (as adjective)	d-c
Feet per minute	fpm
Feet per second	fps
Foot	ft
Foot-pound	ft-lb
Foot-pound-second (system)	fps
Freezing point	fp

[1]Abbreviations for Scientific and Engineering Terms, Z10 1-1941 (*American Standards Association*).

[2]It is recommended that the abbreviation for the temperature scale, F, C, K, be included in expressions for numerical temperatures but, wherever feasible, the abbreviations for *degree* be omitted; as 68 F.

Gallon	gal
Gallons per minute	gpm
Gallons per second	gps
Gram	g
Gram-calorie	g-cal
Horsepower	hp
Horsepower-hour	hp-hr
Hour	hr
Inch	in.
Inch-pound	in.-lb
Indicated horsepower	ihp
Indicated horsepower-hour	ihp-hr
Kilogram	kg
Kilowatt	kw
Kilowatthour	kwhr
Mass	mass
Melting point	mp
Meter	m
Micron	μ (mu)
Miles per hour	mph
Millimeter	mm
Minute	min
Molecular weight	mol. wt
Mol	mol
Ounce	oz
Pound	lb
Pounds per square inch	psi
Second	sec
Specific gravity	sp gr
Specific heat	sp ht
Square foot	sq ft
Square inch	sq in.
Watt	w
Watthour	whr

SYMBOLS

A letter symbol is a single character, with subscript or superscript if required, used to designate a physical magnitude in mathematical equations and expressions. Two or more symbols together always represent a product. The following have been compiled from a selected list of approved standards[3].

Acceleration, due to gravity	g
Acceleration, linear	a
Area	A
Change in specific volume during vaporization	v_{fg}
Density, Weight per unit volume, Specific weight	d or ρ (rho)

$$d = \frac{1}{v}$$

Distance, linear	s
Dry saturated vapor, Dry saturated gas at saturation pressure and temperature, vapor in contact with liquid	*Subscript* g
Entropy. (The capital should be used for any weight, and the small letter for unit weight)	S or s
Force, total load	F
Head	H or h
Heat content, Total heat, Enthalpy (The capital should be used for any weight and the small letter for unit weight)	H or h
Heat content of saturated liquid, Total heat of saturated liquid, Enthalpy of saturated liquid, sometimes called heat of the liquid	h_f
Heat content of dry saturated vapor, Total heat of dry saturated vapor, Enthalpy of dry saturated vapor	h_g

[3]Letter Symbols for Mechanics of Solid Bodies, Z10 3-1942, and Letter Symbols for Heat and Thermodynamics, Z10 4-1943 (*American Standards Association*).

Heat of vaporization at constant pressure .... L or h_{fg}
Internal energy, Intrinsic energy. (The capital should be used for any weight and the small letter for unit weight) .... U or u
Length of path of heat flow, thickness L
Load, total .... W
Mechanical efficiency .... e_m
Mechanical equivalent of heat .... J
Power, Horsepower, Work per unit time .... P
Pressure, Absolute pressure, Gage pressure, Force per unit area .... p
Quantity (total) of fluid, water, gas, heat; Quantity by volume; Total quantity of heat transferred .... Q
Quality of steam, Pounds of dry steam per pound of mixture x
Saturated liquid at saturation pressure and temperature, Liquid in contact with vapor .... *Subscript* f
Specific heat c
Specific heat at constant pressure .... c_p
Specific heat at constant volume .... c_v
Specific volume, Volume per unit weight, Volume per unit mass v
Temperature (ordinary) F or C. (*Theta* is used preferably only when t is used for Time in the same discussion) .... t or θ (*theta*)
Temperature (absolute) F abs or K. (Capital *theta* is used preferably only when small *theta* is used for ordinary temperature) .... T or Θ (*capital theta*)
Thermal conductance[4]: heat transferred per (unit time) (degree) C

$$C = \frac{1}{R} = \frac{kA}{L} = \frac{q}{t_1 - t_2}$$

Thermal conductance per unit area, Unit conductance: heat transferred per (unit time) (unit area) (degree) .... C_a

$$C_a = \frac{C}{A} = \frac{1}{RA} = \frac{q}{A(t_1 - t_2)} = \frac{k}{L}$$

Thermal conductivity: heat transferred per (unit time) (unit area) (degree per unit length) .... k

$$k = \frac{\dfrac{q}{A}}{\dfrac{(t_1 - t_2)}{L}}$$

Surface coefficient of heat transfer, Film coefficient of heat transfer, Individual coefficient of heat transfer: heat transferred per (unit time) (unit area) (degree) .... f

$$f = \frac{\dfrac{q}{A}}{t_1 - t_2}$$

(In general f is not equal to k/L, where L is the actual thickness of the fluid film.)

Over-all coefficient of heat transfer, Thermal transmittance per unit area: heat transferred per (unit time) (unit area) (degree over-all) .... U

$$U = \frac{\dfrac{q}{A}}{t_1 - t_2}$$

Thermal transmission (heat transferred per unit time) .... q

$$q = \frac{Q}{t}$$

[4]Terms ending *ivity* designate properties independent of size or shape, sometimes called *specific properties*. Examples conductivity, resistivity. Terms ending *ance* designate quantities depending not only on the material, but also upon size and shape, sometimes called *total quantities*. Examples conductance, transmittance. Terms ending *ion* designate rate of heat transfer. Examples conduction, transmission.

Thermal resistance (degree per unit of heat transferred per unit time) .... R

$$R = \frac{t_1 - t_2}{q} = \frac{L}{kA}$$

Thermal resistivity.. $1/k$
Vaporization values at constant pressure, Differences between values for saturated vapor and saturated liquid at the same pressure.............................. *Subscript* $_{fg}$
Velocity.. V
Volume (total).. V
Volume per unit time, Rate at which quantity of material passes through a machine, Quantity of heat per unit time, Quantity of heat per unit weight........... q
Weight of a major item, Total weight.. W
Weight rate, Weight per unit of power, Weight per unit of time .... w
Work (total).. W

CONVERSION EQUATIONS[5]

Heat, Power and Work

1 ton refrigeration	= 12,000 Btu per hour 200 Btu per minute
Latent heat of ice	= 143.4 Btu per pound
1 Btu	= 778.3 ft-lb 0.2930 Int. whr 252.0 I.T calorie
1 Int. watthour	= 2656 ft-lb 3.413 Btu 3600 Int. joules 860 I.T. calories
1 Int. kilowatthour	= 3,413 Btu 3.517 lb water evaporated from and at 212 F
1 Int. kilowatt (1000 watts)	= 1 341 hp 56.88 Btu per minute 44,267 ft-lb per minute
1000 I.T. calories 1 I.T. Kilocalorie	= 3.968 Btu 3088 ft-lb 1.1628 Int. whr
1 horsepower	= 0.7455 Int. kw 42.40 Btu per minute 33,000 ft-lb per minute 550 ft-lb per second
1 boiler horsepower	= 33,475 Btu per hour 9 809 Int. kw

Weight and Volume

1 gal (U. S.)	= 231 cu in. 0.1337 cu ft
1 British or Imperial gallon	= 277.42 cu in.
1 cu ft	= 7.481 gal 1728 cu in.
1 cu ft water at 60 F (in vacuo)	= 62.37 lb
1 cu ft water at 212 F (" ")	= 59.83 lb
1 gal water at 60 F (" ")	= 8.338 lb
1 gal water at 212 F (" ")	= 7.998 lb
1 lb (avdp)	= 16 oz 7000 grains
1 bushel	= 1.244 cu ft
1 short ton	= 2000 lb

[5]Checked in 1944 by *National Bureau of Standards.* Abbreviations *Int* and *I.T.* refer to *International* and *International (Steam) Table* respectively

Pressure

1 lb per square inch	=	144 lb per square foot 2.0422 in. mercury at 62 F 2.309 ft water at 62 F 27.71 in. water at 62 F
1 oz per square inch	=	0.1276 in. mercury at 62 F 1.732 in. water at 62 F
1 atmosphere	=	14.696 lb per square inch 2116 lb per square foot 33.94 ft water at 62 F 30.01 in. mercury at 62 F 29.921 in. mercury at 32 F
1 in. water at 62 F (in vacuo)	=	0.03609 lb per square inch 0.5774 oz per square inch 5.197 lb per square foot
1 ft water at 62 F (in vacuo)	=	0.4330 lb per square inch 62.37 lb per square foot
1 in. mercury at 62 F (in vacuo)	=	0.4897 lb per square inch 7.835 oz per square inch 1.131 ft water at 62 F 13.57 in. water at 62 F

Metric Units

1 cm	=	0.3937 in.
1 in.	=	2.540 cm
1 m	=	3.281 ft
1 ft	=	0.3048 m
1 sq cm	=	0.1550 sq in.
1 sq in.	=	6.452 sq cm
1 sq m	=	10.76 sq ft
1 sq ft	=	0.09290 sq m
1 cu cm	=	0.06102 cu in.
1 cu in.	=	16.39 cu cm
1 cu m	=	35.31 cu ft
1 cu ft	=	0.02832 cu m
1 liter	=	1000 cu cm = 0 2642 gal
1 kg	=	2.205 lb (avdp)
1 lb	=	0.4536 kg
1 metric ton	=	2205 lb (avdp)
1 gram	=	0.002205 lb (avdp)
1 kilometer per hour	=	0.6214 mph
1 gram per square centimeter	=	0.02905 in. mercury at 62 F 0.3942 in. water at 62 F
1 kg per sq cm (metric atmosphere)	=	14.22 lb per square inch
1 gram per cubic centimeter	=	0.03613 lb per cubic inch 62.43 lb per cubic foot
1 dyne	=	0.00007233 poundals
1 absolute joule	=	10,000,000 ergs 0.7376 ft-lb
1 Int. joule	=	0.7378 ft-lb
1 metric horsepower	=	75 kg-m per second 0.986 hp (U. S.)
1 I. T. kilocalorie per kilogram	=	1.8 Btu per pound
1 I.T. calorie per square centimeter	=	3.687 Btu per square foot
1 I.T. calorie per second per square centimeter for a temperature gradient of 1 deg C per centimeter	=	2903 Btu per hour per square foot for a temperature gradient of 1 deg F per inch of thickness.

GRAPHICAL SYMBOLS FOR DRAWINGS[6]

GRAPHICAL SYMBOLS FOR DRAWINGS ***Piping***

HEATING

1. High Pressure Steam
2. Medium Pressure Steam
3. Low Pressure Steam
4. High Pressure Return
5. Medium Pressure Return
6. Low Pressure Return
7. Boiler Blow Off
8. Condensate or Vacuum Pump Discharge
9. Feedwater Pump Discharge
10. Make Up Water
11. Air Relief Line
12. Fuel Oil Flow
13. Fuel Oil Return
14. Fuel Oil Tank Vent
15. Compressed Air
16. Hot Water Heating Supply
17. Hot Water Heating Return

AIR CONDITIONING

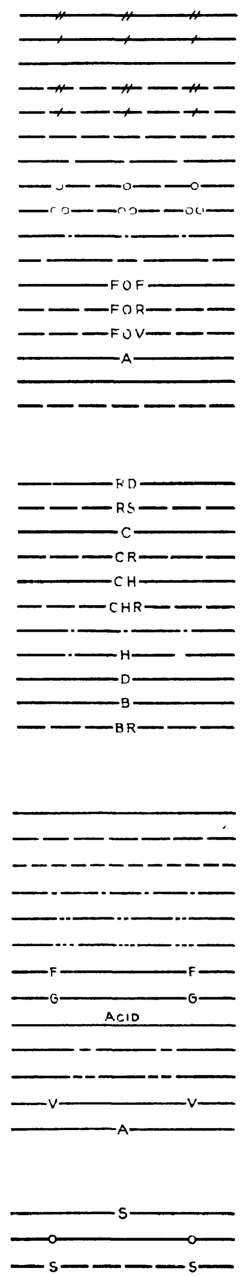
18. Refrigerant Discharge
19. Refrigerant Suction
20. Condenser Water Flow
21. Condenser Water Return
22. Circulating Chilled or Hot Water Flow
23. Circulating Chilled or Hot Water Return
24. Make Up Water
25. Humidification Line
26. Drain
27. Brine Supply
28. Brine Return

PLUMBING

29. Soil, Waste or Leader (Above Grade)
30. Soil, Waste or Leader (Below Grade)
31. Vent
32. Cold Water
33. Hot Water
34. Hot Water Return
35. Fire Line
36. Gas
37. Acid Waste
38. Drinking Water Flow
39. Drinking Water Return
40. Vacuum Cleaning
41. Compressed Air

SPRINKLERS

42. Main Supplies
43. Branch and Head
44. Drain

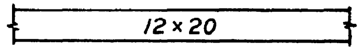


[6]Graphical Symbols for Use on Drawings in Mechanical Engineering, Z32.2-1941 (*American Standards Association*)

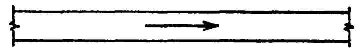
Graphical Symbols for Drawings

Ductwork

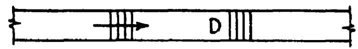
45. Duct (1st Figure, Width; 2nd, Depth)



46. Direction of Flow



47. Inclined Drop in Respect to Air Flow



48. Inclined Rise in Respect to Air Flow

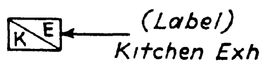
49. Supply Duct Section

50. Exhaust Duct Section

51. Recirculation Duct Section

52. Fresh Air Duct Section

53. Other Duct Sections

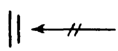


54. Register — R

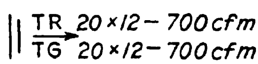
55. Grille — G

56. Supply Outlet

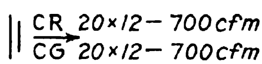
57. Exhaust Inlet



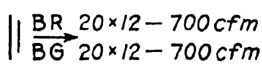
58. Top Register or Grille



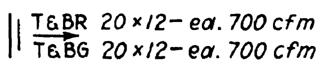
59. Center Register or Grille



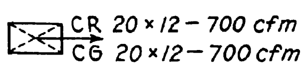
60. Bottom Register or Grille



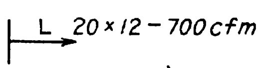
61. Top and Bottom Register or Grille



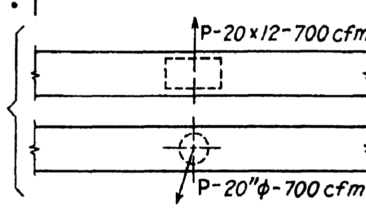
62. Ceiling Register or Grille



63. Louver Opening

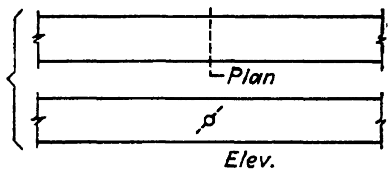


64. Adjustable Plaque

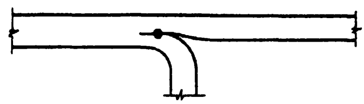


Graphical Symbols for Drawings *Ductwork*

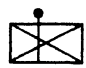
65. Volume Damper



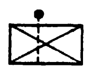
66. Deflecting Damper



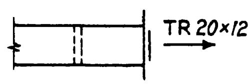
67. Deflecting Damper, Up



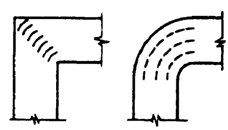
68. Deflecting Damper, Down



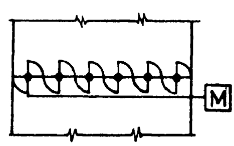
69. Adjustable Blank Off



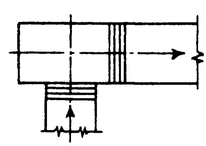
70. Turning Vanes



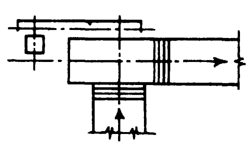
71. Automatic Dampers



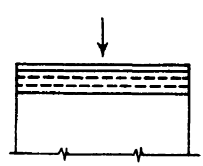
72. Canvas Connections



73. Fan and Motor With Guard



74. Intake Louvers and Screen



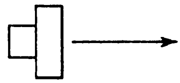
GRAPHICAL SYMBOLS FOR DRAWINGS

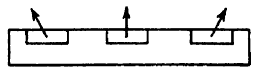
Heating and Ventilating

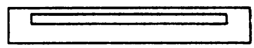
75. Heat Transfer Surface, Plan 

76. Wall Radiator, Plan 

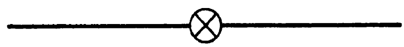
77. Wall Radiator on Ceiling, Plan 

78. Unit Heater (Propeller), Plan 

79. Unit Heater (Centrifugal Fan), Plan 

80. Unit Ventilator, Plan 

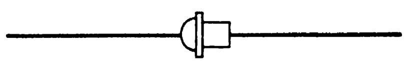
TRAPS

81. Thermostatic 

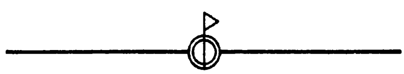
82. Blast Thermostatic 

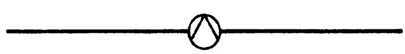
83. Float and Thermostatic 

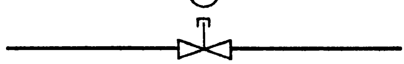
84. Float 

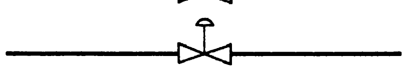
85. Boiler Return 

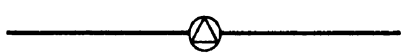
VALVES

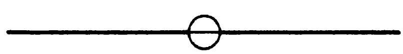
86. Reducing Pressure 

87 Air Line 

88. Lock and Shield 

89. Diaphragm 

90. Air Eliminator 

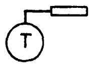
91. Strainer 

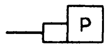
92. Thermometer 

93. Thermostat 

RAPHICAL SYMBOLS FOR DRAWINGS — *Refrigerating*

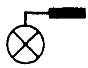
)4. Thermostat (Self Contained)

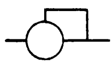

)5. Thermostat (Remote Bulb)


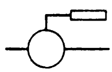
)6. Pressurestat


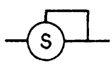
)7. Hand Expansion Valve

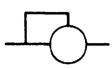

)8. Automatic Expansion Valve


)9. Thermostatic Expansion Valve


)0. Evaporator Press. Regulating Valve, Throttling Type


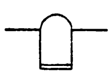
)1. Evaporator Press. Regulating Valve, Thermostatic Throttling Type


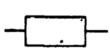
)2. Evaporator Press. Regulating Valve, Snap-Action Valve


)3. Compressor Suction Pressure Limiting Valve, Throttling Type


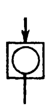
)4. Hand Shut Off Valve


)5. Thermal Bulb

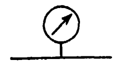

)6. Scale Trap


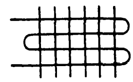
07. Dryer


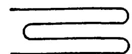
08. Strainer

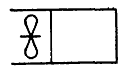

09. High Side Float


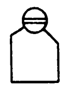
110. Low Side Float


111. Gage


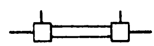
112. Finned Type Cooling Unit, Natural Convection


113. Pipe Coil


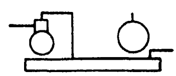
114. Forced Convection Cooling Unit


115. Immersion Cooling Unit


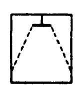
116. Ice Making Unit

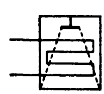

117. Heat Interchanger


118. Condensing Unit, Air Cooled

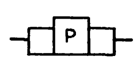

119. Condensing Unit, Water Cooled


120 Compressor


121. Cooling Tower


122. Evaporative Condenser


123. Solenoid Valve


124. Pressurestat With High Pressure Cut-Out


SPECIFIC HEAT

Table 1. Specific Heat of Solids[a]

Materials	Temperature F	Specific Heat	Authority
Alloys			
Brass, Red	32	0.0899	S
Brass, Yellow	32	0.0883	S
Bronze (80*Cu*, 20*Sn*)	57–208	0.0862	S
Monel Metal	68–2370	0.127	S
Aluminum	80–212	0 212	S
Asbestos	68–208	0.195	S
Brickwork	....	0.195	H
Carbon (Graphite)	104–1637	0.314	I
Coal	....	0.278	H
Coke	....	0 201	H
Concrete	....	0 270	H
Copper	64–212	0.0928	S
Fire Clay Brick	77–1832	0.258	I
Glass			
Crown	50–122	0 161	S
Flint	50–122	0.117	S
Gold	64	0 0312	S
Gypsum	....	0.259	H
Ice	32	0.487	S
Ice	−40	0 434	S
Iron, Pure	32	0.1043	S
Iron, Pure	32–600	0 127	M
Iron, Cast	68–212	0.1189	H
Iron, Wrought	59–212	0 1152	H
Lead	32	0 0297	S
Nickel	32	0.1032	S
Masonry	....	0 2159	H
Plaster	....	0.2	H
Platinum	58–212	0 0319	S
Rocks			
Gneiss	63–210	0 196	S
Granite	54–212	0.192	S
Limestone	59–212	0 216	S
Marble	32–212	0 21	S
Sandstone	....	0.22	S
Silver	32	0 0536	S
Steel	....	0 1175	H
Sulphur	240–320	0 220	S
Silica Brick	77–1832	0 263	I
Tin	77	0.0548	S
Woods (Average)	68	0 327	S
Zinc	32	0 0913	S

Table 2. Specific Heat of Liquids

Liquid	Temperature F	Specific Heat	Authority
Alcohol, Ethyl	32	0 548	S
Alcohol, Methyl	59–68	0 601	S
Glycerine	59–122	0 576	S
Lead (Molten)	360	0 041	H
Mercury	68	0 03325	S
Petroleum	70–136	0 511	S
Sea Water			
Sp Gr 1 0043	64	0 980	S
Sp Gr. 1.0463	64	0 903	S
Water	59	1 000	S

Table 3. Specific Heat of Gases and Vapors

Substance	Temperature F	Specific Heat at Constant Pressure	Ratio of Specific Heat C_p/C_v	Specific Heat at Constant Volume (Computed)	Authority
Air	32–392	0 2375	1.405	0.169	S
Ammonia	80–392	0 5356	1.277	0 419	S
Carbon Dioxide	52–417	0 2169	1.3003	0 1668	S
Carbon Monoxide	79–388	0 2426	1.395	0 1736	S
Coal Gas	68–1900	0 3145	....	....	S
Flue Gas	....	0 24 (Approx.)	....	....	H
Hydrogen	70–212	3 41	1.419	2.402	S
Nitrogen	32–392	0.2438	1 41	0.1729	S
Oxygen	55–404	0 2175	1.3977	0 155	S
Water Vapor	212	0 421	1.305	0.322	S
Water Vapor	356	0.51	....	....	S

[a]See also The Specific Heat of Thermal Insulating Materials, by Gordon B. Wilkes and Carl O Wood (A S H V E Transactions, Vol. 48, 1942, p. 493)

Notes When one temperature is given the true specific heat is given, otherwise the value is the mean specific heat between the given limits.

Authorities· S—Smithsonian Physical Tables, 1933; I—International Critical Tables; H—Heating, Ventilation and Air Conditioning, by L. A. Harding and A. C. Willard; M—Engineers' Handbook, by Lionel S. Marks.

STATE CODES

The material in Table 4 was compiled by the A.S.H.V.E. and is based on information furnished by the individual states through a comprehensive survey conducted in 1943. References are given to the titles of all codes, standards or laws which relate to this field of engineering and where copies of these standards may be obtained.

Table 4. State Codes, Standards or Laws Relating to the Heating, Ventilating or Air Conditioning of Buildings

State	Codes, Standards or Laws	Where Material Can be Obtained
ALABAMA	None	
ARIZONA	Law 56-117 Laundry—hours of labor—ventilation—penalty; Law 65-218 Ventilation (with reference to mines)	Secretary of State, State House, Phoenix
ARKANSAS	Installation of all boilers	Dept of Labor, State Capitol, Little Rock
CALIFORNIA	Reference to the ventilation laws may be found in the California Health and Safety Code under the following sections Sec 16800—Construction Requirements, Air Ducts Sec 16820-16835—Vent Shafts Sec. 16900-16905—Gas Appliance Vents Sec. 17080-17088—Garages Sec. 16233-16235, 16270-16271—Rooms Sec 16300-16305—Stairways	Supervisor of Documents, 214 State Capitol, Sacramento
COLORADO	None	
CONNECTICUT	School Building Code (1941), Chapter 6—Structural and Mechanical, D Heating and Ventilation	State Dept. of Education Div. of Instruction, Hartford
	Statutes Section 2566 specifies the number of cubic feet of air for children and adults in tenement and boarding houses	State Dept of Health, Hartford
	Section 2585, amd by 443f, specifies ventilation required for bathrooms and toilets in tenament houses.	
	Sanitary Code· Regulations 280 and 281 concern ventilation to be provided for industrial processes Regulation 127 concerns ventilation of all new water closet compartments	
	Labor Laws (1941) VIII Industrial Safety, A General, Sec 2355—Lighting and sanitary condition of factories and round houses, IX. Industrial Health and Sanitation, Legislation, Sec 2355—Lighting and sanitary condition of factories and round houses	Dept of Labor and Factory Inspection Hartford.
DELAWARE	Minimum Standards for School Buildings and Sites (1931).	State Board of Education, Dover
DISTRICT OF COLUMBIA	Building Code (1941), Section 505—Mechanical Ventilation, Section 702, Article 702-04—Construction of Air Conditioning Ducts.	Engineering Dept., Dept of Inspection, Govt. District of Columbia
FLORIDA	School Code Law; Laws Relating to Construction Hotels, Apartment Houses, and Public Eating Places	Secy of State, Tallahassee
GEORGIA	None.	
IDAHO	None	
ILLINOIS	Laws Relating to Labor and Employment (1941) Health and Safety Act; Occupational Diseases, Compressed Air Employment	Dept. of Labor, Capitol Bldg, Springfield
	Illinois School Law, Section 15, paragraph 20 is amplified in April, 1940, Educational Press Bulletin with sections on Ventilating and Heating	Superintendent of Public Instruction, Springfield
INDIANA	Rules and Regulations of the State Board of Health Governing the Construction, Equipment and Maintenance of Sanitary Features of Public and Parochial School Buildings (1936).	Division of Environmental Sanitation, State Board of Health, 1098 W. Michigan St, Indianapolis.
	Rules and Regulations of the Administrative Building Council—Article VIII, Sections 19-8-1, 19-8-2 and 19-8-3 Pertaining to Ventilation of Rooms and Plumbing Fixtures.	

TABLE 4. STATE CODES, STANDARDS OR LAWS RELATING TO THE HEATING, VENTILATING OR AIR CONDITIONING OF BUILDINGS—(Continued)

STATE	CODES, STANDARDS OR LAWS	WHERE MATERIAL CAN BE OBTAINED
IOWA	Code of Iowa, Chapter 323, Requirements for Sanitation and Ventilation of Single Family and Multiple Dwellings. Code of Iowa, Chapter 73, Ventilation Requirements in Industrial Establishments Pertaining to Health and Safety Department of Public Instruction Rules Relating to Heating and Ventilating of School Buildings	Dept of Health, Div. Public Health Engr. and Industrial Hygiene, Des Moines.
KANSAS	Matter is left to judgment of safety engineer inspector. See Labor Laws of Kansas (1942). G S. 1935, 44-636.	Labor Department, 801 Harrison, Topeka.
KENTUCKY	Statutes of Kentucky: Section 217 280 provides for the proper ventilation of food establishments, Sections 352 020, 352 030, 352.040, 352.050, 352 080, 352 090, 352.120, 352 150, 352 190, 352 210, 352.320, 352.330, 352.340, 352.350, 352 360, 352.370, 352.380, 352.390 and 352.400; also, 352 570, 352.580, 352 590, 352.600 and 352.610 provide for the proper ventilation of mines	Secy. of State, Frankfort.
LOUISIANA	None.	
MAINE	Regulations bearing on this subject in reference to the installation of plumbing fixtures in closed rooms are outlined in Sections 101 and 102 of the State Plumbing Code. Rules and Regulations Relating to Sanitation of Factories and Mercantile Establishments include Sections on Ventilation.	Division of Sanitary Engineering Dept. of Health and Welfare, State House, Augusta
MARYLAND	Standard for Maryland School Buildings, Revised, 1941	State Dept. of Education, Baltimore
MASSACHUSETTS	Laws Relating to the Erection, Alteration, Inspection and Use of Buildings (Form A), Regulations Relative to the Inspection of Buildings Which Are Subject to the Provisions of Chapter 143, General Laws (Form B-1)	Dept of Public Safety, Division of Inspection, 1010 Commonwealth Ave, Boston
MICHIGAN	Housing Law of Michigan (1939)	Secretary of State, Capitol Bldg, Lansing.
MINNESOTA	Laws Relating to Sanitation, Ventilation and Toilets in Factories, etc Also requirements in regard to garages, spray rooms and spray booths. General Orders on Dusts, Fumes, Vapors and Gases.	Industrial Commission, State Office Bldg., St Paul
MISSISSIPPI	None.	
MISSOURI	State Labor and Industrial Inspection Laws (1941).	Dept. of Labor and Industrial Inspection, State Office Bldg., Jefferson City.
MONTANA	Revised Codes of Montana (1935), Section 1175 refers to ventilation of school buildings	Secretary of State, Helena.
NEBRASKA	Safety Codes (1937) and Labor Laws (1943).	Dept. of Labor, Lincoln.
NEVADA	Nevada Compiled Laws (1929), Sections 5894 and 5715 pertain to School Buildings. Compiled Labor Laws (1937), Part 14, Section 4241 Ventilation of Mines, Part 8, Sec 2817 Ventilation Bunk Houses	Secretary of State, Carson City.
NEW HAMPSHIRE	Under the provisions of Chapter 215 of the Revised Laws, Safety and Health of Employees, recommendations are made in mills factories, workshops, commercial and mercantile establishments for proper ventilation	Bureau of Labor, Concord
NEW JERSEY	Industrial Code Bulletin and Labor Laws	State Dept of Labor, Wallach Bldg, Trenton.
NEW MEXICO	Regulations Governing the Heating and Ventilation of Tourist Courts, Tourist Camps, Hotels and Lodging Houses (1939)	Dept. of Public Health, Santa Fe.
NEW YORK	Commissioner of Education is authorised to determine the requirements for proper ventilation of school buildings. Codes are enforced by the Department of Labor which contain sections on ventilation requirements pertaining to a number of industrial processes, such as: No. 10 Foundry Code; No 12 Dust, Fumes and Gases; No. 32 Automobile Spray, No 33 Rock Drilling, No 34 Stone Crushing; No 35 Stone Cutting and Finishing; Nos. 9 and 16 Sanitary Code; No. 17 Mines and Quarries; No 25 Removal of Toxic Gases in Mines, Tunnels and Shafts; No. 37 Military Pyrotechnics	Commissioner of Education, Albany. Secretary, Labor Dept., 80 Centre St, New York

TABLE 4. STATE CODES, STANDARDS OR LAWS RELATING TO THE HEATING, VENTILATING OR AIR CONDITIONING OF BUILDINGS—(Concluded)

STATE	CODES, STANDARDS OR LAWS	WHERE MATERIAL CAN BE OBTAINED
NORTH CAROLINA	Building Code (1936) Chapter 14, Heating and Mechanical Ventilation, Experiment Station Bulletin No 10	North Carolina State College, Raleigh.
NORTH DAKOTA	None.	
OHIO	Ohio State Building Codes: No 102 Theaters and Assembly Halls (1940); No. 103 School Buildings (1938); No 105 Churches (1938); No 106 Hospitals and Homes (1936); No. 107 Hotels and Apartments (1940); No. 108 Public Garages (1938); No 109 (1941), Workshops, Factories, Mercantiles and Office Buildings.	Dept. of Industrial Relations, Div. of Factory and Building Inspection, Columbus.
OKLAHOMA	Bureau of Factory Inspection Bulletin No. 7-A, containing the laws governing the inspection and regulation of factories or other places where labor is employed Bureau of Factory Inspection Book No. 11-A—Petroleum Industry Safety Standards.	Dept of Labor, Oklahoma City
OREGON	General Safety Manual, Logging Safety Code; Sawmill and Woodworking Safety Code; Safety Code for Construction Work; Electrical and Communication Workers Safety Code; Safety Code for Arc Welding, Sandblasting and Spray Painting Operations	Industrial Accident Commission, Salem
	State laws and regulations relating to school buildings, sanitation, plumbing, heating and ventilation, including factory inspections.	Bureau of Labor, State Library Building, Salem
PENNSYLVANIA	Regulations For: Abrasive and Polishing Wheels, Bedding and Upholstery, Brewing and Bottling; Canneries; Cereal Mills, Malt Houses and Grain Elevators; Construction and Repairs, Dry Cleaning, Dry Color Industry; The Storage, Handling and Use of Explosives, Electric Safety Code; Fire and Panic Regulations, Foundries, General Safety Act, Industrial Sanitation; Lead Manufacturing; Paint Grinding, Regulations for Labor Camps; Laundries, Lead Corroding and Lead Oxidising; Logging, Sawmill, Woodworking, Veneer and Cooperage Operations; Mines other than Coal Mines, Miscellaneous Hazards and Conditions of Employment, the Manufacture of Nitro and Amido Compounds; Plants Manufacturing or Using Explosives; Printing and Allied Industries, Spray Coating, Tunnel Construction and Work in Compressed Air; Textile Industries	Dept of Labor and Industry, Harrisburg
RHODE ISLAND	Rhode Island Labor Laws (1942)	Dept of Labor, State House, Providence.
SOUTH CAROLINA	None.	
SOUTH DAKOTA	State Code, Chapter 13 2018 Care of Steam Boilers, Chapter 27 1711 Ventilation of Hotels, Rooming Houses, Restaurants, Tourist Camps	Office of State Engineer, Pierre
TENNESSEE	Williams Code of Tennessee, Sections 5341-5343, which deal with the ventilation of workshops and factories, places of amusement, etc.	Secy of State, Nashville.
TEXAS	School Building Law, Bulletin 382, February, 1938, Article No 2920, No. 2921, No. 2922.	Dept of Education, Austin
UTAH	35-1-16. Sanitary Code for Public Buildings and Railway Cars, 35-1-17. Supervision of Bathing Places.	Dept. of Engineering, 439 State Capitol Bldg, Salt Lake City.
VERMONT	Public Laws Rules and Regulations Relating to Public Buildings, Sanitation, Plumbing, Heating and Ventilation (1941)	State Board of Health, Burlington.
VIRGINIA	Mining Laws of State of Virginia have provisions regarding ventilation in mines	Dept of Labor and Industry, Finance Bldg, Richmond.
WASHINGTON	General Safety Standards: Standard No 49 Blower and Exhaust Systems; No 50 Respirators, Helmets, etc., No 51 Carbon Monoxide Gas; No. 49 Ventilation; No. 210 Laundry Ventilation; No 154 Dry Cleaning with Volatile, Inflammable or Explosive Liquids. Coal Mining Laws; Occupational Disease Code, Metal Mine Standards; Safety of Persons Employed in Tunnels, Quarries, Caissons or Subways; Construction Code.	Dept of Labor and Industries, Olympia.
WEST VIRGINIA	Installation Regulations for Power Boilers	Dept. of Labor, Charleston.
WISCONSIN	Heating, Ventilation and Air Conditioning Code (1939). Applies to public buildings and places of employment General Orders on Spray Coating (1939) General Orders on Dusts, Fumes, Vapors and Gases (1941).	Industrial Commission of Wisconsin, Madison.
WYOMING	None.	

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MANUFACTURERS' CATALOG DATA

(PAGES 843-1152)

HEATING
VENTILATING
AIR CONDITIONING
GUIDE
1945

On pages 843-1152 will be found the Catalog Data of 211 manufacturers whose products are described and illustrated.

For the convenience of the user of THE GUIDE 1945 there are eight main divisions:

On pages 815-840, under each of the index headings—Air Cleaning Equipment, Fans, Humidifiers, Ventilators, etc., will be found a list of manufacturers of any desired products, fully cross-indexed, and the page numbers in the Catalog Data Section where the products are described.

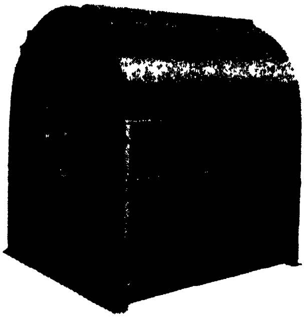
By reference to these indices, the manufacturers names and the page numbers, any item of equipment or materials, and the producers address, may be located quickly.

In the interest of paper conservation the listings have been limited by the advertisers.

American Blower Corporation

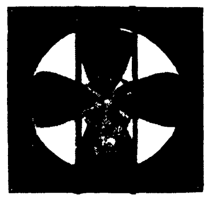
P. O. Box 58, Roosevelt Park Annex, **Detroit 32, Michigan**
CANADIAN SIROCCO COMPANY, LTD.
in Canada, 310 Ellis Street, Windsor, Ontario
Branch Offices in Principal Cities
Division of American Radiator and Standard Sanitary Corporation

AIR CONDITIONING — HUMIDIFYING — DEHUMIDIFYING — COOLING VENTILATING — HEATING — VAPOR-ABSORPTION — DRYING — AIR WASHING AND PURIFICATION — EXHAUSTING EQUIPMENT AND MECHANICAL DRAFT APPARATUS

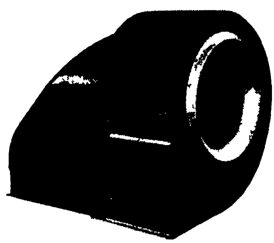


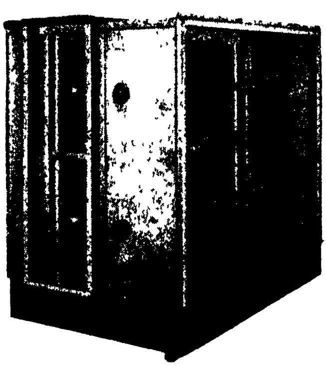
Double Inlet "ABC" Multiblade Fan—above, is a heavy duty ventilating fan. Its wheel has narrow, forward pitched blades. Low tip speeds assure quiet operation. Request Bulletin A-701. Write for Bulletin A-403 for backwardly inclined, non-overloading H. S. Fan.

Commercial V-Belt Drive Ventura Fans—for ventilating applications without duct systems, where extremely quiet operation is desired. Inlet-outlet streamlined for high efficiency. Also direct connected. Request Bulletin B-2529.

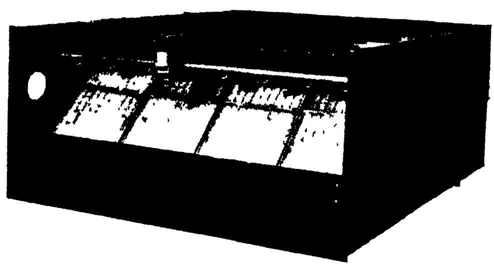


"ABC" Utility Sets—right, complete packaged units, direct connected or V-Belt short coupled drive, for duct applications. Famous "ABC" Multiblade Wheel operates at low tip speeds. Quiet, compact. Bulletin B-2529.





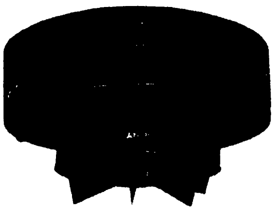
American Blower Air Washer—above, cleans, purifies and freshens the air, removes dust, odors and bacteria, cools if desired and provides an effective method of controlling humidity. Bulletin 3623.



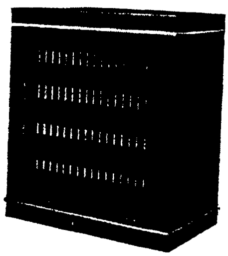
American Blower Capillary Air Washers—above, for high efficiency in cleaning, humidification, cooling and dehumidification of air. A highly efficient surface contact mechanism, the capillary cell, is used. Air is forced at low resistance through long, irregular passages of small size formed by a large amount of thoroughly wetted glass surface. Unit includes a substantial metal casing and tank of air washer design, capillary cells, improved low head sprays, metal or glass fibre low resistance moisture eliminators, non-ferrous, extended surface cooling or heating coils. Write for Bulletin 3723.

TYPES OF AMERICAN BLOWER CORPORATION AIR HANDLING AND CONDITIONING EQUIPMENT

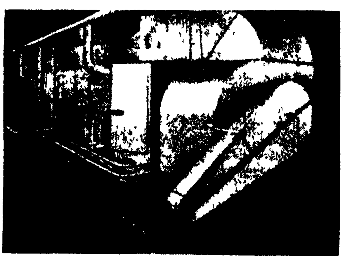
All types of air handling and air conditioning equipment for industrial applications, process work, drying, cooling; also equipment for stores, offices, shops, public buildings, power plants, etc., and attic ventilation for homes.



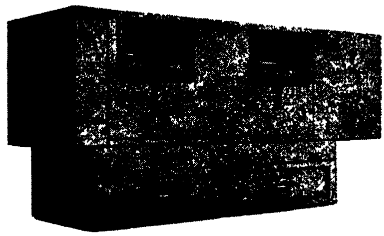
"ABC" Vertical Heaters—for ceiling applications, give an even, wide floor area distribution of heat. For either steam or hot water heating systems. Variable speed, 2-speed and constant speed models. Write for Bulletin A-9418.



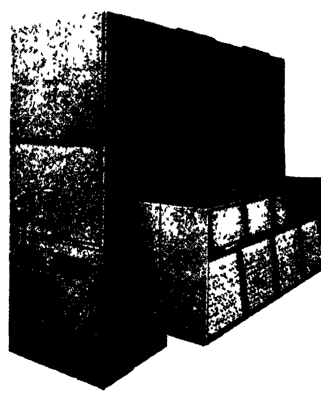
Venturafin Unit Heaters—for many general purpose heating jobs. Wall or ceiling mounting. Streamline construction, rugged heating elements. Steam or hot water. Write for Bulletin A-8218.



Air Conditioning Central Systems—provide an effective way of cooling, heating, humidifying, dehumidifying and purifying air in all classes of business and public buildings where a dust system is desirable. Write for Special Data.



"HV" General Purpose Units—with air filters and Aileron control. Ideal wherever attractive, quiet and economical heating and ventilating units are required. Wall, floor or ceiling mounting. Offer great flexibility of design and arrangement to meet specific needs. Write for Bulletin 5927.



American Blower Series "H" Air Conditioners with Sprayed Coils—are usually applied for industrial uses where air washing and evaporative cooling are required. Sprayed coils give cleaner air, cut coil maintenance and refrigeration costs, reduce necessary air volumes, permit use of smaller ducts and grilles. Horizontal or floor types (as shown). Aileron control provides simple method of regulating flow of air from the fans. Write for Bulletin 6027.

The Bahnson Company

AIR CONDITIONING ENGINEERS

Winston-Salem, N. C.

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HUMIDIFYING—COOLING—HEATING—AIR CLEANING—VENTILATING

Bahnson Industrial Air Conditioning and Humidifying Equipment has been installed throughout the United States and in over 30 foreign countries during the past 28 years.

In designing various types of equipment to meet the requirements of Industrial Air Conditioning and Humidifying, the Bahnson Company has considered as paramount· Flexibility of capacity and control, Economy of Operation, and Simplicity of Design.

CENTRISPRAY

The Bahnson Company has completed the development of a central station air washer so radically new in design, that a basic patent has been obtained on the principle involved.

The Centrispray utilizes the efficient principle of centrifugal atomization to produce a very fine and readily absorbed mist in the air washer. Saturation at 24 degrees differential between the wet and dry bulb can be obtained through this principle with one-tenth of 1/100 the amount of water handled by conventional type air washers. This affords notable savings in pumping costs.

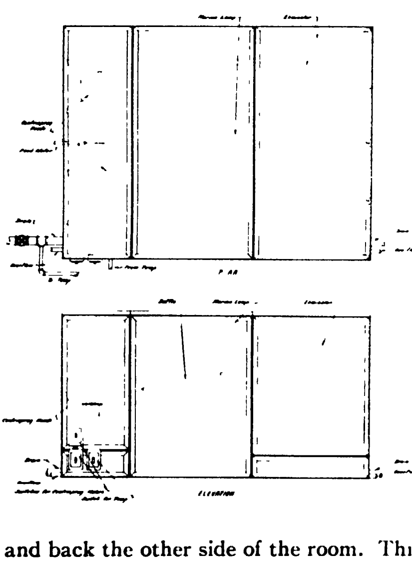
Saturation to 96 per cent or higher can be obtained through the Centrispray washer with velocities of 700 feet per minute. Due to the relatively small amount of water that is circulated, the tank of the Centrispray can be made much smaller than the tank of a conventional washer. This results in an overall operating weight substantially less than other washers of the same capacity.

The Centrispray Washer is designed in several unit sizes up to 15,000 cfm for unit air conditioning with saturated air. Washers from 4000 cfm up to any desired capacity are designed for central station air conditioning. The Centrispray air washer is combined with heating, refrigeration, air filtering, chemical dehumidifying, to meet any air conditioning requirements.

AIR-VITALIZER

The Bahnson Air-Vitalizer air conditioning system employs a dry duct unit distribution system combined with the Bahnson Centrifugal Humidifier or the Bahnson Atomizer, to obtain evaporation.

The Vitalizer Units are usually arranged in the room so that the general movement of air is carried down one side of the room



and back the other side of the room. This feature, known as *Horizontal Circulation,* evens out the condition in the room despite the concentrated heat loads in motor alleys and sections of machinery with high horsepower requirements.

The flexibility of the Vitalizer system permits the system to be designed for any air handling or evaporative capacity.

ECONOMIZER

ATOMIZER

The Bahnson Economizer, an atomizer embodying a new principle of variable capacity, will produce a finer spray at a given evaporation, with minimum air pressure and air consumption.

It is an outstanding development in the atomizer field, featuring Flexibility, Economy, and Simplicity of installation, operation and design.

The Bahnson Company

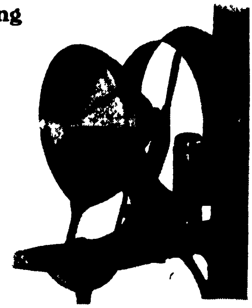
Winston-Salem, N. C.

CENTRIFUGAL HUMIDIFIER

With this unit, Bahnson has pioneered in developing the principle of centrifugal atomization for over 30 years. Continuous refinement has produced three improved types of units with varying capacities to meet different requirements. (Types D, H and L).

The Bahnson Centrifugal Humidifier requires only feed water at city main pressures (15 to 30 lbs.), a small amount of electric current at the usual voltage specifications, and a drain line. Equipped with an individual control, it is the only type of self-contained industrial humidifying unit that will produce a fine mist, distribute this mist throughout the space to be conditioned, maintain air circulation within that space, and automatically control the humidity within that space.

These units may be equipped with the Type J individual control, or may be operated with the highly sensitive Master B group control that actuates an electric motor valve on the feed water line to a group of humidifiers.



Type H with J Control

The Atomizing Disc is mounted on one end of the fractional horsepower motor. A row of stationary atomizing teeth surround the periphery of this disc. Water, fed onto the rapidly whirling disc, is hurled against the stationary teeth and broken into a fine, cloudlike mist. A fan mounted on the back end of the motor shaft distributes this fine mist and produces a directional air flow. This maintains air circulation and uniformity of condition throughout the space to be conditioned.

Write for Bulletin 327S, or special bulletins on special applications.

HUMIDUCT

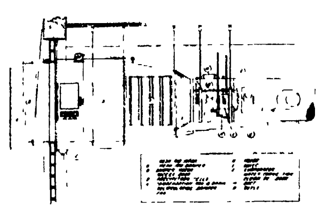
AIR CONDITIONING SYSTEM

The Bahnson Company has led in the development of two new principles of industrial air conditioning during the last ten years with its "Humiduct" system of air conditioning.

The Bahnson Humiduct was the first practical unit for utilizing the advantages of delivering saturated air plus entrained moisture from an air conditioning system, rather than saturated air alone to meet certain air conditioning requirements. Where it is necessary to maintain constant relative humidities in areas with concentrated heat loads, this principle of delivering entrained moisture so that part of the evaporation may take place in the room, shows marked advantages in more accurate control of the humidity. The humidity can be maintained by the use of this principle with a much lower volume of air than with saturated air types of systems.

Write for further information on the psychrometrics of this principle.

The Bahnson Humiduct is designed as a unit system of air conditioning. It has shown marked advantages in maintaining uniform conditions throughout the rooms of industrial plants that have relatively high heat loads or varying concentrations of machinery and heat. These advantages together with the flexibility of operating the unit system, are becoming widely recognized. The Bahnson Company introduced the principle of *Horizontal Circulation* with a unit system. The units are installed to produce a general air movement down one side of a room and back the other. This produces unusual evenness in the room conditions together with a sensible cooling effect.



Carrier Corporation

Home Office and Factories: **Syracuse 1, N. Y.**

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NEW YORK 17, N. Y.

INTERNATIONAL DIVISION:
122 EAST 42nd ST.
NEV YORK 17, N. Y.

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Dealers in Principal Cities

AIR CONDITIONING

Unitary and central system air conditioning equipment is available from Carrier Corporation in a wide range of capacities and for all types of applications—industrial and comfort. The equipment includes:

Room Coolers—provide summer comfort air conditioning in individual rooms, private offices and similar sized enclosures. Each is complete, containing the air handling and refrigerating system in a decorative compact cabinet.

Self Contained Air Conditioners—provide summer comfort in retail shops, general offices, beauty salons and commerical spaces of medium size. Each unit is complete, containing the air handling and refrigerating system in a decorative cabinet for location within the air conditioned space.

Assembled Air Conditioners—provide summer air conditioning in laboratories, general offices, stores and similar interiors. Unit is complete, with air handling and refrigerating equipment contained in a housing, for location outside air conditioned space. Ducts are necessary for distribution of the conditioned air.

Unitary and Central Station Air Conditioners—for groups of rooms, as offices and laboratories, and for large spaces as stores, factories, theatres, industrial plants and other interiors requiring industrial or comfort year round air conditioning Equipment includes complete air handling apparatus and is available in floor and suspension models. For cooling and dehumidifying, the unit is supplemented by refrigerating apparatus.

"Weathermaster" Systems—for multi-room buildings as apartments, hospitals, hotels, office buildings. System consists of room units with distributing system and central station air conditioner. Combines individual room control advantages with efficiency and economy of central station system. Practicable for old or new buildings. Conduit Weathermaster system uses conduit ducts effecting great savings in floor area. Individual room control of temperature is independent of other rooms.

Dehydration Units—for industrial applications, as drying, chemical processing and production requiring low moisture content air. Ducts are employed for the distribution of dehydrated air. When low temperatures are required, the unit is supplemented by refrigerating apparatus.

Blast Freezers and Cold Diffusers—for food freezing and storage, meat packer operations and other industries requiring low temperature air. Units include complete air handling apparatus and are available in suspended or floor type, water defrosting and with or without brine spray. Used within the space to be refrigerated or remotely located with duct system for air distribution. Supplemented by refrigerating apparatus.

REFRIGERATION

Centrifugal and reciprocating refrigerating equipment, covering a wide range of capacities and for temperatures well below —100° F is available from Carrier Corporation. There is a machine of correct capacity and at the correct temperature for each installation requirement.

Centrifugal Refrigerating Machine—to provide temperatures well below —100° F for industrial processing and chilling water for air conditioning in industrial and comfort applictions.

Reciprocating Refrigerating Machines—to provide refrigeration for industrial and comfort air conditioning and for low temperature air conditioning in food freezing and preservation, meat packing, and other applications requiring sub-freezing temperatures.

Evaporative Condensers—for condensing refrigerants in reciprocating refrigerating systems and for cooling of engine jacket water, coolants, lubricating oils and other fluids. Efficient and economic removal of heat is effected by utilizing evaporation of moisture from wetted coil surfaces for heat disposal Units are available in capacities suitable for a wide range of applications.

Commercial Refrigerating Compressors—for storage refrigerators, display cases milk coolers, ice makers, baker refrigerators, farm and home food freezers in wide range of capacities and temperatures

INDUSTRIAL HEATING

Unit heaters, propeller fan type in suspended models and centrifugal fan type in suspended and floor models, are available from Carrier Corporation. The wide range of sizes and capacities are suitable for steam or hot water circulation.

Unit Heaters—for commercial and industrial uses. Suspended, propeller fan type available in capacities ranging from 18,000 to 579,000 Btu per hour.

Heat Diffusing Unit—for commercial and industrial buildings. Suspended or floor models, centrifugal fan type with selective air distribution. Capacities range from 105,000 to 1,035,000 Btu per hour.

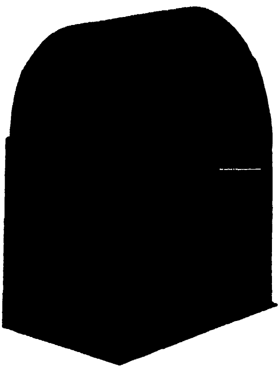
Clarage Fan Company

Kalamazoo, Michigan

Application Engineering Offices In Principal American Cities

(Consult Telephone Directory)

Clarage Air-Handling and Conditioning Equipment



Fans for ventilating and air conditioning.
28 sizes; 200 to 100,000 cfm

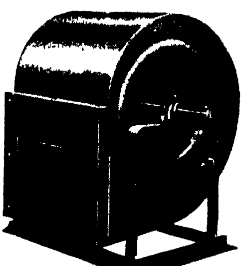
Fans • Blowers • Air Washers Air Conditioning Systems and Units • Unit Heaters & Coolers

For over a quarter-century Clarage has been a leading manufacturer of equipment for ventilating, heating, cooling, drying, air cleaning, humidifying, dehumidifying, complete air conditioning, exhausting, pneumatic conveying and mechanical draft This equipment is designed to meet all types of industrial, commercial, public building and marine requirements Whatever your air-handling or conditioning problem, Clarage is an excellent source of supply

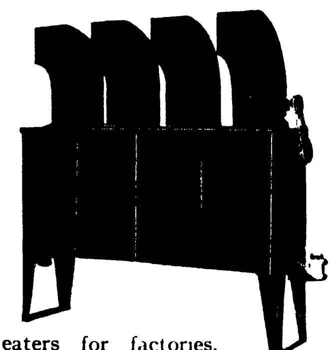
★

We build many other types of fans and allied equipment. Write for a Clarage catalog covering our complete line.

★



Fans for warm air furnaces, oil burners, stokers, etc.
200 to 5000 cfm.



Unit Heaters for factories, stores, offices, etc.—floor and suspended type units.
24 standard sizes

Fans for exhaust systems, pressure blowing, etc.
500 to 50,000 cfm ; built for a wide range of pressures

Air Conditioning central systems and units to solve any temperature and humidity control problem.

Hastings Air Conditioning Co., Inc.

Hastings, Nebr.

Manufacturers of

AIR CONTROL AIR CONDITIONING

Air Conditioners.
Unit Heaters.
Utility and Package Blowers.

Dealers and Representatives in Principal Cities

A Complete Line of Highly Successful COLD WATER Air Conditioners. Capacities listed depend on entering air and water temperatures.

All equipment available for combination heating and cooling.

FLOOR MODELS

Floormasters—Unusual design and special features permit maximum installation possibilities with minimum floor space and installation costs.

Air Delivery—2240 cfm. Cooling Capacity—3 to 6 tons. Dimensions—Height 93 in., Width 48 in. Depth 25 in. Motor—½ hp. Filters—3 16 in x 25 in

Royal—For offices, homes, hospitals, etc

Air Delivery—590 cfm. Cooling Capacity—1 to 2 tons. Motor—1/6 hp. Filter—1 16 in x 25 in. Dimensions—Height 40 in., Width 28 in., Depth 20½ in.

CENTRAL PLANTS

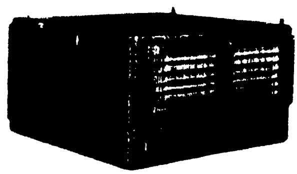
Sectional construction for ease of handling. Motors inside mounted to provide very neat appearing compact units.

SPECIFICATIONS

Size	CFM	Motor Hp	Filters	Capacity Tons
CP 30	3,000	1	5	4- 9
CP 40	4,000	1	8	6-12
CP 60	6,000	2	10	9-18
CP 80	8,000	3	12	12-24
CP120	12,000	5	20	18-36

GENERAL UTILITY MODELS

Master—Singly or in multiple are suitable for any business or space size. Large jobs handled without duct work by proper location of units.

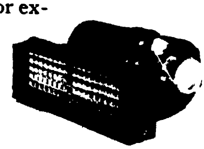


Air Delivery—2,240 cfm. Cooling Capacity—3 to 6 tons. Dimensions—Height 29 in., Width 49 in., Depth 50 in. Motor—½ hp. Filters—4 16 in. x 23 in.

Majestic—Similar to Master except size. Air Delivery—1120 cfm. Cooling Capacity—1½ to 3 tons. Motor—¼ hp. Filters—2 16 in. x 25 in. Dimensions—Height 26 in., Width 28 in., Depth 40 in.

Zephyr—Same capacity, motor and filter as the Royal. For use where suspended or concealed units are desired. Dimensions—Height 26 in., Width 24 in., Depth 28 in

UNIT HEATERS

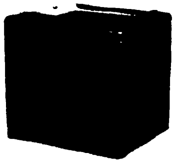


Centrifugal Type for extreme quietness and efficiency.

Steam pressure—to 150 lbs per sq in.

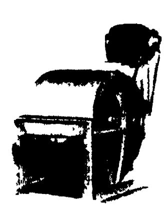
Finish—Brown wrinkle enamel and stainless steel louvers.

PACKAGE AND OPEN TYPE BLOWERS



May be knocked down for narrow doorways. Finished in attractive green wrinkle.

Utility type blowers are available with or without motors and in any discharge desired.



All sizes from 9 in to twin 21 in.
Air deliveries from 1000 cfm to 16,000 cfm.

Write for Catalogues, Literature, or Information

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AIR CONDITIONING AND COMMERCIAL REFRIGERATION DIVISIONS

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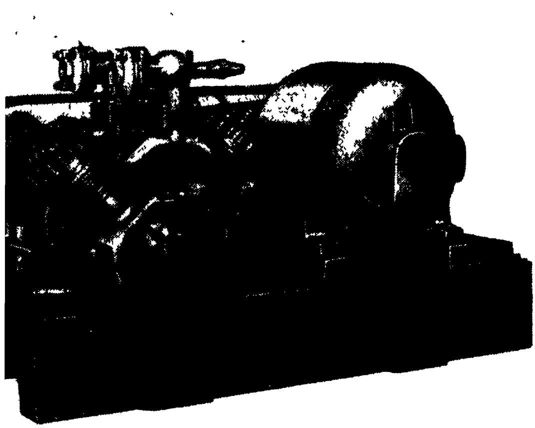
A COMPLETE LINE OF REFRIGERATION PRODUCTS WHICH ARE AVAILABLE FOR ARMY, NAVY, INDUSTRIAL AND OTHER ESSENTIAL USES



CONDENSING UNITS

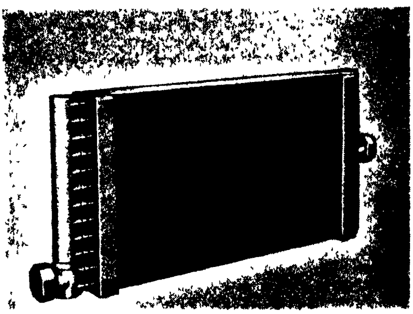
Applications: Refrigeration for industrial use, food preservation and aboard maritime commission and navy ships. Also for applications where air conditioning or refrigeration is required for processing and testing.

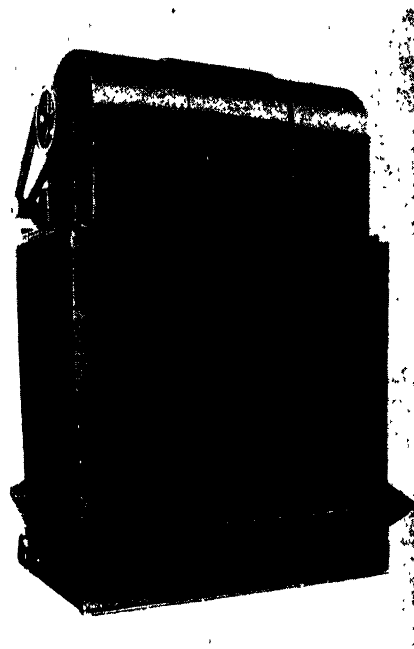
Product Data: G-E condensing units are available in a full range of standard sizes rated from 1/6 to 125 horsepower. Modified units are available which permit multistage operation to refrigerant temperatures as low as −130F.

EVAPORATIVE CONDENSERS AND EVAPORATIVE COOLERS

Applications: Condensation of vapors, such as "Freon-12" and steam; cooling of liquids . . . industrial or coolant oils, water, non-freeze mixtures, and other fluids used in industry.

Product Data: A complete line providing condensing capacities up to 100 tons of refrigeration with "Freon-12". New nested coil assembly simplifies installation and also permits quick and easy disassembly for cleaning of scale and algae.

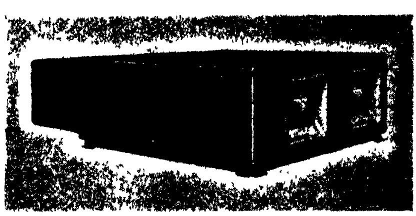




HEATING AND COOLING COILS

Applications: Ventilating systems, blast heating, industrial air conditioning, dehumidifying, refrigeration and cooling systems; special Army and Navy applications.

Product Data: Available in wide range of sizes, as allowed by current regulations, to meet requirements of installation.



INDUSTRIAL CONDITIONERS

Applications: Industrial dehumidifying, cooling, heating; special Army and Navy applications.

Product Data: Available in sizes ranging from 5 to 30 tons of refrigeration, with comparable capacities for heating service Furnished for direct expansion of "Freon-12" refrigerant, or for chilled water or cold well water. Heating coils use steam or hot water.

Niagara Blower Company

General Sales Office: 6 East 45th Street, **New York 17, N. Y.**

CHICAGO-5: 37 W Van Buren St. BUFFALO-7: 673 Ontario St. SEATTLE-4: Fourth and Cherry Bldg.

District Engineers in Principal Cities

Over 30 Years' Experience in Industrial Air Conditioning, Liquid Cooling and Air Drying

---

NIAGARA AERO HEAT EXCHANGER

For cooling industrial liquids, water, oils, solutions, chemicals, compressed air and gases, with Niagara "Balanced Wet-Bulb" temperature control to improve efficiency and obtain precise results. Patented (U. S. Nos. 2,296,946 and R. I 22,553). Ask for Bulletin 96.

NIAGARA AIR CONDITIONING SYSTEMS

For human comfort and for all industrial applications requiring controlled conditions of temperature, relative humidity, air purity and air movement.

NIAGARA AIR CONDITIONER, TYPE A

High precision apparatus using saturation to obtain control of R. H. to 1 per cent for laboratory work and control of hygroscopic materials. Ask for Bulletin 58.

NIAGARA AIR CONDITIONER, TYPE C

A year around air conditioning unit providing heating and humidifying or dehumidifying. Ask for Bulletin 80

NIAGARA FAN COOLER AND DISK FAN COOLER

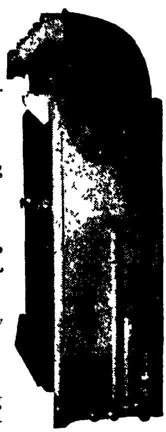
For comfort cooling, process cooling, low temperature storage for dairies, fruits, meats, food products, fur storage vaults, etc. Bulletins 72 and 78.

NIAGARA SPRAY COOLER

For all cooling applications requiring high humidity or high capacity in small space. Ask for Bulletins 72 and 78.

NIAGARA "NO FROST" SYSTEM

Using Niagara "No Frost" Liquid in spray coolers, prevents frosting of cooling coils, automatically keeps spray solution at proper concentration, gives freedom from brine troubles, corrosion. Constant, efficient operation Temperature to −100 F. Ask for Bulletins 83 and 95.



Niagara Spray Cooler

NIAGARA EXTENDED SURFACE COILS

Encased for use with heating, cooling or air conditioning systems Full range of sizes. Ask for Bulletin 92.

NIAGARA DUO-PASS AERO CONDENSER (Illustrated)

Saves power and water cost utilizing atmospheric air to remove heat of condensation Patented Duo Pass prevents scaling, saves power Ask for Bulletins 91 and 93

Concentrator for use with Niagara "No-Frost" System

NIAGARA "DUAL" COOLERS

Simultaneously cools a room and furnishes chilled water as a refrigerant. Saves equipment cost, operating expense. Patented. Ask for Bulletin 70.

NIAGARA FAN HEATERS AND DISK FAN HEATERS

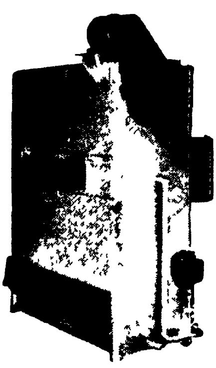
For heating and ventilating large areas. Units of the highest quality in engineering, material and workmanship. Ask for Bulletin 97.

NIAGARA AIR SUPPLY HEATER

Balances exhausted air in factories when exhaust systems are operating, saves steam and power, gives more effective heating. Patent pending. Ask for Bulletin 74.

NIAGARA MOTOR BLOWERS

One, two and three-fan units. High and low static pressure models. Ask for Bulletin 89.



Niagara Aero Condenser with Duo Pass

Parks-Cramer Company

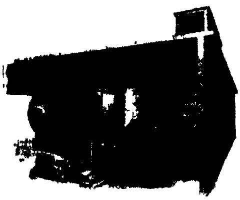
Fitchburg, Mass. **Charlotte, N. C.**

CERTIFIED CLIMATE

Complete Air Conditioning Systems including Heating, Cooling, Humidifying or De-humidifying, Air Changing, Refrigeration, Air Filtering, Air Washing

AUTOMATIC REGULATION

Merrill Process System of Hot Oil Circulation for Heating Industrial Materials



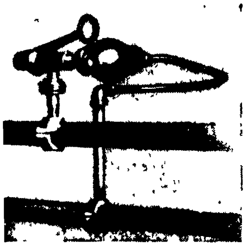
Central Station

Central Station Air Conditioning

Centrally located AIR WASHER. Proper moisture. Positive, pre-determined air removal or re-circulation. Heating coils and refrigeration optional. Helps such industries as Celluloid; Cement; Ceramics; Cereals; Cigars, Cigarettes and Tobacco, Clothing; Confectionery; Glassine; Leather; Paper and Envelopes; Printing and Lithographing; Shoes; Starch and Dextrine; Storage of Perishables; Textiles; Wood Products. Similar installations effective in Hospitals, Art Galleries, Auditoriums, Restaurants.

Air Washer or Central Station Units.

Nozzles for Central Station Air Washers



Turbomatic Humidifier

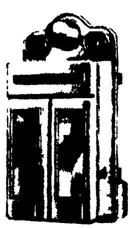
Turbomatic Humidifier

Efficient humidifier of the atomizer type. For direct humidification, as humidity boosters for Central Station systems of all makes Self-cleaning, both air and water ports Streamlined to prevent lint and dirt accumulation.

High Duty Humidifier

(not illustrated)

Water under pressure generates spray. Excess water returns to filter tank and re-circulates. Evaporation per unit high; two sizes of heads each with three sizes of nozzles give flexible capacity for varying conditions. Circulation increased by individual motor-driven fan. Spray thoroughly diffused and distributed over wide area.



Psychrostat

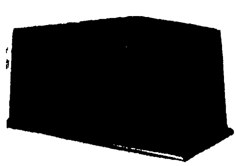
Parks Automatic Airchanger

(not illustrated)

For use with High Duty or Turbomatic Humidifiers. Insures fixed humidity and maximum evaporative cooling.

Automatic Regulation

The Psychrostat for accuracy, durability, sensitivity. Employs the principle of the Sling Psychrometer, used in all U. S. Weather Bureau Stations. Hygrostat (not illustrated) where requirements are not so exacting. An Air Conditioning System is no better than its Regulation.



Pettifogger

The Pettifogger

A compact humidifier for offices, stores, storerooms, laboratories, or other isolated departments. Self-contained in lacquered copper casing. Permanently though flexibly connected to water and electrical supplies. Automatic control. Adjustable capacity. Reduces dust. Neutralizes drying effect of heating.

WESTINGHOUSE AIR CONDITIONING EQUIPMENT

IMPORTANT FACTS ABOUT THE PRODUCT

THE IMPORTANT distinguishing feature of Westinghouse equipment is in the compressor—vital heart of any air conditioning system.

It's Hermetically-Sealed. Westinghouse developed the hermetically-sealed air conditioning compressor in 1935. It was the first, and is still the only application of the hermetically-sealed principle to multiple horsepower Freon-12 compressors. Thousands of installations have proved these advantages:

No Seal Leaks, because shaft seals are eliminated.

Installation in Unventilated Spaces. Practical because the motor is refrigerant-cooled.

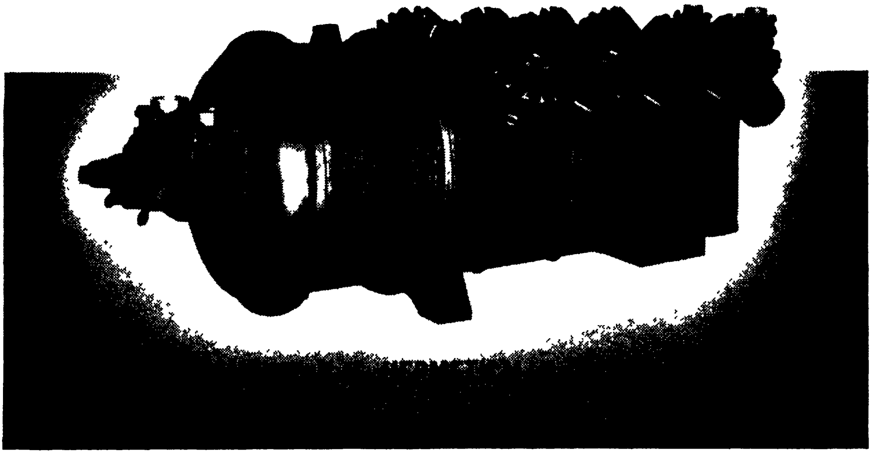
High Efficiency. Direct-drive mechanism eliminates the power loss common to belt-driven equipment.

Completely Enclosed operating mechanism. Power is sealed in, trouble is sealed out. There are no visible moving parts, yet all parts are readily accessible without dismantling, through side service plates.

Light Weight. Lightest weight per ton of cooling effect of any compressor on the market. No special foundations needed. Compact size is a natural collateral advantage.

Capacity Modulation is built in. The floating-reed design of the patented, improved head valve permits the built-in application of a capacity modulator that provides completely flexible operation of the compressor to meet exact load requirements.

Westinghouse Hermetically-Sealed Compressors are currently available in sizes from 5 hp to 100 hp.



Westinghouse Matched Equipment

For each Westinghouse Compressor, there is equipment designed and built to operate efficiently in a complete Westinghouse system that meets the specific needs of the job. This equipment includes:

Heat Transfer Surfaces

Coils for Freon, steam, hot and cold water, are available in such a wide range of sizes that any practical design specifications are readily met. The patented header of the Freon evaporator assures immediate and complete distribution of the gas over the entire coil surface.

Condensers

Water-cooled and evaporative condensers are matched to the performance ranges of each compressor. Where water is plentiful and cost low, Westinghouse Water-cooled Condensers meet every requirement. Westinghouse *Aquamiser* evaporative condensers combine the most desirable features of water-cooled condensers, air-cooled condensers and cooling towers.

Air Conditioning Units

Vertical and horizontal units, for air conditioning and refrigeration, are provided in sizes to meet all needs. The cabinets are complete with quiet, blower-type fans, and facilities for installation of cooling and heating coils, filters and humidifiers.

Factory-Built Systems

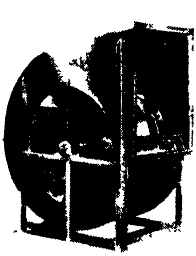
Factory-built complete within the space and central-plant systems known as the *Unitaires* are available in eight sizes—from 2 to 25 hp. For requirements within these ranges they offer ease of handling, quick installation, low cost. They combine the complete refrigeration cycle, controls and blowers within a single, compact, easily installed cabinet.

United States Air Conditioning Corporation

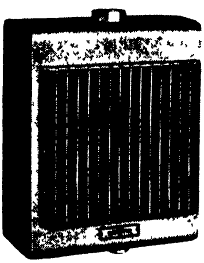
Heating, Cooling, Ventilating and Air Conditioning Equipment

For Industrial, Commercial and Residential Applications

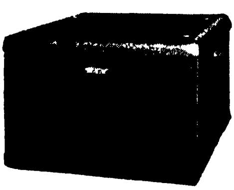
General Offices and Factory: Northwestern Terminal,
Minneapolis, Minn.



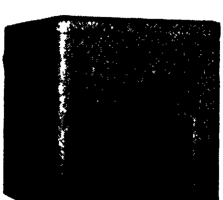
Type A Backwardly Curved Blade



Unit Heater



Coil Cooling Unit



Blower-Filter Unit

USAirCo Blowers
Heavy and light duty blowers, single or double inlet, in sizes and capacities for any heating, cooling, ventilating and air conditioning application.

USAirCo Air Washers
Single, double or triple stage 2,500 to 100,000 cfm for cleansing, cooling by cold water or refrigerant, humidifying or dehumidifying.

USAirCo Heating Units
Suspended types with Deflecto diffusing grilles. Floor or wall type blower heaters Sizes and types for every heating need.

USAirCo Cooling or Heating Cores
Five standard series for central station heating or cooling applications.

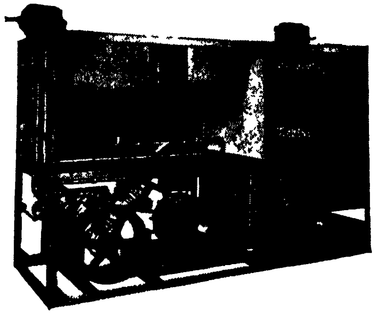
USAirCo Cooling Units
Suspended type for cold water or direct expansion applications.

USAirCo Blower Filters
Complete assemblies for warm-air furnace applications.

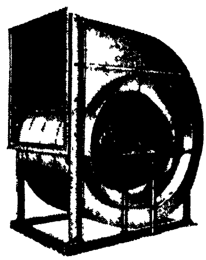
Kooler-aire Package Units
Complete self-contained units for refrigerative, cold water and evaporative cooling. Also room coolers and humidifiers.

USAirCo Deflecto Grilles
Patented diffusing grilles for controlled directional distribution of air.

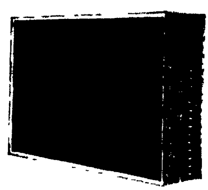
Write for Latest USAirCo Catalog



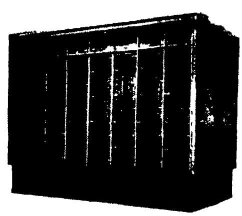
Refrigerated Kooleraire



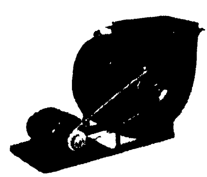
Type FC Forward Curved Blade



Coil Cooling and Heating Unit



Air Washer



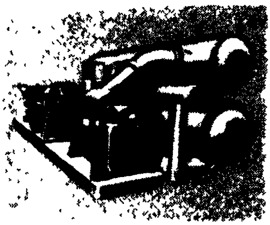
Type E Single Inlet Light Duty

York Corporation

York, Pennsylvania

Factory Branches and Distributor Engineering and Sales Offices throughout the World.

Air Conditioning and Refrigeration for maintaining proper atmospheric conditions for industrial processes essential to the war. Installations of unit and central systems in a complete range of capacities and types for every design requirement.

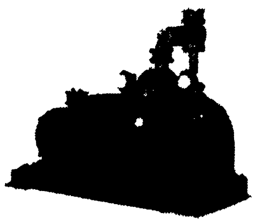


York Turbo Compressor

Condensing and Water Cooling Systems—Centrifugal brine and water cooling systems available over wide range of capacities up to 1500 tons refrigeration, steam turbine or motor drive.

Self-contained dynamically balanced, non-vibrating V/W type reciprocating compressors available in capacities up to 350 tons refrigeration in a single unit, with water cooled or economizer type condensers.

Efficient automatic capacity reduction available for economical operation at reduced load.



York V-W Condensing Unit

The York Economizer—A combined forced-draft cooling tower and refrigerant condenser, is available for installations where prohibitive water costs or inadequate drainage facilities preclude the use of a water cooled condenser. Standard factory constructed and built-up units may be used singly or in multiple for applications of any specified capacity. Economizers for use with Freon as the refrigerant are furnished, as standard, with a liquid sub-cooling coil. Economizers also designed for cooling of quench oil and other liquid coolants.

York Sectional Economizer

Air Conditioning Units: A complete line of finned coil, dry coil, wetted surface and spray type sectional air conditioners for horizontal or vertical applications, designed to facilitate installation and the distribution of air. Standard units can be equipped with by-pass feature and arranged for cooling and dehumidifying, heating and humidifying, for year-round processing.

Yorkaire 550 Unit Air Conditioner—A compact, self-contained model occupying but 21 x 42 inches of floor space and requiring only water, drain and electrical connections to operate. Special features provide utmost flexibility to meet varying conditions. Temperature dial control provides both automatic and manual temperature control. Air volume and motion may also be adjusted by a special control and the directional grille provides *directed air flow*—up, down or from side to side. May be used with ducts if desired.

The Yorkaire 550 is ruggedly built, quiet in operation, equipped with standard fan and compressor motors for AC or DC.

Yorkaire 550 Unit Air Conditioner

Dehumidifiers—For central station systems where a large volume of air is to be handled and where control of humidity is an essential requirement, the York dehumidifier is especially applicable. Construction features insure a minimum space demand and maximum performance conditions. Standard washers are available in a full range of capacities for industrial installation.

American Foundry and Furnace Co.

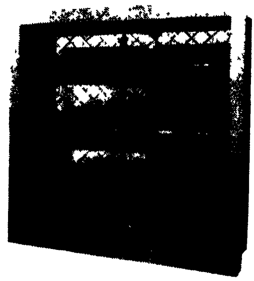
Sales Offices in Principal Cities

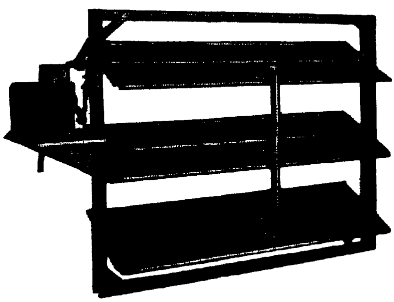
P. O. Box 904, Bloomington, Ill.

AIR CONTROLLING SPECIALTIES

KD Grille and Fire Damper

At the right is shown a Type KD Grille and Fire Damper consisting of a wire grille, with an angle iron frame and eight inch steel sleeve, and a ball bearing louver damper, with pull chain and two fusible links. The steel sleeve is made to fit the unit into a wall opening. The chain operator works through two pulleys and is held in place by a claw on the damper frame. The fusible links melt at 160 deg., thus allowing the damper to close by its own weight in case of fire.

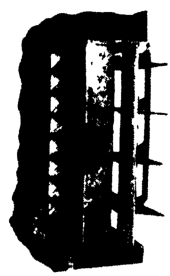




F-12 Louver Damper

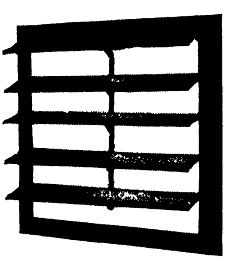
At the left is shown a Type F-12 Louver Damper. These dampers are made on order to fit any size opening. Blades are 16 gauge steel, frames are 2 in. x ½ in. x ⅛ in. channel. Ball bearings are standard, with steel trunnion bearings optional. Motor, linkage and bracket shown are not standard but may be ordered as an extra.

S-454-F Radio Range Stormproof Louver Combination



Consists of a galvanized iron frame with 26-gauge galvanized iron stationary horizontal stormproof louver blades, riveted securely to outside frame and all built into outside wall (fits 8-in. thick wall). Apron extends over sill. Back of stormproof louver is a No. 16 mesh screen.

Back of screen is a multiple-blade, ball bearing louver damper (similar to F-12 but with off-center axle) to control volume of air admitted. Louver damper blades of 16-gauge steel, galvanized. Frame of 2 x ½ x ⅛-in. galvanized channel iron. Dampers can be automatically or manually controlled. Blades all work in unison.



FL-Aluminum Automatic Fan Outlet Louver

These louvers are for use where the air from a fan discharges into atmosphere and the velocity of the discharged air is less than 2000 fpm. Aluminum blades, working in unison, open from fan pressure and close automatically when fan is not in operation. Frames of angle iron drilled for fastening to wall, duct or penthouse. Standard finish of frame—black or gray enamel or prime coat. Regularly made in 14 sizes from 12 x 12 in. to 50 x 50 in. *Special sizes on request.*

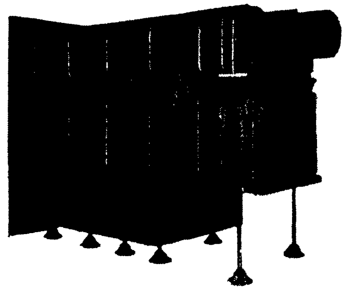
HEAVY DUTY HORIZONTAL HEATERS

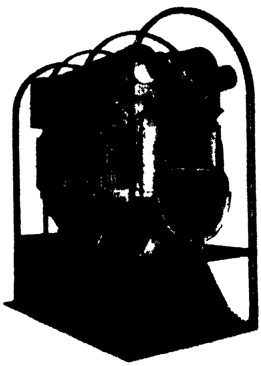
CENTRAL PLANT SYSTEMS

American SUPERFIN Heaters are made entirely of cast iron and designed for use with blower, air filters, humidifier, etc., in Central Plant Winter Air Conditioning Systems.

Fireboxes and top arches are finned heavily to provide maximum durability and furnish largest possible heating surfaces. Design is such that all air is forced over hottest surfaces of the heaters at all times.

Finned, pear-shaped radiators, streamlined, add enormously to the radiating surface of the heaters, thus adding to their efficiency.





UNIT HEATER SYSTEMS

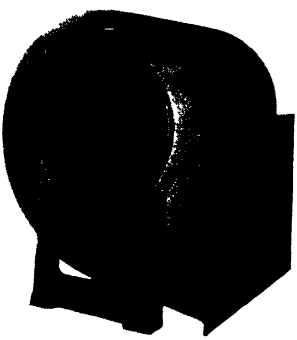
American Gas or Oil Fired Unit Heaters have been designed for use in commercial and industrial applications. These units have the multiple blowers underneath the heating section. Air is drawn into the heater at the floor line, then passes over the heater, and may either re-enter the room from the top of the heater or at the floor line on the opposite side of the heater.

Heater section is made entirely of cast iron and is designed to use either oil or gas as a fuel.

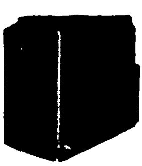
SUPERIOR BLOWERS

At the right is shown a standard Superior Blower. Superior blowers are all that their name implies. Rugged design, quiet operation of mechanical parts, freedom from vibration, have been stressed throughout the construction of the entire line. Made in wheel sizes ranging from 10 to 65 in., inclusive, having wheel diameters of corresponding values at 5-in. intervals, with 12 and 18-in. sizes in addition.

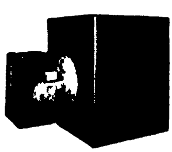
Each size blower is available in single or double width, single or double inlet, any discharge arrangement. Deliveries range from 800 to 105,000 c.f.m., with uniform capacity ranges between.



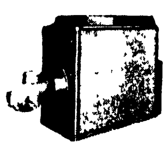
DOMESTIC HEATING EQUIPMENT



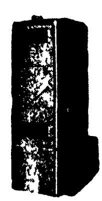
June-Aire Gas Fired



June-Aire Oil Fired



June-Aire Oil Fired



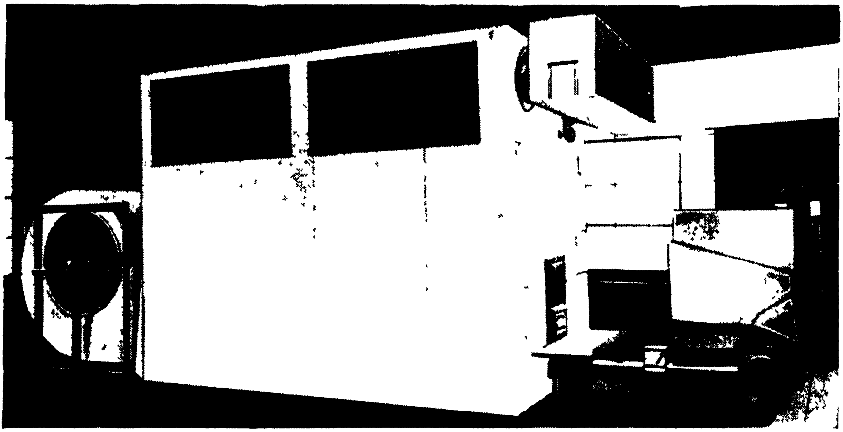
June-Aire Vertical Gas Fired

Campbell Heating Company

31st and Dean, **Des Moines, Iowa**

SUMMER and WINTER AIR CONDITIONING

Industrial, Commercial—Institutions, Residences



Campbell Heater in factory of National Mfg & Stamping Co Des Moines Iowa

The Campbell Heater shown above is installed in a two-story factory building 80 x 120 ft. with 12 ft. ceiling height on each floor There are no ducts—this heater sends heat to all parts of the building without ducts. On a zero day thermometers near four corners of the first floor varied only from 59 to 62 deg —the north west corner was 60 deg. When warm air is desired to remote rooms it can be provided by a properly designed duct system.

This heating performance was accomplished in spite of the fact that the heater is in the south east corner of the building. Heated air traveling to the north end of the room must overcome the obstruction to air flow caused by the timbers above—the air must flow across the timbers, not with them.

In this installation the heater rests upon the floor of the room to be heated Where floor space is limited and valuable the blower can be mounted on top of the heater or put under it by raising the heater, provided the ceiling height is sufficient.

Five Advantages of Campbell Heaters

1. **Easy to Install**—In most cases the Campbell Heater can be put into operation within 72 hours after delivery
2. **Especially Adaptable to Ductless Convection Heating**—Elimination of ducts reduces cost in heating a single large room
3. **Compact in Design**—Over-all dimensions can be adapted to meet any reasonable limitations of floor space or ceiling height
4. **Efficient with any Type of Fuel**—Especially adaptable to stoker firing, but operates equally well with fuel oil or gas.
5. **Low in Cost**—Because of standardized design and efficient manufacturing methods, Campbell Heaters are low in cost.

Campbell Heaters are guaranteed to deliver full rated capacity.

CAMPBELL "WINTER-CHASER" AIR CONDITIONING SYSTEM

The Campbell "Winter-Chaser" System provides all the essentials of winter air conditioning: Simultaneous control of temperature, humidity, air circulation and air cleanliness, besides providing fresh air for ventilation, quick heating, flexibility; and a summer cooling effect Campbell equipment is built of the best materials obtainable, and has been developed through over sixty years of experience. The system is designed by competent experienced engineers and installed by experienced mechanics. It is guaranteed as to results and for 10 years as to durability.

Advantages of Campbell "Winter-Chaser" Design

Compact design, requiring minimum space—the heater can be placed readily in any ordinary boiler or furnace room

Efficient radiating surface—permits an immense amount of radiating surface in a small space. Ratio of prime radiating surface to grate area is more than 30 to 1. This produces very low flue gas temperature with consequent high efficiency

Vertical self-cleaning radiating surfaces constitute most of the heating surface—hence are not insulated with soot and ashes.

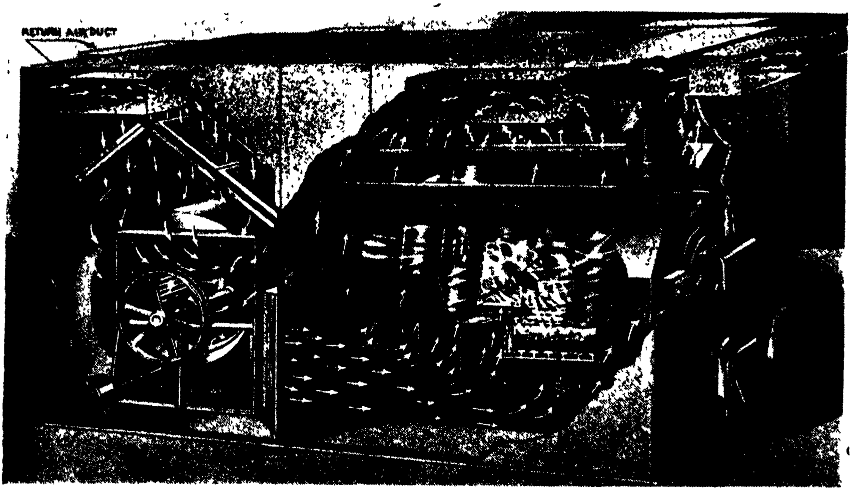
Vertical air-warming tubes completely surround the fire, subjecting greater prime heating surface to direct contact with the fire—less liable to be overheated

Extra large combustion space above grate permits smoke and air to mix and burn before striking heating surface—less unburned gases escape up the chimney.

Long return smoke travel—the tubes form a dividing wall each side of the fire, separated in the back to let smoke pass through behind the tubes and to the front of heater. This assures long contact with heating surfaces.

Two convenient cleanouts permit easy cleaning of the smoke passages without putting out the fire.

Basement heating on the same level as the heater, is solved by this Campbell System. Entire capacity of heater can be forced into the basement and thus heat a large room in a few minutes.

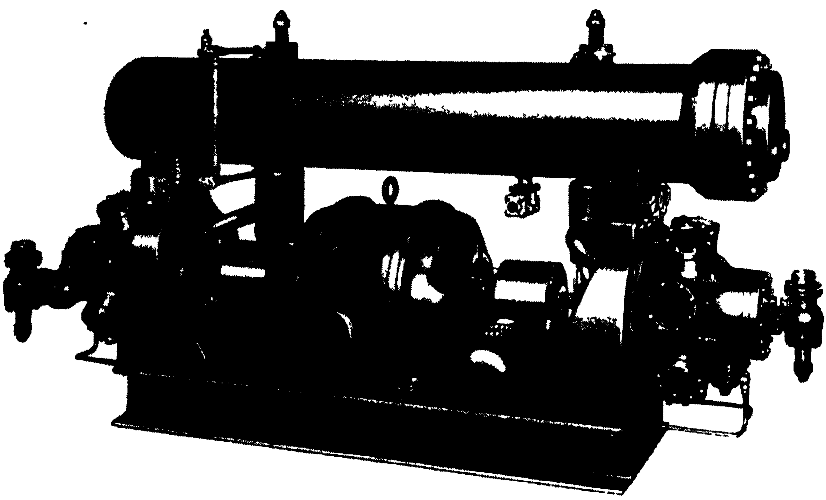


Furnace Number	Sq Ft Grate	Heating Surface	For Building Heat Loss Btu	Maximum Capacity Btu	Blower CFM	Size Motor	Dimensions Casing	Addnl Space for Blower	Approx. Shipping Weight
8075	7½	280	483,000	725,000	8850	¾	75 x 80— 88" high	54"	4500
8100	10	320	596,000	893,000	10900	1	75 x 93— 96" "	60"	5500
8125	12½	360	720,000	1,080,000	13200	1½	75 x 105— 96" "	66"	6500
8150	15	440	850,000	1,275,000	15600	2	75 x 118—102" "	66"	7500
8175	17½	480	960,000	1,440,000	17500	2	75 x 130—102" "	72"	9000
8200	20	600	1,150,000	1,725,000	21100	2	94 x 137—120" "	76"	10000
8250	25	750	1,440,000	2,160,000	26400	3	94 x 157—120" "	76"	12000

Unit includes furnace, casing, blower, motor, V-flat drive.

CHRYSLER AIRTEMP

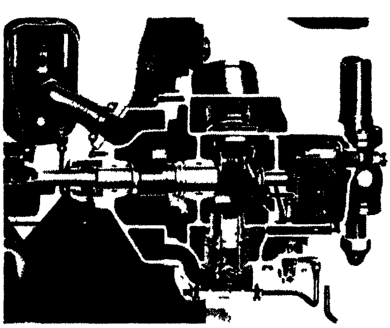
AIRTEMP DIVISION OF CHRYSLER CORPORATION, DAYTON, OHIO



14 CYLINDER MODEL AIRTEMP RADIAL CONDENSING UNITS— Available in 10 to 75 Horsepower Capacities

This heavy-duty radial compressor for use with Freon is especially adapted for refrigeration, for industrial processes or air conditioning. Airtemp radial compressors are directly connected and have force-feed lubrication. The automatic starting unloader and automatic capacity-reduction unloader give high operating efficiency. Light in weight and economical to operate, these compressors are shipped ready to run. They are especially easy to install since vibration is practically eliminated and no special foundations are necessary.

- AUTOMATIC CAPACITY REGULATION
- UNLOADED STARTING
- DIRECT CONNECTED



- SIMPLIFIED INSTALLATION
- COMPACT DESIGN
- PRACTICALLY NO VIBRATION
- NO SPECIAL FOUNDATIONS NEEDED
- INTERCHANGEABLE PARTS
- LONG LIFE
- ECONOMY

AUTOMATIC UNLOADER

This automatic cylinder unloading device permits starting the compressor under no load and keeps the compressor automatically adjusted to varying loads with no stopping and starting during operation.

CHRYSLER AIRTEMP

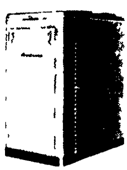
AIRTEMP DIVISION OF CHRYSLER CORPORATION, DAYTON, OHIO

VAPORIZING OIL-BURNING WINTER AIR CONDITIONER

55,000 Btu output. Completely automatic. Models for gravity—forced-air or complete winter air conditioning. Sure-Draft fan assures highest overall efficiency. Bonderized and insulated jacket. Approved for closet installation, Underwriters Laboratories, Inc.

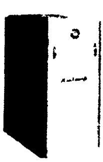
OIL-FIRED WINTER AIR CONDITIONER

Heats, humidifies, filters and circulates the air. Five models, from 70,000 to 160,000 Btu output. "Bonderized" and insulated jacket. Metal combustion chamber, seam-welded firebox of copper-bearing steel; large, slow-speed, rubber-mounted fan. Airtemp conventional or Twin Airflow oil burners on larger models.



GAS-FIRED WINTER AIR CONDITIONER

Heats, humidifies, filters and circulates the air. Steel-firebox models from 70,000 to 130,000 Btu output. Cast-iron firebox models from 50,000 to 200,000 Btu output. "Bonderized" and insulated jacket. The Airtemp "Silent Flame" Gas Burner starts, stops and operates quietly, has many exclusive features—no popping or flash-backs. Approved, A.G.A. Laboratories.

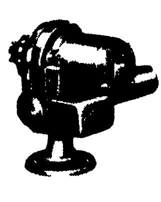


PERCOLATOR BOILERS—OIL- OR GAS-FIRED

Three oil-fired models from 400 to 1700 E.D.R. (steam). Three gas-fired models from 400 to 1050 E.D.R. (steam) Fire chamber surrounded with water on all sides and bottom. High efficiency and faster heating result from "percolator" principle. Steam or hot water. "Bonderized" and insulated jacket.

COAL FIRED FURNACES—FORCED-AIR—GRAVITY

Forced-air models in cast iron and steel. Sizes from 226 to 346 sq in. grate area in steel—and from 258 to 384 sq in. grate area in cast iron. Oversized blower motor mounted in rubber, has automatic overload and low-voltage protection. Gravity models in cast iron and steel—sizes from 226 to 452 sq in. grate area, steel; 204 to 452 sq in. grate area, cast iron.



OIL BURNERS

Model A-8: A conventional oil burner, .75 to 1.35 gallons No. 3 furnace oil per hour.

Model B-9: The exclusive Airtemp Twin Airflow Oil Burner, 1.35 to 3.0 gallons No. 3 furnace oil per hour. Air for combustion is furnished by two fans and adjusted at the outlet, not the inlet of the fans.

Model B-10 and C-10: A conventional pressure-atomizing oil burner, 1.35 to 4.5 gallons No. 3 furnace oil per hour.

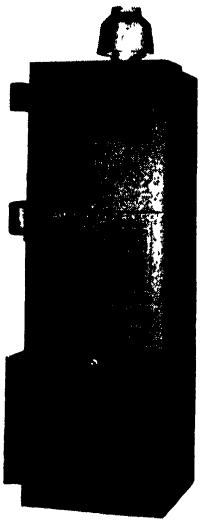
All Airtemp Oil Burners are approved by Underwriters' Laboratories, Inc., and bear the seal of the Official Inspection Agency of the Oil Burner Industry as evidencing compliance with commercial Standard CS75-39, as issued by the National Bureau of Standards of the U. S. Department of Commerce.

Let the pup be furnaceman

THE BRYANT HEATER COMPANY

17825 St. Clair Avenue - - - Cleveland, Ohio

Engineering, Sales and Installation information on Bryant Equipment available through Bryant Distributors, Dealers and Gas Companies in principal cities.



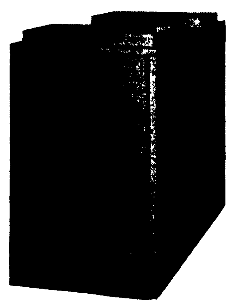
Vertical Winter Air Conditioner

Gas-Fired Boiler

Bryant Gas designed boilers include tubular cast iron sections, ribbed lower tubes, large steam liberating areas, all heating surfaces readily accessible for cleaning. Insulated metal jacketed covers and Bryant gas controls. Complete range of *AGA* inputs from 45,000 to 3,996,000 Btu/hr for steam and hot water heating systems, volume water heating and industrial process.

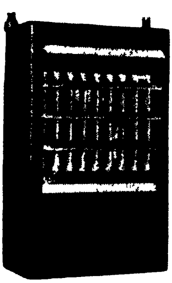
Bryant Vertical Winter Air Conditioners complete with blowers, humidifier and filters are compactly designed for small housing, office and industrial use. Bryant tubular cast iron section design and quality controls are standard equipment. Capacities range from 55,000 to 115,000 Btu/hr *AGA* inputs.

Complete line of forced Warm Air Gas-Fired equipment from 60,000 to 750,000 Btu/hr *AGA* inputs. Efficient cast iron heating sections of vertical tubular construction and large capacity blowers are featured. Humidifiers, filters and Bryant Automatic controls are standard equipment.



Forced Warm Air Gas-Fired Equipment

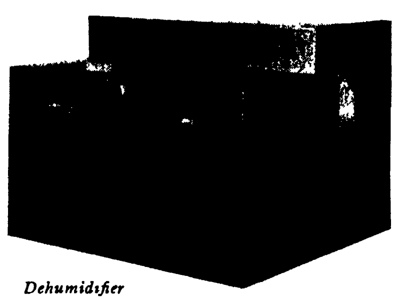
Bryant suspended type Gas-Fired Unit Heaters available in five sizes ranging from 65,000 to 255,000 Btu/hr *AGA* inputs. Efficient heat exchange of staggered vertical tube construction. Available in both cast iron combustion chamber, alloy steel tube and all steel types. Quick, clean, efficient heat for all types of industrial and commercial space. Flexible, automatic control and large volume air circulation produce ideal space heating results.



Suspended Type Gas-Fired Unit Heater

Bryant Dehumidifiers with rotary silica gel bed and completely automatic control find new demands for food dehydration, powder drying, in the manufacturing of airplane valves, telescopic sights and signal flares—in the testing of airplane engines—and the storage of engine parts, bomb sights and submarine parts. Especially adaptable to industrial requirements and all types of air drying installation.

See your local Bryant Distributor or write for complete details and specifications.



Dehumidifier

THE MEYER FURNACE COMPANY

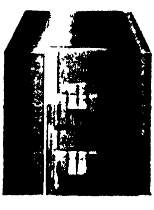
PEORIA, ILLINOIS

Manufacturers of Heating and Air Conditioning Equipment for Coal, Gas and Oil Burning

Branches and Distributors

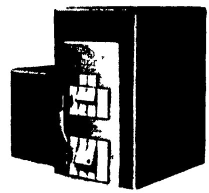
ALBUQUERQUE, N. M.
CHICAGO, ILL.
COLUMBUS, O.
DES MOINES, IA.
FLORENCE, S. C.
GREEN BAY, WIS.
KANSAS CITY, MO.
NEW YORK, N. Y.
OMAHA, NEBR.
PHILADELPHIA, PA.
PITTSBURGH, PA.

WEIR and MEYER Steel Warm-Air Furnaces, of welded-and-riveted gas-tight construction, have a 60 year reputation for efficiency, dependability and durability. Service in hundreds of military establishments both here and abroad has again attested to their satisfactory performance. They are available for small and large requirements and for all fuels in a wide variety of firing applications.

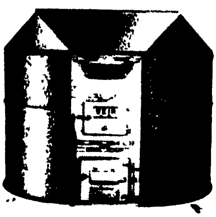


U Series Gravity

The U SERIES WEIR Hand-Fired Coal Furnace embodies a new construction principle (patent applied for) which, when combined with its other time-tested Weir features, provides an outstanding heater. The gravity furnace, as shown on the left, ranges in register capacity from 50,000 to 170,000 Btu per hour. The rectangular-cased forced-air furnace, shown on the right, ranges from 50,000 to 250,000 Btu per hour output at the register. This series may be stoker-fired, though other Weir stoker-designed furnaces are available.

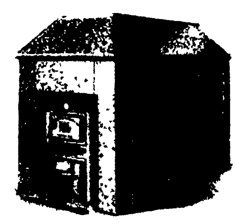


U Series F-A

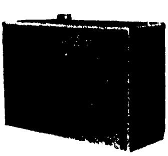


Heavy Duty 38-R

WEIR Heavy-Duty Furnaces, consisting of the R Series on the left and the 500 Series on the right, are ideal for industrial and commercial service and for schools, churches and other large spaces. Capacities from 300,000 to 1,500,000 Btu per hour. Designed only for forced-air circulation. Suitable for firing with coal by hand or stoker, with gas or with oil.



Heavy Duty 544B



Oil Fired A-100-M

The WEIR Oil-Fired Air Conditioner does a complete job of winter air conditioning. Designed for oil fuel and forced circulation. Completely self-contained in low compact casing enclosing burner and blower as well as heater and all controls, yet with everything easily accessible.

The MEYER Gas-Fired Air Conditioner automatically provides completely controlled winter air conditioning. Efficient performance, compact design, modern appearance. Heavy gauge welded steel heating element; die-formed casing. **A.G.A.** approved.

Gas Fired F-10

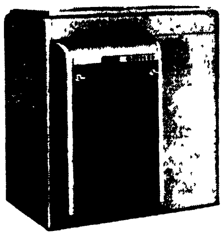
For any warm air heating or drying problem employing the use of any fuel, call our nearest office or representative or write to the home office.

L. J. Mueller Furnace Co.

ESTABLISHED 1857

2009 W. Oklahoma Ave., **Milwaukee 7, Wis.**

Mueller Climatrol Gas-Fired Equipment



Series "EPS"

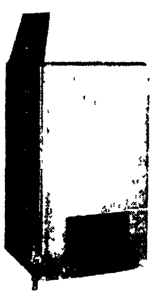
This winter air conditioner is available in three sizes, with A.G.A. input ratings from 90,000 to 180,000 Btu. Each size is available with various size blowers in accordance with individual air delivery requirements.

Series "SHP"

Winter air conditioner. May be installed in basement or utility room. Available in three sizes, with A.G A. input ratings from 60,000 to 100,000 Btu per hour. Provides wide range of air deliveries to meet specific needs.

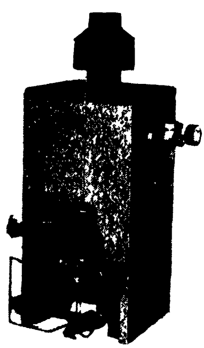
Series "CVP"

An all-cast-iron winter air conditioner for basement or utility room installation. Furnished in one size, with A G A. input rating of 100,000 Btu per hour. Adjustable motor sheave permits variation in air deliveries.



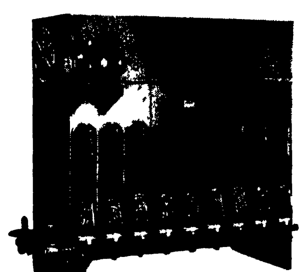
Series "G"

A gas-fired gravity unit for home applications. Made in two casing styles, i.e., square, as shown, and round. Furnished in two sizes, with A.G.A. input ratings of 90,000 and 135,000 Btu per hour.



Gas Era Boiler

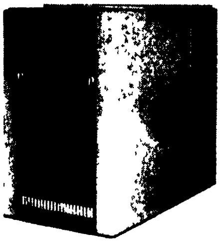
Series "AE" (shown) and "A" have A.G.A. ratings from 180 to 1,260 sq. ft. steam, and 290 to 2,015 sq. ft. hot water. "C" boilers from 420 to 12,600 sq. ft. steam, 670 to 20,100 sq. ft. hot water.



Unit Heater

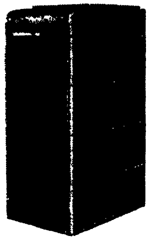
For large space-heating requirements, and industrial applications. Units are available in 45 sizes, A.G.A. approved, from a 4-section unit with input of 180,000 Btu per hour, to the 48-section size, with input of 2,160,000 Btu per hour.

Mueller Climatrol Oil-Fired and Coal-Fired Units



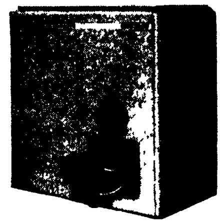
Series "50"

Oil-fired winter air conditioner. Available with Mueller pressure atomizing, or vaporizing burner. Three sizes, 100,000 to 225,000 Btu.



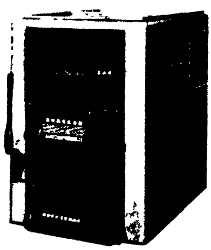
Series "OVP"

Oil-fired winter air conditioner. Equipped with Mueller Vaporizing Oil Burner. One size, with maximum capacity of 80,000 Btu at bonnet.



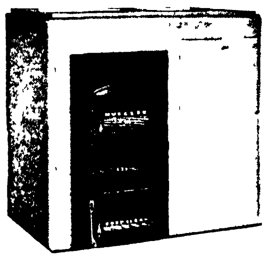
Series "OHP"

Horizontal type oil-fired winter air conditioner. Equipped with Mueller vaporizing burner. One size, 80,000 maximum at bonnet.



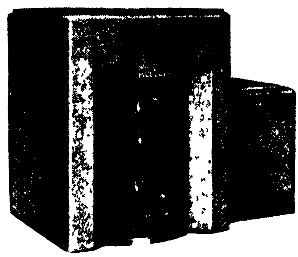
Series "FB"

Cast-iron, coal-fired winter air conditioner. Six sizes, with capacities at register from 69,000 to 199,000 Btu.



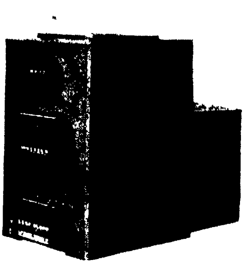
Series "P-400"

Steel, coal-fired winter air conditioner. 4 sizes, 20 in. to 27 in. drums, with capacities at register ranging from 79,000 to 141,500 Btu per hour.



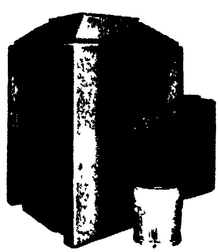
Series "AP"

Cast-iron, coal-fired winter air conditioner. Six sizes, 20 in. to 33 in. firepots, with forced air capacities ranging from 93,000 to 269,000 Btu.



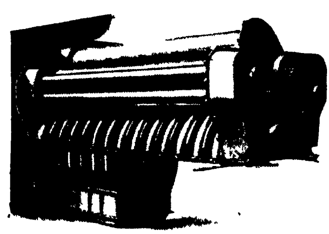
Series "WR"

Cast-iron, coal-fired furnace. Two styles, i.e., forced air and gravity. One size only. Forced air capacity, 72,000 Btu. Gravity, 42,000 Btu.



Stoker Furnace

Winter air conditioner. Any stoker may be used. Cast-iron heating unit. Available in two sizes, with capacities of 110,000 and 175,000 Btu.



Horizontal Tubular

Adaptable for schools, churches and other large buildings. Three sizes, with capacity range from 1,188,000 to 1,390,000 Btu per hour. For gravity or forced air applications.

Lee Engineering Company

Union National Bank Bldg., **Youngstown, Ohio**

LEE DIRECT WARM AIR HEATING

The Lee System of warm air heating generally costs less to install than steam or hot water; utilizes fuel with a high degree of efficiency; distributes the heat exactly where needed; responds promptly without lag; requires little or no maintenance; and needs no licensed attendant. Heaters for use with the Lee System are made in the four types illustrated and described briefly below.

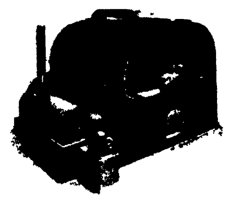
BRICK-SET TUBULAR HEATER

For use with central heating system in connection with duct distribution. Single heater capacities from 2,800,000 Btu per hour to 8,000,000 Btu per hour. Two heaters, installed as a battery serving as one unit, provide capacities over 10,000,000 Btu per hour.

Brick Set Tubular Heater

STEEL ENCASED TUBULAR HEATER

For use with central heating systems in connection with duct distribution over a capacity range of from 2,000,000 Btu per hour to 6,000,000 Btu per hour. Heater may be installed in heated area without enclosure, requires no foundation, and may be moved from one location to another by taking unit apart and reassembling.



Steel Encased Tubular Heater

TUBULAR UNIT HEATER

For use either as a central system in connection with duct distribution or with adjustable outlet nozzles as a unit heater. Capacity range from 2,000,000 Btu to 6,000,000 Btu per hour. In sizes up to 4,000,000 Btu heater is shipped as a completely assembled unit with all but mechanical equipment, refractory lining and controls in place. Heaters require no foundation and are equipped with crane hooks so that they may be moved from one location to another.

Tubular Unit Heater

SHELL UNIT HEATER

For use with or without distributing duct system. Heaters have capacity range of from 400,000 Btu to 2,000,000 Btu per hour. Available for either stoker or hand firing. All units are shipped completely assembled, wired and ready for operation. Stoker-fired units are shipped assembled with all parts in place except mechanical equipment, refractory lining and controls. Available in both hand and stoker fired models.

Shell Unit Heater

For further information write for Catalog HV-44.

National Heater Co.

401 Essex Bldg., **Minneapolis 2, Minn.**

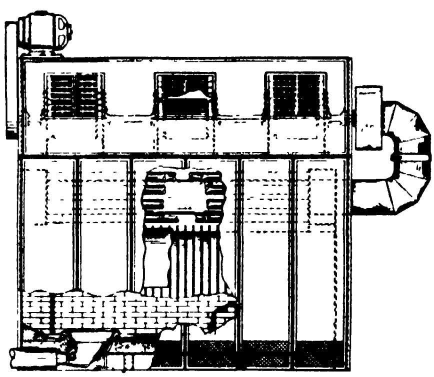
NATIONAL CHAMPION

STEEL S-P-A-C-E HEATERS

OIL—GAS—STOKER HAND FIRED UNITS

As shown in diagram these heaters are built as a complete unit consisting of heater, casing, firing equipment, multiple blower assembly, motor, drive, induced draft blower and controls. NATIONAL HEATERS are constructed of welded copper bearing hot rolled one quarter inch boiler plate steel instead of the usual lighter ten guage plate. This provides positive assurance of life-long gas and smoke tight operation with ample provision for absorbing all strains resulting from expansion and contraction.

The body of the heat generator is long and narrow in design, allowing complete fuel consumption with properly spaced radiating fins to facilitate maximum heat transmission from this primary heating surface.



640,000 to 4,000,000 Btu's

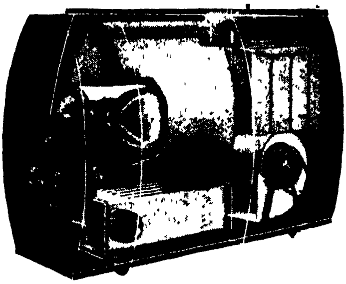
Casings are sturdy with insulated double metal walls to reduce wasteful radiation to a minimum. Panel type construction with rigid angle iron frames and structural iron reinforcement at both top and bottom assures a strong durable installation. Casings are designed for quick and convenient assembly.

ENGINEERING RATING CHART

HEATER NUMBER	BTU OUTPUT CAPACITY	1/2" S P C.F M MAX	GRATE AREA SQ FT	SQ. FT HEATING SURFACE	VOLUME CU FT INSIDE HEATER	BLOWER MAXIMUM	BLOWER MOTOR		TUBES		APPROXIMATE WEIGHT LESS STOKER
							HP MAX	RPM	NO	DIA	
300	640,000	8,200	6 5	320	75	16'TWIN	3	1750	2	8"	8,850
350	860,000	11,000	8 0	430	100	18"TWIN	5	1750	2	8"	14,420
500	1,215,000	18,150	12 0	600	130	21"TWIN	7 5	1750	10	4"	15,860
550	2,000,000	32,000	14 7	844	175	21'TRIPLE	10	1750	12	4"	18,610
600	2,500,000	45,000	16 0	1421	315	27"TRIPLE	15	1750	16	4'	20,850
650	3,000,000	52,000	20 0	1685	375	30"TRIPLE	20	1750	16	4"	24,090
700	3,500,000	56,000	24 0	1810	450	30"TRIPLE	25	1750	18	4"	26,340
750	4,000,000	60,000	28 00	1935	450	30"TRIPLE	30	1750	20	4"	28,470

Gar Wood Industries, Inc.
Heating Division

7924 Riopelle Street
Detroit 11, Michigan

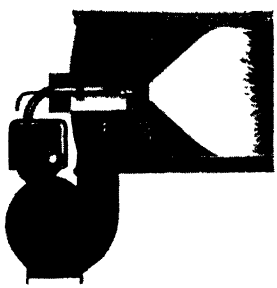


Tempered-Aire Unit

TEMPERED-AIRE UNIT

A high efficiency, direct fired heating unit incorporating a large-volume firebox, an integral economizer, a coordinated oil burner combustion chamber firing unit, a flash-type humidifier, washable cloth filters, and a low speed, resilient-mounted blower.

The primary transfer of heat occurs in the large firebox, having its outlet at the bottom, through which the hot gases pass down into the economizer. Here the gases divide into long thin slices, within the economizer tubes. Each tube is swept at high velocity by cold return air from the blower. The rapidly flowing cool air absorbs a maximum of heat from the economizer surfaces.



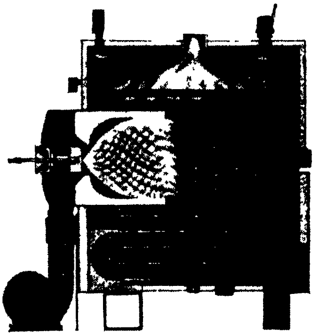
Oil Burner Combustion Chamber Firing Unit

TYPE "O" OIL FIRING UNIT

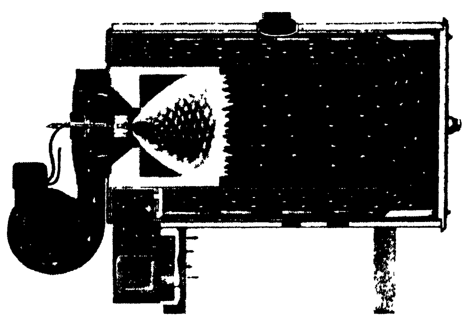
The burner and fire bowl combustion chamber are built together as an actual unit, with the back end of the fire bowl forming a windbox containing air for combustion at sufficient pressure to offset the effect of draft variations. Air from the windbox passes through metering chutes, set at an angle to cause rotation. Both air and oil rotate and intermingle in a conical spray, resulting in a definitely controlled flame entirely contained within the fire bowl.

OIL-FIRED Boiler-Burner UNITS

An internally fired, downdraft, high efficiency Boiler-Burner Unit employing the same firing unit used in the Tempered-Aire Furnace-Burner Unit.



Steam Type Boiler-Burner Unit



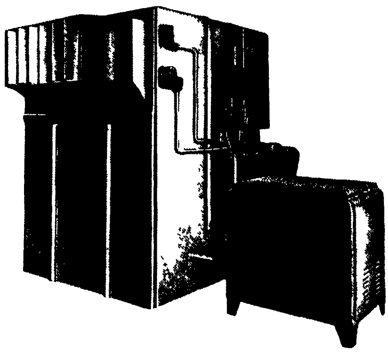
Hot Water Type Boiler-Burner Unit

The oil-fired Boiler-Burner Units are made in two types, an obround design having a steam chest for steam systems and a cylindrical design for hot water systems. The steam type is available with a built-in tankless water heater. The hot water type can be supplied with a domestic water storage tank having an integral water heater all contained within the boiler jacket.

Airtherm Manufacturing Company

728 S. Spring Ave.
St. Louis 10, Mo.

THE ENGINEERED LINE OF UNIT HEATERS



AIRTHERM
Direct-Fired
UNIT HEATERS

HEAT LARGE AND SMALL PLANTS ECONOMICALLY

They require no costly pipe or duct work and can be installed in a few hours. The heat is evenly distributed and thrown out as far as 200 feet. About 80 per cent of the fuel energy is utilized.

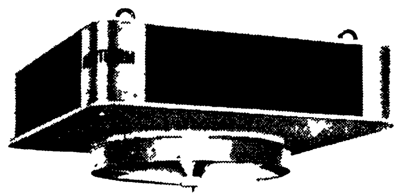
Airtherm Direct-Fired Unit Heaters do not require any maintenance. All you need do is set the automatic controls.

For coal, gas, and oil.

Made in 6 sizes (300,000--1,500,000 Btu).

ENGINEERING SERVICE

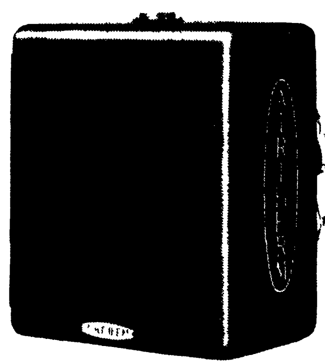
The Airtherm Mfg. Co. Engineering Department and District Representatives are at all times available for consultation. At your request we will place experienced engineering aid at your disposal. *Representatives in all principal cities.*



For Vertical Delivery

AIRTHERM
COPPER COIL, STEAM
UNIT HEATERS

Equipped with copper coils for longer life, durability, and resistance to corrosion. Individually controlled for economical operation. Deflectors are designed to give full efficiency from every ounce of steam expended. Write for bulletin showing specifications of complete line.



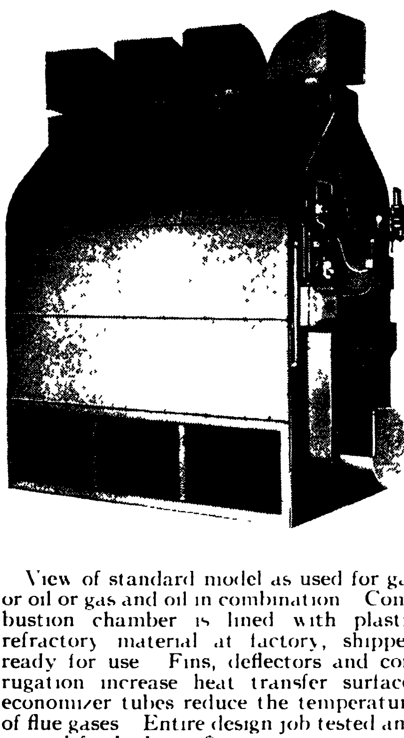
For Horizontal Delivery

DRAVO CORPORATION

HEATER DEPARTMENT

Dravo Bldg., 300 Penn Avenue, PITTSBURGH 22, PA.

Sales Offices in Principal Cities



View of standard model as used for gas or oil or gas and oil in combination Combustion chamber is lined with plastic refractory material at factory, shipped ready for use Fins, deflectors and corrugation increase heat transfer surface, economizer tubes reduce the temperature of flue gases Entire design job tested and proved for highest efficiency

Permanent savings in fuel consumption, cost of maintenance and cost of labor in operation are factors influencing favorable acceptance of Dravo Heaters

Conditions often demand speed of installation and curtailment in the use of critical materials Dravo Heaters are shipped with refractory and firewall in place, are installed by simply connecting to fuel and power supply and providing foundation breeching and vent stack, and their construction represents a saving in critical metals of 30 to 40 per cent over conventional steam plants with distributing systems

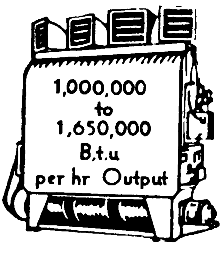
Each heater is complete and operated individually A heating system for any size structure may be formed with a combination of one or more heaters Dravo Direct-Fired Heating Systems save man-hours, money, transportation -all essential to the war effort.

Dravo Direct-Fired Heaters are used for permanent installation and also often used to supply temporary heat in new construction or plant expansion.

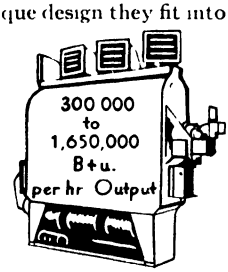
The exceptionally high heat transfer efficiency is the result of two fundamental design features first, accurately controlled combustion of fuel and air, and second, highly effective transfer of heat to air. Standard stock sizes are obtainable ranging from production of 300,000 to 4,000,000 Btu output per hour Dravo Bulletin No. 509-A with detailed description mailed on request

Specification Data Sheets for any type or capacity are furnished on request.

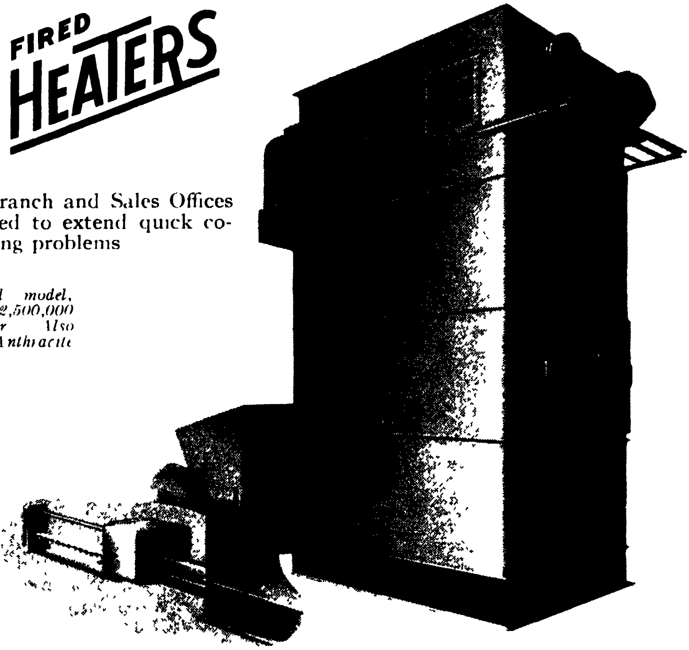
GAS-OIL COMBINATION . . . Floor Set . . . Top Discharge . . . Front Fired . . . Standard V-Belt Drive -This series of heaters is equipped to burn either Heavy or Light oil with alternate gas burner Equipment consists of a complete, separate burner for each fuel, each burner having a complete set of controls The controls and wiring are so arranged that either burner can be put into operation by simply throwing a switch, and valves



GAS . . . Floor Set . . . Top Discharge . . . Front Fired . . . Standard V-Belt Drive -This series as well as all others is designed to deliver the maximum output per square foot of floor space occupied Because of their unique design they fit into the heated space and are dependable and efficient. They readily lend themselves to a wide variety of applications, using either natural or manufactured gas.



DRAVO DIRECT FIRED HEATERS



There are 47 Branch and Sales Offices strategically located to extend quick co-operation on heating problems

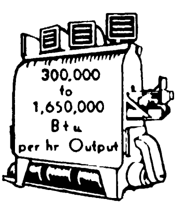
Right Hopper feed model, capacities 750,000 to 2,500,000 Btu output per hour Also available for bin-feed, Anthracite or Bituminous coal

COAL FIRED SERIES the Dravo coal-fired, self-contained Heater is particularly adaptable in areas where coal is the cheapest or easiest fuel to obtain Coal fired heaters may be converted to either gas or oil firing should conditions change The entire series of Dravo Direct-Fired Heaters has the utmost flexibility Enough applications, combinations, and adaptations are available to meet any set of reasonable conditions The same unique principle of welded combustion chamber is employed The entire series has the DRAVO non-clinkering air-cooled setting incorporated in the design Available for either bituminous, anthracite or lignite coal, equipped with either hopper feed or bin-feed stoker, installed under either end of the heater. Other series also available 300,000 through 650,000 and 2,750,000 through 10,000,000 Btu per hour output

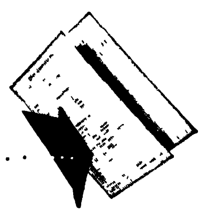
Overhead or underground duct systems may be used for air distribution

Specification Data Sheets for any type or capacity are furnished on request.

OIL, Floor Set, Top Discharge, Front Fired Standard V-Belt Drive This series obtainable in light oil or heavy oil fired models in all Dravo Direct-Fired Heaters Economy of fuel consumption is the result of careful over-all designing and constant improvement over many years In summer they may be used for air circulation.



heating analysis . . .



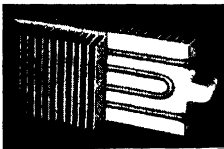
How much would an installation cost?

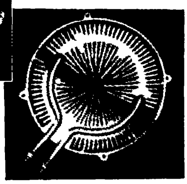
We have a form called "Dravo Building Survey" that we furnish free on request. You give us the vital statistics on this form and we'll do the rest. Write for your copy

Electric Air Heater Company

ELECTROMODE
Electric Unit Heaters

426 So. Byrkit Street, **Mishawaka, Indiana**

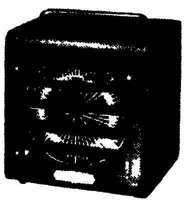


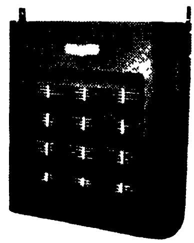


Cross-section views of Elements of Electromode Finned Heater—Circular and Rectangular Types

All-Electric Air Heaters

Cast aluminum grids in electric heaters are a radically different innovation from the popular conception of electric heating units. Aluminum, a metal of highest thermal conductivity, is cast on a tubular element. This process seals the heating element, preventing all oxidation and deterioration.





To the cleanliness, safety and convenience of Electromode Electric Unit Heaters, there are the further advantages of ease of installation, simplicity of control and economy.

The 1.5 to 7.5 KW sizes are for either portable use or suspension mounting, the larger sizes for suspension mounting only. By means of thermostat, heat can be automatically controlled. For complete data, see *Bulletin 44-U*.

1.5 KW to 7.5 KW Heaters

Model	KW	BTU	CFM Approx	Shipping Weight Approx	Price
[a]AA-15	1.5	5122	100	20 lb	$30.00
[b]AN-20	2	6830	200	40 lb	46.00
[b]AN-30	3	10245	200	40 lb	52.50
BN-40	4	13660	350	50 lb	59.00
[c]BN-50	5	17075	350	50 lb	64.00
[d]CN-7½	7.5	25613	550	80 lb	91.00

All heaters 1.5 KW to 4 KW, inclusive, are furnished with ON and OFF switch, 10 feet of heavy duty cord and plug.

[a]Model AA-15 (1.5 KW) may be plugged into usual branch lighting circuit,—made for use on 115 volt circuits only.

[b]AN-20 and AN-30 are for 115 V or 230 V,—specify which.

[c]BN-50 heater is furnished with switch but without cord or plug. 230 volts only.

[d]CN-7½ is furnished without switch, cord or plug. 230 volts only.

For Blast Coils and Air Ducts

Electromode Finned Heater Elements are made for a wide range of heating applications. Engineering help is at your disposal.

Home and Office Heaters

The production of Portable and Bilt-in Wall Heaters has been restricted during war, but will again be available, postwar.

10 KW to 60 KW Heaters

Model	KW	BTU	CFM Approx	Shipping Weight Approx	Price
[a]14-10	10	34150	800	100 lb	$110.00
[a]14-12	12	40980	800	100 lb	125.00
[a]18-15	15	51225	1500	180 lb	138.00
18-20	20	68300	1500	180 lb	190.00
20-25	25	85375	1800	200 lb	245.00
20-35	35	119525	1800	200 lb	255.00
27-45	45	153675	4000	360 lb	380.00
27-60	60	204900	4000	360 lb	390.00

For 230 or 460 volts specify which voltage when ordering.

[a]Models 14-10, 14-12, 18-15 are for single or 3 phase,—others only for 3 phase.

A contactor is necessary to provide operation of safety switch or thermostat.

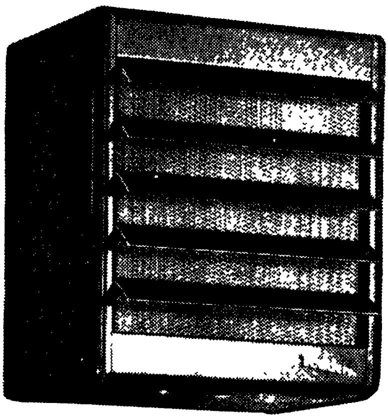
Heaters can be furnished with 460 single or 3-phase heater elements with 115 or 230 volt single phase motor at no increase in price. For 460 volt elements and motor, add 10% to price.

Contactor

A contactor is necessary on any unit above 5 KW capacity and on 3 phase service and direct current in order to provide operation of the safety switch and thermostat. Automatic thermostat and contactors are available for any size unit.

FEDDERS

MANUFACTURING COMPANY, INC.
85 TONAWANDA ST.
BUFFALO, NEW YORK



Fedders Series 15 Horizontal Unit Heaters with Streamline Copper Tubes and Fins

FEDDERS UNIT HEATERS
Horizontal Type

Fedders streamline tubes provide aerodynamic efficiency and the ample-area saddle of the fins gives generous bonded metal-to-metal contact between copper tubes and fins for high thermal efficiency Handsome, rugged cabinets complete relief of expansion stresses within the core as well as between core and cabinet latest type broad blade fans provide large air volume, quiet operation and maximum efficiency Large BTU capacity with low final temperatures assure ideal working conditions Write for Bulletin 15C-1

FEDDERS UNIT HEATERS
Vertical Type

Designed for high ceiling installations where supply and return piping will not interfere with overhead equipment such as craneways, shafting, tall machinery High velocity fan delivers heated air down to the working zone where draftless diffusion is accomplished by using suitable directional outlet to fit conditions

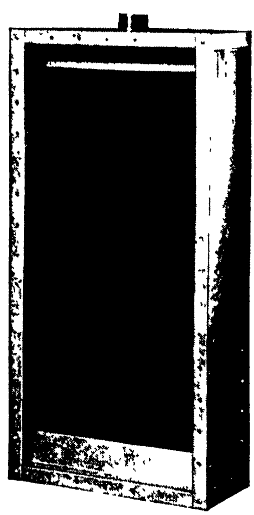
Fedders Series 10
Down Blow or Vertical
Unit Heaters.

Capacities
up to 2,500 EDR

Heat
TRANSFER
SPECIALISTS
SINCE 1896

Also Manufacturers of Blast Heating Coils,* Unit Coolers*, and Heat Exchange Equipment.

*Out of production for the duration watch for announcement

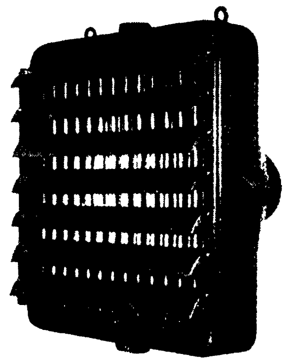


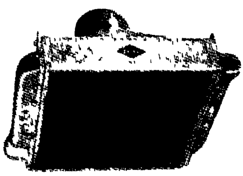
ILG Electric Ventilating Co.

2832 North Crawford Ave., **Chicago, Ill.**

Offices in 38 Principal Cities

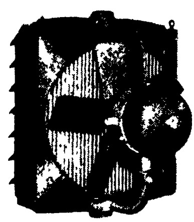
Horizontal Type Unit Heaters—have Self-Cooled Motor which counteracts coil heat—never "slow-roasts." Two piece header for "balanced" steam distribution—orifice bushings expand tubes uniformly in header plate. Fins are pressed into round tubes for permanent union—no brazing, soldering, or welding. Certified ratings "One-Name-Plate" Guarantee 40 capacities. Get Catalog No 341.



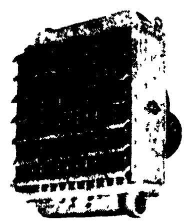


Vertical Type (left)—recommended for extremely high or extremely low ceilings.

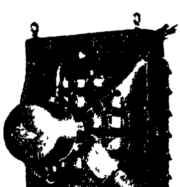
Textile Type (right)—for use where lint, etc., ordinarily adheres to fin surfaces and clogs up the coils. 10 capacities.



Low-Ceiling Type (right)—has side steam connections which make possible an extremely compact installation in locations where head-room is at a premium



ILG Electric Unit Heaters

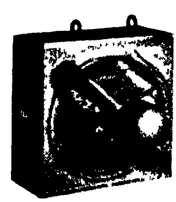


STANDARD TYPE

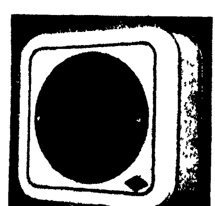
For instant, clean, safe, dependable heating. Coil is of black heat type which operates below 400 degrees Protected patented automatic thermal cut-out and magnetic starter Sizes, 5 KW to 48 KW. Get Bulletin No 802

TYPE "HT"

For installations requiring a small volume of heat Exceptionally efficient Suitable for constant duty Black heat type coil with individually interchangeable elements Automatic circuit breaker protects ILG Self-Cooled Motor from overload. 1½ to 4 KW



ILG Cooling and Air Conditioning Units



Ceiling type units for use singly or in multiple with remotely located refrigerating machine circulating direct expansion "Freon" or methyl chloride; or with cold water. For cooling, dehumidifying and recirculating. Also combination units for both cooling and heating. Write for Catalog.

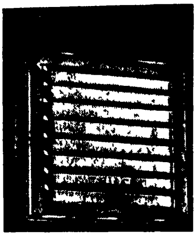
For ILG Fans and Blowers, see pages 950-951

Kramer Trenton Co.

Manufacturers of

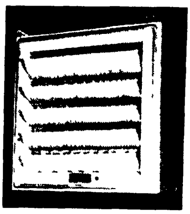
HEATING, COOLING AND REFRIGERATION DEVICES

Trenton, New Jersey



Kramer Unit Heaters

All-copper condensing coil with fused metallical fin-tube bond Positive compensation for thermal expansion High discharge air velocity Moderate final air temperature Silent in operation

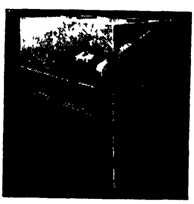


Kramer "Coolmaster" Product Coolers

An outstanding product cooler designed to give positive refrigerant distribution and maximum performance Built-in heat exchanger. Adjustable louvers

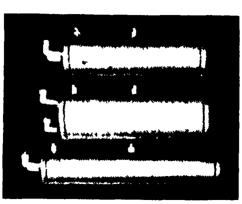
Kramer Panel Type Unit Coolers

Designed for application where space is at a premium Easily accessible for adjustments and service without use of tools All-copper evaporating coil Built-in heat exchanger



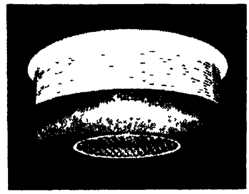
Kramer Floor Type Product Coolers

Designed for large cold storage installations Capacities range from 3 to 10 tons For all refrigerants Top or front discharge



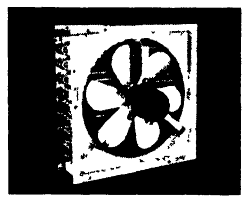
Kramer Fin-Tube Condensers

Combined water-cooled condenser and liquid receiver



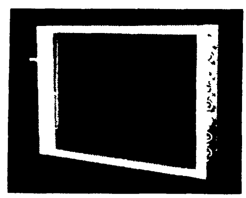
Kramer Radial Unit Cooler

Even air distribution in all directions Low discharge velocity High relative humidity Oilless ball bearing motors. Maximum space economy Pleasing appearance



Kramer Unicon

Self-contained, remote type air cooled condenser complete with fan May be used as booster in combination with air or water cooled systems



Kramer Turbo-Fin Surface

All-copper blast surface, for heating and cooling Used for steam, water or direct expansion refrigerants Air side and refrigerant side flow-disturbers for increased heat transfer Fin-tube ratio designed for maximum latent heat removal Coil construction designed to allow thermal expansion Electro-tin plate finish

Kramer Coolant Coolers

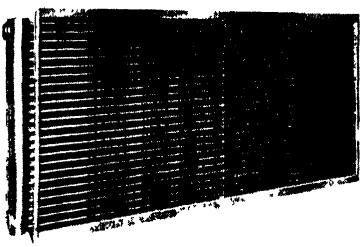
Shell and fin tube heat exchangers using water, brine or direct expansion refrigerant as cooling medium Wide variety of applications, including cooling of cutting fluids, quenching oils, hydraulic oils, plating solutions, and other chemicals Cooling coil easily removable for cleaning

KENNARD CORPORATION

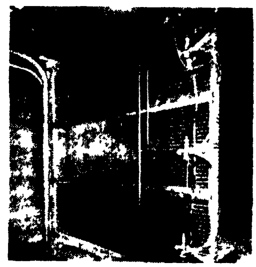
4821 Easton Ave., **St. Louis 13, Mo.**

Manufacturers

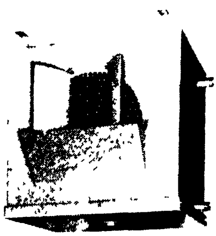
AIR CONDITIONING BLOWER UNITS—REFRIGERATION BLOWER UNITS
AIR CONDITIONING COILS - HEATING COILS



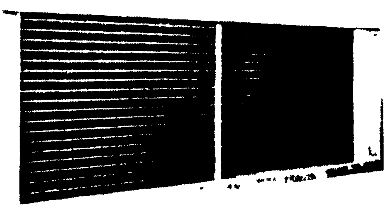
KENNARD Water Cooling and Heating Coils. Non-ferrous coils for cooling with brine or cold water and for heating with hot water



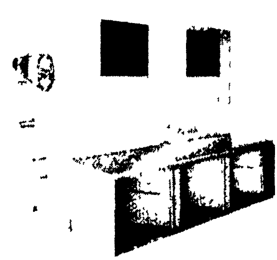
KENNARD Direct Expansion Coils. For all refrigerants A complete range of sizes to meet all conditions and capacities



KENNARD RS Unit. 100 per cent utilization of coil the Kennard Way Full coil surface is utilized when air is drawn through coil Units for brine, freon, ammonia and methyl-chloride. 9 sizes basic capacity ratings 410 Btu's to 1521 Btu's per degree temperature difference



KENNARD Steam Coil. A complete range of copper steam heating coils Can be furnished in non-freeze type with steam distributing tubes



KENNARD Floor and Ceiling Type Air Conditioning Units. Ten sizes Two to 35 tons cooling capacity range and 40,000 to 1,250,000 Btu's range in heating Various arrangements of discharge, filter box, motor drive

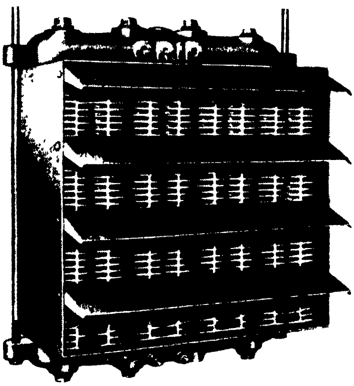
WRITE FOR COMPLETE INFORMATION

D. J. Murray Manufacturing Co.

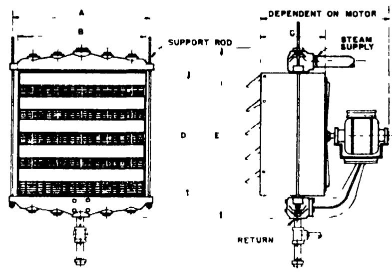
Wausau, Wisconsin

Offices in Principal Cities

MANUFACTURERS OF THE GRID UNIT



One piece construction "fin" heating sections of high test cast iron—no soldered, brazed, welded or expanded connections. Patented.



Overall dimensions for installation of Cast Iron GRID Unit Heater

Designed and tested to operate with steam or hot water systems—for steam pressures from 2 lbs to 250 lbs. Engineered along the same lines as the standard GRID Unit which had aluminum heating sections and has been on the market since 1929.

CI (CAST IRON) SERIES GRID UNIT HEATER DATA

Model No	Dimensions A	B	C	D	E	Motor Hp	Rpm	Vol Fan Cfm	Capacities 5 Psi Steam 60° F Btu/Hr	Final Air TMP°F	Approx Shipp. Wt. Lbs	Pipe Size Supply	Return	Support Rod Diam
CI-1000	11 3/4	10 1/2	12 1/4	10 7/16	16	1/20	1750	675	30010	101	160	1 1/4	1 1/4	1/2
CI-1200	14 1/8	13 1/8	12 1/4	13 5/16	19 1/8	1/8	1750	1142	54000	103	230	1 1/4	1 1/4	1/2
CI-1500	17 1/2	15 [illegible]	12	16	23 7/8	1/8	1750	1500	76500	107	300	1 1/2	1 1/4	1/2
CI-2000	22 1/8	20 9/16	12	21 1/16	28 [illegible]/8	1/6	1150	2600	143000	110	500	2	1 1/4	1/2
CI-2504	27 1/2	25 7/8	13	25 7/8	35 1/4	1/4	1150	3300	206000	117	675	2*	1 1/4	5/8
CI-2500	27 1/2	25 7/8	13	25 7/8	35 1/4	1/2	1150	4350	224000	107	700	2*	1 1/4	5/8
CI-3000	32 5/8	31	13	31	40 1/2	1/2	850	6300	332000	108	1025	2 1/2*	1 1/4	5/8
CI-3000	32 5/8	31	13	31	40 1/2	1/2	1150	8000	380000	103	1075	2 1/2*	1 1/4	5/8

*Furnished also with 2 in top supply connection inlet

NO ELECTROLYSIS TO CAUSE CORROSION

Low maintenance expense.
More air changes per hour.
Positive "directed" heat.
No leaks—no breakdowns
Lower outlet temperature.
Larger air volume.
No soldered, brazed or expanded joints.
Open design that keeps units clean.

Send for complete catalog information

Send for information on Blast coils and radiation.

McQuay, Inc.

1602 Broadway, N.E., **Minneapolis, Minn.**

MANUFACTURERS OF AIR CONDITIONING EQUIPMENT

Sales Offices in all Principal Cities

- **Air Conditioners**
- **Air Conditioning Coils**
- **Blast Heating Coils**
- **Refrigeration Coils**
- **Convection Radiation**
- **Unit Heaters**
- **Unit Coolers**

PROVEN McQuay PRODUCTS

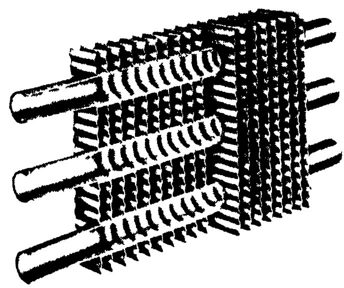
- **Comfort Coolers**
- **Blower Coolers (Suspended & Floor Type)**
- **Room Coolers (Cabinet Type)**
- **Ice Cube Makers**
- **Icy-Flo Accumulators**
- **Zeropak Low Temp. Units**

THE EXCLUSIVE McQUAY RIPPLE-FIN COIL ASSEMBLY

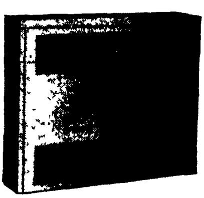
The McQuay Fin and Tube assembly in all McQuay coils and cores is one of the reasons McQuay products are considered "Tops in Over-All Efficiency" by many heating and refrigeration authorities.

Heat transfer efficiency primarily depends on three elements in coil construction First, "Area of Contact," Second, "Contact Pressure" and finally "Quality of Contact" between collar and tube.

In McQuay coils *all three* necessary elements are found developed to their highest degree. The famous McQuay "Wide Fin Collar" plus Exclusive Hydraulic Expansion together with the polished surface, secured by "spinning" the fin collar, truly provides the last word in Heat Transfer



RIPPLE-FIN COIL



STANDARD CONVECTOR

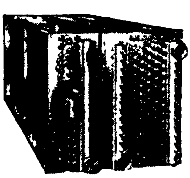
McQUAY STANDARD CONVECTORS

The Standard all purpose Convector has been designed to meet all heating requirements They are available for free standing, partially recessed, fully recessed and wall mounting applications

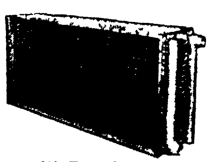
All enclosures are constructed from high grade steel, properly reinforced to make a sturdy cabinet

The heating element is constructed of a series of round tubes to which are attached die formed radiating fins, which are bonded to the tubes by an exclusive process

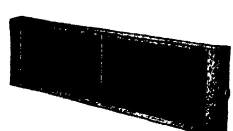
We offer the services of our Engineering and design department to help solve your heating problems



COMBINATION COOLING COIL



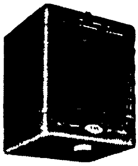
WATER COIL



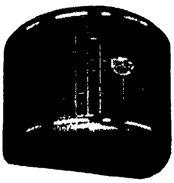
STEAM BLAST COIL

MORE THAN 1,000,000 STANDARD COIL TYPES AND SIZES

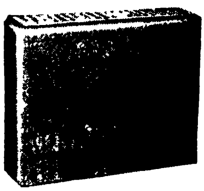
McQuay manufactures the most complete line of Standard Coils in the Industry.
Coils for Heating—1 to 10 rows deep using low or high pressure steam or hot water.
Non-Freeze—(steam distributing tube) type coils 1 and 2 rows deep
Removable Plug—(cleanable tube) type coils 1 to 12 rows deep
Water Coils for Cooling—1 to 12 rows deep.
Direct Expansion Coils—for cooling 1 to 10 rows deep.
Refrigeration Coils—all types and sizes.
Special Coils—of various materials furnished on order for special applications



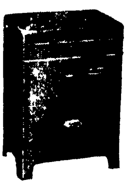
STANDARD UNIT HEATER



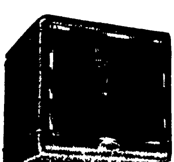
RADIAL UNIT HEATER



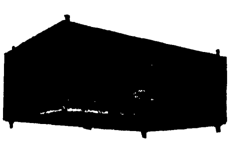
CABINET TYPE UNIT HEATER



COMFORT COOLER



COMFORT COOLER



AIR CONDITIONER (YEAR-ROUND)

UNIT HEATERS

Critical materials must be conserved. McQuay Unit Heaters have a steel core of highest heat transfer efficiency, and a sturdy steel cabinet. Truly it can be said that McQuay construction provides "greatest Btu per pound of metal used."

All McQuay Unit Heaters are furnished in sizes to meet with government regulations and with motors to meet all electrical current characteristics, making it convenient to select the proper size heater for every installation.

McQUAY COOLERS

For Industrial Application

Made in two types—one for use with water or brine, another for Freon or methyl chloride Eight sizes in each type—all with 4-speed motors

AIR CONDITIONERS—COLD WATER AND FREON TYPES

For Industrial Application

Choice of recirculation of indoor air, entire intake of outside air, or a combination of both Cold water or brine used in one type, Freon or methyl chloride in another Modern "sound isolated" construction assures quiet operation Capacities to 6 tons

CENTRAL SYSTEM AIR CONDITIONING UNITS

Suspended and floor types, cools, dehumidifies, filters, and circulates air in summer, heats, humidifies, filters and circulates air in winter Extreme flexibility and accessibility "built-in" Cooling capacities from 5 to 50 tons in both Suspended and Floor Type

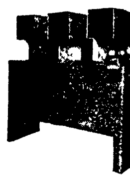
McQUAY ICY-FLO ACCUMULATORS

The new practical "Storage-Battery" for refrigeration effect is now available for handling heavy loads of short duration.

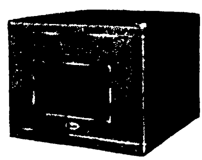
All McQuay Units are available in a large range of sizes, under applicable government regulations for Army, Navy, Maritime Commission, Coast Guard and essential industries. Write for Descriptive Bulletins McQuay, Inc, 1602 Broadway St N.E., Minneapolis, Minnesota.

DOWN FLOW UNIT HEATER

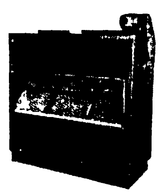
BLOWER TYPE UNIT HEATER



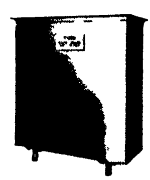
BLOWER TYPE UNIT HEATER



AIR CONDITIONER



AIR CONDITIONER (YEAR-ROUND)



ACCUMULATOR

Modine Manufacturing Company

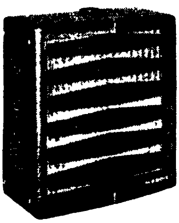
Heating and Air Conditioning Division

General Offices: 17th and Holburn Sts., **Racine, Wis.**

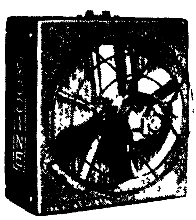
Factories at Racine, Wis. and La Porte, Ind.

Branches in all Principal Cities

MODINE UNIT HEATERS



Front View



Back View

HORIZONTAL DELIVERY MODELS

Condenser Copper and copper alloy, inlet to outlet, for maximum resistance to internal and external corrosion Pure copper fins are metallically bonded to round, seamless, heavy-gauge red brass tubes for permanent contact and to insure uninterrupted heat conduction from primary to secondary surface Modine-patented expansion bend permits tubes to expand or contract individually as temperature requires All steam and condensate carrying passages are brazed into an integral pressure-resisting unit

Direct - Pipe - Suspension — Modine patented feature permits suspending unit directly from supply line without additional time and labor wasting supports, also permits complete rotation of unit for redirection of air stream

Bonderized Casing—Casing protected from rust by Parker Bonderizing

Safety Fan Guard—Staunch, steel safeguard built into unit protects against danger of unshielded fan

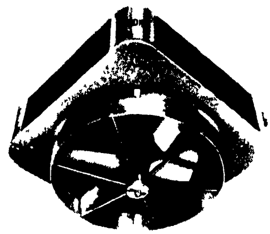
CAPACITIES AND DIMENSIONS

Model No	Btu Hr.*	Cfm	Rpm	Over-All Height	Width	Depth Less Motor
H-102	24,500	410	1580	14″	11⅛″	6″
H-144	34,600	585	1580	17¾″	14¼″	9″
H-184	44,200	685	1580	20⅛″	14¼″	9″
H-254	61,000	1020	1140	22½″	19″	11¼″
H-364	87,400	1500	1140	24″	19″	11¼″
H-474	113,800	1900	1140	27½″	23″	11½″
H-584	141,000	2400	1140	27½″	23″	11½″
H-704	169,000	2650	1140	29½″	24½″	11½″
H-894	214,500	3600	1140	33¾″	26½″	13¼″
H-1154	277,000	4300	1140	33¾″	26½″	13¼″

★

VERTICAL DELIVERY MODELS

★



Bottom View

Condenser All copper and copper alloy Tubes are seamless red brass Headers, inlet and outlet connections are heavy-walled copper pipes Fins are pure copper Brazing of all steam carrying passages and metallic bonding of fins to tubes insures rugged unit and continuously excellent heating Hollow-square condenser literally grows as it heats up, retreats to normal size as it cools with complete distribution of stresses and absence of strains where injury might result Parallel alignment of fins reduces possibility of dirt lodging between fins and clogging condenser

Cone-Jet Deflectors Verticals are regularly equipped with radial or spoke-like deflector assemblies illustrated above Individually adjustable deflector blades make possible delivery of high, narrow jet-like cone of heated air from high elevation . . or low, broad, softened-velocity cone from low elevation

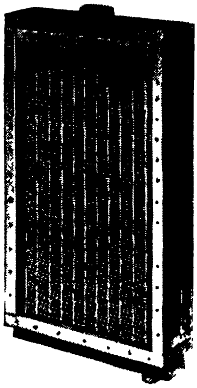
TrunCone Deflectors—Verticals can be furnished with special deflectors for use on units mounted at heights lower than recommended for Cone-Jet Deflectors

CAPACITIES AND DIMENSIONS

Model No	Btu/Hr*	Cfm	Rpm	Overall Dimension (Square)
V-604	145,000	2700	1140	33″
V-724	173,800	3400	1725	33″
V-794	190,200	3300	1140	33″
V-974	233,500	4300	1725	33″
V-1084	260,000	4550	1140	33″
V-1404	337,000	5600	1140	42¼″
V-2004	481,000	8000	1140	42¼″
V-2504	601,000	10,000	1140	42¼″

***Btu at 2 lb steam, 60 deg Ent Air**

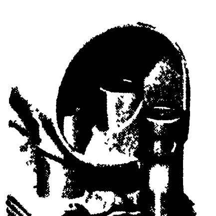
BLAST HEATERS



For modern air conditioning, heating, ventilating, drying and processing systems Condenser is pure copper and copper alloy, inlet to outlet, for resistance to corrosion, durability, light weight and high heat transfer "Floating" condenser construction provides safe expansion and contraction Fins are metallically bonded to tubes to prevent corrosion from reducing heat transfer capacity Die formed fins promote greater heat transfer, permit use of smaller, lighter coils Modine design permits use of ducts having cross-sectional areas no greater than those of condenser face Wide range of capacities and sizes available in Standard and Booster Unit types

STEAM DISTRIBUTION TYPE BLAST HEATERS

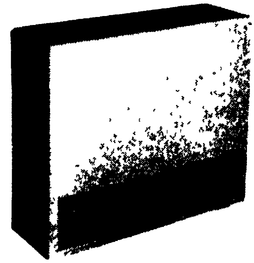
Cross-section supply end showing double tube construction



These coils are characterized by the uniform distribution of steam throughout entire heating surface even when steam is partially throttled to meet system temperature demands Inserted in each condenser tube is a steam-distributing tube with small, accurately-sized and spaced orifices along its entire length Steam entering distributing tube is uniformly rationed through orifices into external or condenser tube then flows as condensate into return header Uniform steam distribution minimizes tendency of condensate freezing and tube damage; eliminates need for preheaters or tempering coils, thus simplifying system design and control, eliminates air stream stratification

CABINET UNIT HEATERS

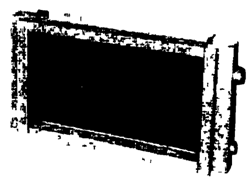
Designed for heating offices, lobbies, corridors, etc . . . wherever quick, positive distribution of heat combined with quiet operation and gentle air movement are desirable. Has considerably greater heat output than convector of equivalent size. Employs copper heating coil for use on steam or hot water system Mounted on wall or ceiling Capacities 105 Edr and 310 and 450 Edr



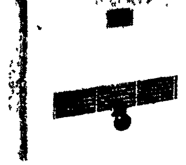
MARINE HEATING PRODUCTS

Unit Heaters Horizontal and Vertical Delivery types with all-copper and copper-alloy condensers, as described on preceding page

Ventilation Heaters -- Standardized Navy and Commercial Marine types in preheater and reheater design Extensively used on U S Navy and Maritime Commission ships. All copper condensers. Shock-proof construction. Protected against salt air corrosion. Variety of sizes and fin spacings available Navy type in accordance with latest Bureau of Ships General Type Plan



Convectors—Standardized Navy and Commercial Marine types. In wide use aboard Navy and Maritime Commission ships. All copper heating unit. Enclosures are light weight, but structurally strong and built for rugged seagoing service Navy type (illustrated) in accordance with latest Bureau of Ships General Type Plan and available for immediate delivery



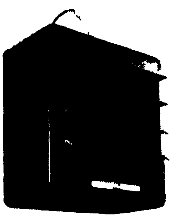
Technical Bulletins on complete line of Marine heating equipment supplied on request.

THE HERMAN NELSON CORPORATION

General Offices at Moline, Illinois
Factories at Moline and Chicago, Illinois

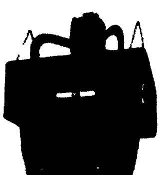
Herman Nelson Branch or Product-Application Engineer Offices in the Following Cities:

Cape Elizabeth, Me	Cleveland, Ohio	Memphis, Tenn	Minneapolis, Minn
Boston, Mass	Cincinnati, Ohio	New Orleans, La	Dallas, Tex
Baltimore, Md	Columbus, Ohio	Indianapolis, Ind	El Paso, Tex
New York City, N Y	Washington, D C	Louisville, Ky	Houston, Tex
Syracuse, N Y	Richmond, Va	Chicago, Ill	San Antonio, Tex
Buffalo, N Y	Charlotte, N C	Milwaukee, Wis	Albuquerque, N M
Philadelphia Pa	Birmingham Ala	Des Moines, Ia	Denver Colo
Pittsburgh, Pa	Miami, Fla	St Louis, Mo	Salt Lake City, Utah
Detroit, Mich	Jackson, Miss	Kansas City, Mo	Spokane, Wash
Saginaw, Mich	Atlanta, Ga	Omaha, Nebr	Los Angeles, Calif
Grand Rapids, Mich	Nashville Tenn	Duluth, Minn	San Francisco Calif



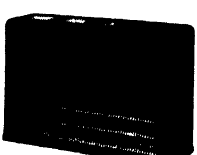
HERMAN NELSON HORIZONTAL SHAFT PROPELLER-FAN TYPE UNIT HEATER

Designed for ceiling suspension, this Unit Heater projects warm air downward in an angular direction Copper heating element for use with steam or hot water, incorporates patented stay tube which maintains proper relationship between headers without increasing strain on loops thus prolonging life of unit Forty-eight models, sizes and arrangements



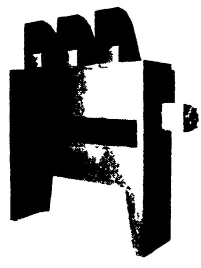
HERMAN NELSON VERTICAL SHAFT PROPELLER-FAN TYPE UNIT HEATER

Discharges air vertically downward, or at an angle to vertical in various directions Long life copper heating element for use with steam or hot water incorporates patented stay tube Unit available with either high or low velocity discharge, each with a wide range of capacities



HERMAN NELSON DE LUXE UNIT HEATER

Efficient, economical, compact, quiet and attractive, this Unit Heater provides the ideal method for heating offices, showrooms, corridors, and stores, etc Copper heating element incorporates patented stay tube Unit may be placed on floor, wall or suspended from ceiling Eighteen models, sizes and arrangements



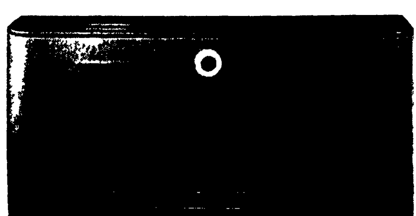
HERMAN NELSON BLOWER-FAN TYPE UNIT HEATER

For efficient heating of large areas Can be supplied with by-pass damper for introduction of indoor and outdoor air Design of copper heating element assures durability and contributes to high velocity discharge For floor, wall, ceiling or inverted wall mounting One hundred and fifty models, sizes and arrangements

HERMAN NELSON UNIT VENTILATOR

Maintains desired air conditions for room or auditorium areas where large groups of people gather Classroom type unit has damper control Auditorium type unit has radiator control Copper heating element for use with steam or hot water is sturdily constructed for long life and designed for greatest heating efficiency

Exclusive "draw-through" design prevents unhealthful drafts and eliminates unnecessary overheating Locating motor in end compartment provides additional space for fan assembly and use of larger fans running at slower tip speeds



Herman Nelson Unit Heaters and Unit Ventilators are tested and rated in accordance with the Standard Test Code adopted jointly by the *Industrial Unit Heater Association* and the American Society of Heating and Ventilating Engineers.

THE HERMAN NELSON CORPORATION

General Offices at Moline, Illinois
Factories at Moline and Chicago, Illinois

HERMAN NELSON DIRECT DRIVE PROPELLER FANS

Tried and proven on installations everywhere and found to be the most economical form of quality ventilation available. There are eighteen standard sizes available with wheel diameters from 10 to 48 in and capacities from 670 to 15,650 cfm For operation against resistance pressures there are sixteen high powered models with wheel diameters from 12 to 48 in and capacities from 1260 to 22,690 cfm An acid resisting model can be obtained where desired When explosive gases or fumes are being handled a Herman Nelson Explosion Resisting Fan should be used

Standard

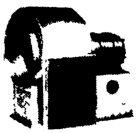
High Powered

HERMAN NELSON BELT DRIVE PROPELLER FANS

Most economical and efficient for public and commercial building installations where slow speed, quiet operation is required There are six sizes of the standard Herman Nelson Belt Drive Fan with wheel diameters from 24 in to 54 in There are also six sizes of the High Powered Belt Drive Fan with the same wheel diameters for operation against slight resistance pressures Capacities 5000 to 23,000 cfm

HERMAN NELSON BELT DRIVE UNIT BLOWERS

Fully self-contained unit including motor, drives and housing, forwardly curved blade wheel, adjustable motor pedestal with vibration dampers, universal discharge; nine sizes with 52 drive combinations. These blowers are available with any rotation and discharge Wheel diameters from 11 in to 30 in and capacities from 1035 to 16,940 cfm



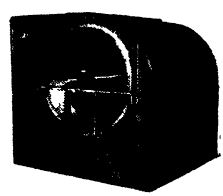
HERMAN NELSON DIRECT DRIVE UNIT BLOWERS

These Herman Nelson Blowers are designed for many applications, such as fume hoods, toilet ventilation, chemical laboratories, industrial processing and drying problems These compact, direct connected, motor driven units have universal discharge and mount on floor, wall or ceiling Available in five sizes with 14 speed combinations Wheel diameters from $6\frac{1}{2}$ in to 15 in and capacities from 360 to 4320 cfm

HERMAN NELSON TYPE "H" AND TYPE "HB" CENTRIFUGAL FANS

For heavy duty ventilating and air conditioning installations Type "H"—forwardly curved blade wheels, Type "HB"- backwardly curved blade wheels incorporating non-overloading power characteristic

Type HB



Type H

The Complete Line of Herman Nelson Propeller Fans and Blowers is tested and rated in accordance with the Standard Test Code adopted jointly by the *National Association of Fan Manufacturers* and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

John J. Nesbitt, Inc.

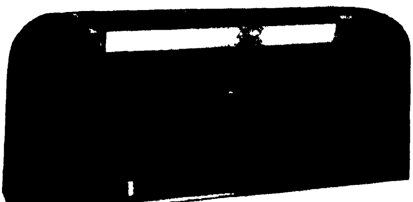
Holmesburg, Philadelphia 36, Pa.

11 Park Place, **New York 7, N. Y.**

Manufacturers of

THE NESBITT SYNCRETIZER Heating and Ventilating Unit,
sold by John J. Nesbitt, Inc., and American Blower Corporation;
NESBITT HEATING SURFACE with Dual Steam-distributing Tubes,
NESBITT SERIES H HEATING SURFACE, and
NESBITT SERIES W COOLING SURFACE,
sold by leading manufacturers of fan-system apparatus;
WEBSTER-NESBITT UNIT HEATERS (See page 1099),
distributed in U. S. A. by Warren Webster & Company.

Consult Manufacturer for changes due to war time restrictions



Interior View

NESBITT SYNCRETIZER—Series 400

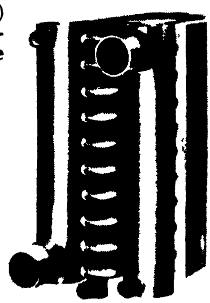
The last word in heating and ventilating units for schoolrooms, offices, etc., where the continuous introduction of outdoor air is desired. For engineering data, get Publication No. 225-1, for "The Story of Syncretized Air," Publication No 231-1

Nesbitt Series B Thermovent

For heating and ventilating auditoriums, gymnasiums, assembly halls, and similar gathering places. Publication No. 227-1

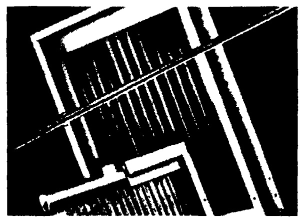
NESBITT COOLING SURFACE

Series W (Water) Surface with exclusive drain feature



Uncased Surface Showing Drain Header

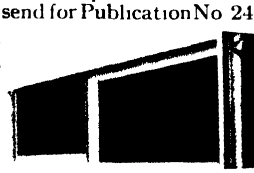
For air cooling and cooling and dehumidifying (with cold water) or air heating (with hot water). Constructed of copper tubes and plate-type aluminum fins. Available in either continuous or cleanable tube type, in single sections having one to eight rows of tubes deep, in three fin spacings, in eleven fin widths, and up to sixteen finned tube lengths. Sturdy galvanized casings. For particulars and engineering data send for Publications No. 233.



NESBITT HEATING SURFACES
With Dual Steam-distributing Tubes

Copper tube-and-fin surface for low-pressure applications Perfectly adapted to close, continuous automatic control with modulating steam valves. Steam-distributing tubes within the condensing tubes carry the steam equally to the full section assuring UNIFORM discharge temperatures even under a throttled steam supply; eliminating temperature stratification; preventing tube freezing without preheaters; giving ideal system results

Cased or uncased units of many sizes and capacities For full particulars and engineering data, send for Publication No 242.



For above advantages plus uniform distribution in extended fin lengths from 86 to 122 ins., specify *Nesbitt Duplex Heating Surface* with Dual Steam-distributing Tubes Publication No 242.

Nesbitt Series H Heating Surface

A lightweight, enduring, highly efficient blast-coil heating surface designed for use with steam pressures up to 200 lb gauge. Well suited to high-pressure as well as low-pressure applications. Four types, each in five fin widths and up to seventeen finned lengths—a total of 288 sizes from which to select. Send for Publication No. 243 for complete engineering data.

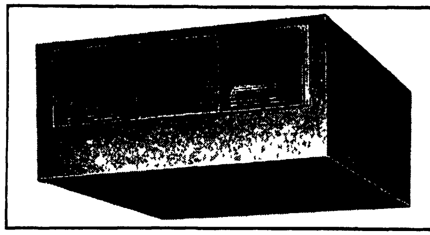
Refrigeration Economics Co., Inc.

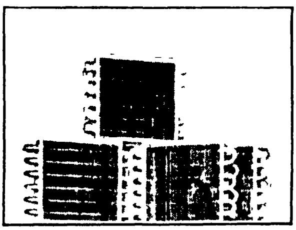
Canton, Ohio

RECOY PRODUCTS

RECOY AIR CONDITIONING UNITS of the suspended type as shown, or vertical floor type, are made for all season purposes, also for summer cooling or winter heating and humidifying

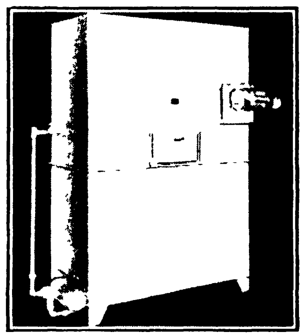
Capacities range from one ton up to any size required. Cooling and heating surface, and filter area are liberally proportioned and blowers are of moderate speed, all to insure the highest efficiency and quiet, satisfactory performance. Bulletin "E"





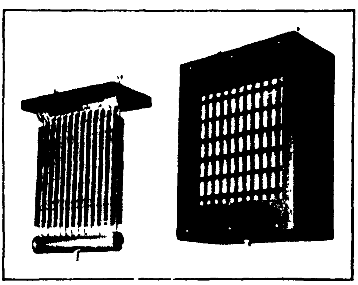
RECOY CONTINUOUS FIN BLAST COILS

for cooling or heating are constructed of copper tubing with aluminum fins, or all steel hot dip galvanized after fabrication and are suitable for use with any cooling or heating medium Bulletin "F"



RECOY EVAPORATIVE CONDENSERS

are cooling towers and condensers combined into one efficient unit for use indoors or out They reduce the water consumption 95 per cent and are used with no water at all in cold weather Made in sizes from one to one hundred tons Ceiling Type 2 to 12½ tons Bulletin "G"



RECOY BLAST HEATERS have all welded coils and headers so will stand any steam pressure and remain tight for years. Coils are copper tubing with aluminum fins or all steel hot dipped galvanized after fabrication The entire unit is suspended on the top header and coil is free to expand.

Fan motors are oversize to insure continuous service and fans are guarded.

The casings have liberally rounded corners and are beautifully finished in baked crinkle enamel.

Quotations and data on request.

RECOY CEILING DIFFUSERS as illustrated are ideal for air conditioning or refrigeration in that the air is distributed at the ceiling and causes no drafts on occupants.

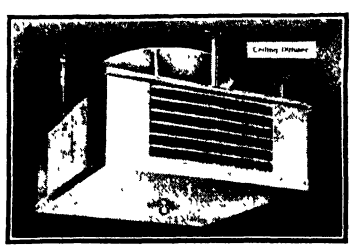
For heating the construction is reversed and air is diffused radially at bottom

Fan motors are oversize and ball bearing for continuous, satisfactory service

Coils are three rows deep, welded and made of copper tubing with aluminum fins or all steel hot dipped galvanized after fabrication, and will stand any pressure and remain tight for years.

Casings are heavy steel attractively finished with baked on crinkle enamel

Quotations and data on request.



The Trane Company

2021 Cameron Avenue, La Crosse, Wisconsin

THE WORLD'S MOST COMPLETE LINE OF Heating and Air Conditioning EQUIPMENT

Over 80 U. S. Branch Offices

Albany, N Y ; Allentown, Pa , Amarillo, Texas, Appleton, Wis , Atlanta, Ga , Baltimore, Md , Birmingham, Ala , Boston, Mass ; Buffalo, N Y , Canton, Ohio; Chattanooga, Tenn , Chicago, Ill , Cincinnati, Ohio, Clarksburg, W Va , Cleveland, Ohio, Dallas, Texas, Davenport, Iowa, Dayton, Ohio, Denver, Colo , Des Moines, Iowa, Detroit, Mich , Dodge City, Kan , Flint, Mich , Fort Wayne, Ind , Gainesville, Fla , Grand Rapids, Mich , Greenville, S C , Harrisburg, Pa , Hartford, Conn , Houston, Texas, Indianapolis, Ind , Kalamazoo, Mich , Kansas City, Mo ; La Crosse, Wis ; Lake Charles, La , Little Rock, Ark ; Los Angeles, Calif , Louisville, Ky , Memphis, Tenn , Mexico, D F , Milwaukee, Wis , Missoula, Mont , New Orleans, La , Newark, N J , New York, N Y , Oklahoma City, Okla , Omaha, Neb , Philadelphia, Pa , Phoenix, Ariz , Pittsburgh, Pa , Portland, Ore , Richmond, Va , Rochester, N Y , Salt Lake City, Utah, San Francisco, Calif , Seattle, Wash , Sioux City, Iowa, South Bend, Ind , Spokane, Wash , St Louis, Mo , St Paul, Minn , Syracuse, N Y , Trumbull, Conn , Washington, D C , White Plains, N Y , Wilkes-Barre, Pa , Wilmington, Del

Sales Connections All Over The World

Export Dept. La Crosse, Wisconsin

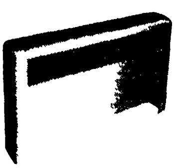
In Canada TRANE COMPANY OF CANADA, LTD , Mowat and King Sts , W , Toronto, Ont (12 Branches)

A COMPLETE LINE

The Trane Company fabricates the most complete line of heating, cooling, and air handling equipment available from one manufacturer. Long years of experience with practical knowledge gained from close field contact, have developed products for every heating and air conditioning need.

Application data on Trane Equipment is available to qualified persons

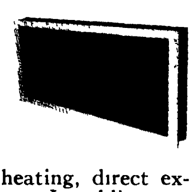
TRANE CONVECTORS



The modern successor to the old fashioned radiator, the Trane Convector is a compact, lightweight, easy-to-install unit Available for either steam or hot water heating system. It is equally suited to war plants, army camps, offices, or the finest homes, combining attractive appearance with long life and economical service.

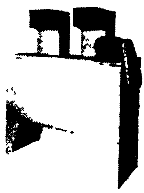
Currently available in steel or nonferrous enclosures Obtainable in four cabinet models for floor or wall installation, with or without sloping tops

TRANE COILS

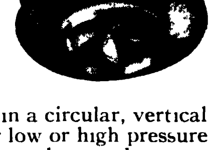


There are Trane Extended Surface Coils for every heating or cooling, comfort or process application, in all types and sizes Types include coils for steam, hot water or booster heating, direct expansion or water cooling. In addition, a wide variety of coils is available for specialized or process work.

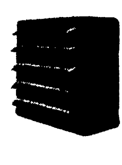
TRANE UNIT HEATERS



Blower Types—Better known as Torridors, Trane Blower Type Unit Heaters are available for large space or duct work applications. Manufactured in direct or belt driven types and floor, ceiling, or wall models, the Torridor may be provided with thermadjust or face and by-pass dampers and a variety of discharge devices for greater flexibility. Ideal for heating large spaces, exposed areas requiring a blanket of heat, and for process applications.



Projection Type—A Trane development, the Projection Heater taps the usually wasted heat reservoir at the ceiling bringing it down where it is needed. Installed at 8 to 50 ft heights, it projects warm air to the floor in a circular, vertical air stream Ideal for low or high pressure systems in factories, warehouses, hangars, etc.



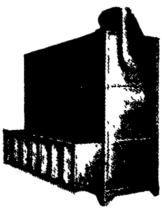
Propeller Type—Featuring a quiet operating, wide-bladed fan that pushes rather than bats the heated air through the coil, the Trane Propeller Unit Heater incorporates many unusual features, including adjustable louvers or grilled outlets directing heated air to the floor line, rugged motor supports, attractive appearance.

TRANE AIR CONDITIONING MANUAL

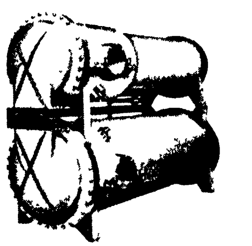
(New 1945 Edition)

Trane offers the engineering profession a straight-forward and unbiased textbook covering the fundamentals of air conditioning, in a new enlarged 1945 edition. The Manual not only shows how to design every type of air conditioning system, but also clarifies underlying principles, enabling both student and engineer to reason out their own problems Price $5 00

TRANE COOLING EQUIPMENT



Climate Changer—Trane Climate Changer, a unit type air conditioner, is designed for summer, winter, or year 'round air conditioning, commercial and industrial application, comfort or process installations These units incorporate efficient Trane Coils, quiet blower fans in sturdily constructed casings in horizontal or vertical models Available in various coil combinations with or without humidification equipment.



Refrigeration Equipment—Outstanding in the refrigeration field is the Trane Turbo-Vacuum Compressor, a completely self-contained hermetically sealed centrifugal type water chiller, available in 50, 70, 100, and 200 ton sizes Year 'round efficiency in constant operation with a minimum of maintenance is assured by the scientific simplicity of this machine

Trane also furnishes a complete line of Reciprocating Compressors, 3 to 50 tons

TRANE VENTILATORS

Heating and Ventilating Units—The Trane Heating and Ventilating Unit of the Climate Changer type is manufactured in six standard sizes, all available in horizontal, and four of which can be obtained in vertical, models Furnished with heating coils and recirculating and outside air dampers, these units are adaptable for ventilating and heating large areas

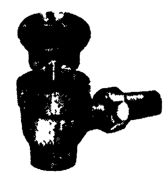
Roof Ventilators—The Trane Roof Ventilator can be used wherever an effective roof ventilator is required. This unit is available for either exhaust or supply purposes It is not only light, but weather and bird proof as well. Hoods of the units are hinged for easy maintenance.

TRANE FANS

Blower Type—Recommended for all types of heating, cooling, and air handling applications. In direct or belt-driven units, single or double widths, and all standard discharges in both backward and forward curved blade construction Capacities 200 to 300,000 cfm.

Propeller Type—The exclusive broad-bladed propeller fan is available in 2 or 4 blade construction Can be used for vertical or horizontal installations for exhaust or supply applications. Direct or belt driven Capacities, 500 to 100,000 cfm.

TRANE HEATING SPECIALTIES



Steam There are over fifty valves, traps, vents, strainers, all allied specialties in the Trane Line Trane Steam Specialties meet the rigid specifications of the U S. Navy Most are available with brass or cast iron bodies depending on final use

Hot Water Included among Trane Hot Water Heating Specialties are the Trane Circulator, Flo Valves, and Fittings. They combine with Trane Convectors or Unit Heaters to provide an ideal Warm Water Heating System for a great variety of applications

OTHER TRANE EQUIPMENT

Trane's complete line also includes Evaporative Condensers to condense refrigerants in the air conditioning system with a minimum use of water; Shell and Tube Heat Exchangers for cooling and heating vapors or liquids in process applications; Evaporative Coolers for cooling fluids in a closed system; Transformer Oil Coolers, Oil Heat Exchangers, Air Washers, Spray Nozzles, Condensation Pumps, etc.

Factories:
NEWARK, N. J.

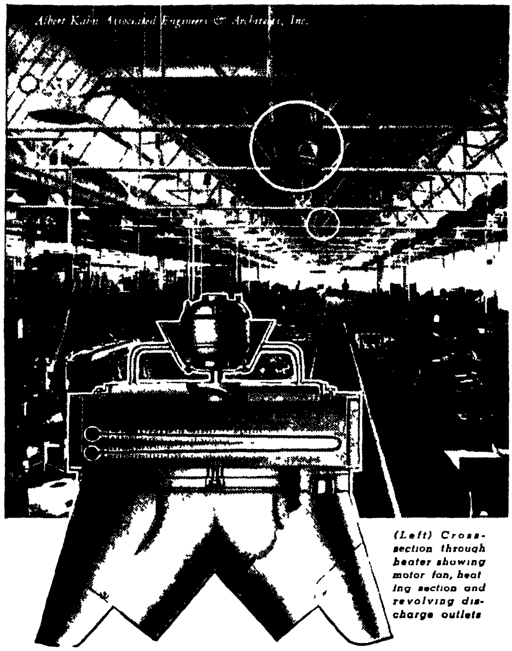
L. J. Wing Mfg. Co.

59 Seventh Avenue, **New York 11, N. Y.**

Canadian Factory:
MONTREAL

Branch Offices in Principal Cities

WING REVOLVING UNIT HEATERS



(Left) Cross-section through heater showing motor fan, heating section and revolving discharge outlets

This innovation in the method of distributing heat produces a sensation in heating comfort never before attained—a sensation of fresh, live, invigorating air

The fact that the outlets revolve assures uniform and thorough distribution of comfortably warmed air throughout the entire working area, without drafts, hot spots or cold spots.

Such an unprecedented high efficiency in distributing heat is the result of nearly 20 years of constant study by Wing engineers to improve on the Floodlight System of heating originated by WING in 1920. This method projects the heated air vertically downward by means of light-weight, ceiling-suspended unit heaters.

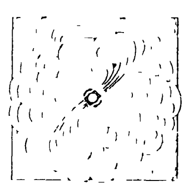
It has needed only this latest refinement of slowly revolving discharge outlets to bring that method to perfection.

The WING Revolving Discharge type supplements the WING line of standard fixed discharge outlets, illustrated and described on the following page.

Bulletin HR-4.

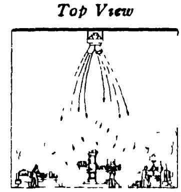
The latest type of WING Unit Heater—with Revolving Discharge Outlets—is just as great a contribution to the art of industrial heating as was the Ceiling-Suspended Unit Heater, originated by WING in 1921.

The area covered by a WING Revolving Unit Heater is slowly swept by the heated air discharged by the outlets which move through an arc of 360 deg. covering every direction of the compass successively.



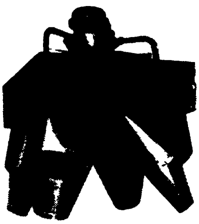
Top View

By maintaining an active, constant circulation of air throughout an industrial plant at all times, a new sensation of refreshing, invigorating comfort to workers is produced.



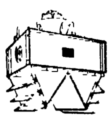
Elevation

WING FEATHERWEIGHT UNIT HEATERS

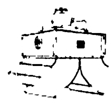


Type "HC" Fixed Discharge

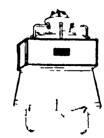
The first light-weight, ceiling-suspended, unit heater. Eight different designs of outlets meet the requirements of every type, size and height of building or occupancy. Located near ceiling or roof, the accumulation of hot air in the upper spaces, with the accompanying costly waste of heat, is prevented They project the air, comfortably warmed, downward to the working area. *Bulletin HR-3.*



Design No 3

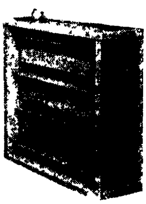


Design No 4



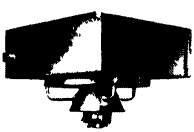
Design No 8

VARIABLE TEMPERATURE SECTIONS



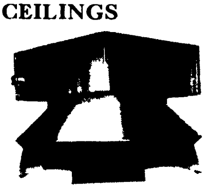
Invaluable in supplying fresh air for space heating or process work Close control of the delivered air temperature is obtained without danger of freezing Manual or automatic control *Bulletin HS-2.*

DOOR HEATERS GARAGE HEATERS



WING originated the vertical cone-discharge heater in 1921 and today it is still applicable for heating the inrush of cold air at large doorways and for garage heating Often cuts heating costs in half *Bulletin HR-4.*

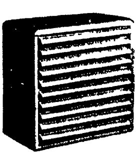
FOR LOW CEILINGS



Type "LC"

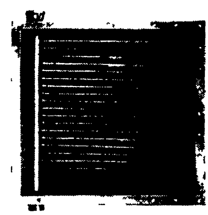
In this type of WING Unit Heater the position of fan and motor are reversed to meet conditions of ceiling or roof height, form and shape of building, coverage, etc *Bulletin HR-4.*

WING UTILITY UNIT HEATERS



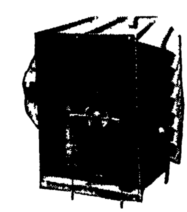
A lightweight suspended unit heater for delivering heated air in one general direction. Has the same powerful fan and rugged heating element as WING Featherweight Unit Heaters. This is the latest refinement of the original horizontal lightweight heater which was developed by WING. *Bulletin HR-4.*

FEATHERFIN HEATER SECTIONS



For heating or cooling air for any purpose by steam, hot or cold water or refrigerant. The heating element is extremely light and, for equal heat transfer, offers little resistance to air flow. Available for any desired final air temperature. *Bulletin HS-2*

WING INDUSTRIAL FOG ELIMINATORS

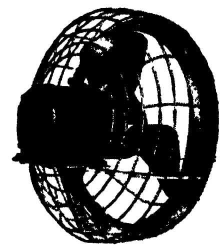


Eliminate fog, odor and fumes in dyeing, bleaching and finishing plants, creameries, pasteurizing, bottling, canning and packing plants, chemical works, paper mills, steel pickling plants, etc. No ducts are required *Bulletin FE-12.*

TURBINE-DRIVEN HEATERS

Any WING Unit Heater can be furnished with steam turbine-driven fan for locations where high-pressure steam is available. Photo shows turbine-driven revolving unit heater. Can also be supplied for fixed discharge or utility type heater. *Bulletin HR-4.*

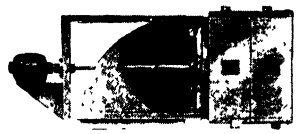
WINGFOIL SAFETY VENTILATING FANS



An axial flow fan that will deliver air against static pressure, quietly and efficiently

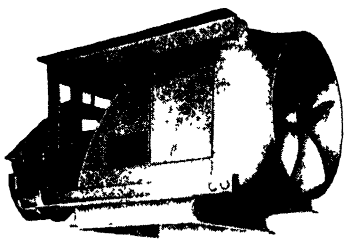
Moves the air forward in straight lines with minimum eddy. Capacities to 100,000 cfm. *Bulletin F-9*

WING FEATHERFIN PROCESS HEATING UNITS



For manufacturing processes such as drying, aging, etc., requiring the recirculation of the heated air Motor or turbine located outside air current. *Bulletin P-2.*

WINGFOIL DUCT FANS



For economically moving air wherever ducts are used It combines the efficient WINGFOIL AXIAL FLOW Fan with a housing which places the motor entirely outside the air duct Motor and drive remain cool and clean and are easily accessible

The powerful WINGFOIL Fan delivers high air volume with low power consumption against any pressures for which duct systems should be designed V-belt or direct drive

Light, compact and easy to install *Bulletin F-9*

WING SYSTEM OF CONTROLLED COMBUSTION

For low pressure heating boilers and small power boilers Increases capacity and permits use of lowest cost fuel. Includes Type EM Blower equipped with fully enclosed dustproof motor with speed regulating rheostat and automatic control Eliminates necessity of frequent firing, allowing intervals as great as 24 hours even in zero weather *Bulletin M-96.*

Installation of Wing System of Controlled Combustion in a large school

WING TURBINE-DRIVEN BLOWERS

Applied to hand, stoker, oil or pulverized fuel fired boilers, increase boiler capacity, maintain constant steam pressure and permit complete combustion of low-cost fuels The exhaust steam, free from oil, can be used for heating or processes. *Bulletin T-98.*

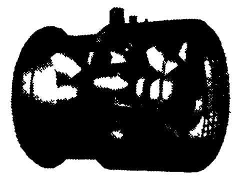
WING DRAFT INDUCERS

Installed in breeching or flue, or on chimney top; provide positive, exact draft regardless of weather conditions or inadequate chimney or breeching construction. Suitable for coal, oil, or gas-fired boilers; industrial furnaces and kilns. *Bulletin I-10.*

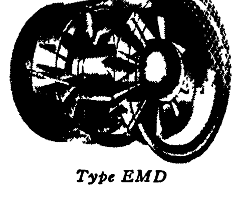
Chimney-Top Installation

WING MOTOR-DRIVEN BLOWERS

Type COM for static pressures over 5 in. and volumes up to 35,000 cfm. Type EMD for moderate static pressures up to 5 in Both blowers have fully-enclosed dustproof constant speed motor and built-in adjustable control vanes. Type COM has double-staged axial flow fan; Type EMD, single stage fan. Extremely compact; discharge can be vertical, horizontal or inclined. *Bulletin CO-5.*



Type COM



Type EMD

Young Radiator Co.

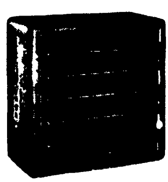
Dept. 175, **Racine, Wis.**

Sales and Engineering Offices in Principal Cities

HEAT TRANSFER ENGINEERS

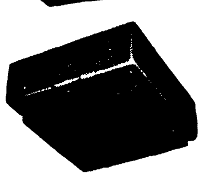
"*STREAMAIRE*" *UNITS*

"Streamaire" units are a development of Young's quarter century of experience in building heat transfer products. Personalized, "on-the-job" engineering service by field men– backed by modern research and manufacturing facilities - assures the practical, economical installation as required in your plans



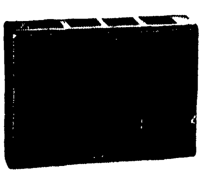
Type "SH" unit heaters for horizontal air discharge

Available with capacities from 19,000 to 325,000 Btu per hour *Catalog 2738*



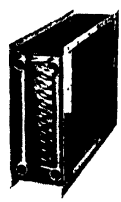
Type "V" or Vertiflow unit heaters for vertical air discharge

Capacities from 30,000 to 480,000 Btu per hour *Catalog 2541*



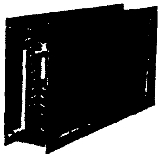
Type "FC" units have the appearance of convectors plus the capacity and circulating features of unit heaters.

Four models *Catalog 6539*



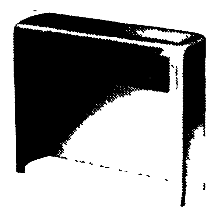
Type "W" water coils for cooling or heating with central plant systems

Five widths, 11 to 35 in., 2 to 8 rows of tubes, many lengths Write for details



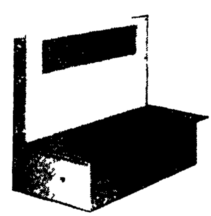
Type "E" evaporator coils for direct expansion cooling systems using Freon or Methyl chloride

Four widths—two to six rows of tubes, many lengths *Catalog 5039*



Several types of convector units—circulate rather than radiate heat Used with steam or hot water systems

Styles to blend with architecture and room furnishings



Type "DAC" units provide individual room, winter air conditioning—filter, humidify, heat, circulate *Catalog 7041*

"YAC" units for year-round air conditioning Also for winter or summer conditioning only

Horizontal and vertical models Capacities from 400 to 16,625 cfm *Catalog 7541*

Types "B" and "A" blast coils for central plant heating and air conditioning systems

Steam distributing tube type available *Catalogs 4640 and 4542*

Type "C" commercial heat transfer coils for use in factory built air conditioning units.

Steam distributing tube type available One, two and three rows of tubes *Catalogs 4640 and 4542*

The Air-Maze Corporation

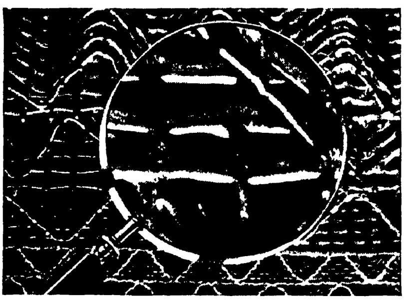
5202 Harvard Avenue, Cleveland, Ohio

ENGINEERS AND MANUFACTURERS OF AIR FILTERS EXCLUSIVELY

DIRECT FACTORY REPRESENTATIVES IN ALL INDUSTRIAL AREAS.

DISTRIBUTORS IN PRINCIPAL CITIES AND TOWNS THROUGHOUT THE UNITED STATES

Viscous Impingement Type Air Filter—Air-Maze viscous impingement cleanable type air filters are the result of nearly two decades of designing and building air filters exclusively Air filter media consists of alternate layers of crimped galvanized screen cloth of various mesh, arranged to break up the air stream into minute, swirling currents Dirt is impinged on the wire baffles, previously coated with an adhesive



Progressive Density Provides Increased Dirt-Holding Capacity—An important feature of Air-Maze construction is the progressive density of the filter media, permitting lint and large particles to be collected on the upstream section of filter with little impedance to air flow. As shown in photos above, large quantities of dirt can be collected before filter requires cleaning This design also lends itself to quick, thorough cleaning and recharging.

Dirt Arrestance—Efficiency of dirt arrestance varies, depending on filter design, type of test dust, dust feed, etc. Specific information on any type filter gladly furnished on request

Resistances—Filters available with initial resistances as low as 0 045 in water at 300 fpm (Graphs showing resistance at various velocities are available for each type of filter)

Face Velocities Air-Maze offers air filters designed to operate efficiently at velocities from 106 fpm to 2000 fpm, depending upon application, pressure drop available, etc Our representative will gladly furnish full information.

Sizes Air filter panels made in rectangular shapes of any reasonable size and thickness 1 in , 2 in. and 4 in thicknesses are standard. Specially shaped filters also built to special order

Holding Frames—Holding frames, complete with felt seal and choice of locking devices, are supplied when requested For filter banks, Air-Maze furnishes either assembled frames or will drill frames for assembly on the job

Cleaning Data—Filters easily cleaned with either hot water, steam or any commercial solvent Charged by immersing in recommended adhesive or *S A.E.* 30-50 oil Complete instructions available upon request

Write for Details—Write for name of nearest representative or information on any type of filter. Representatives in most principal cities. See classified section of your telephone directory.

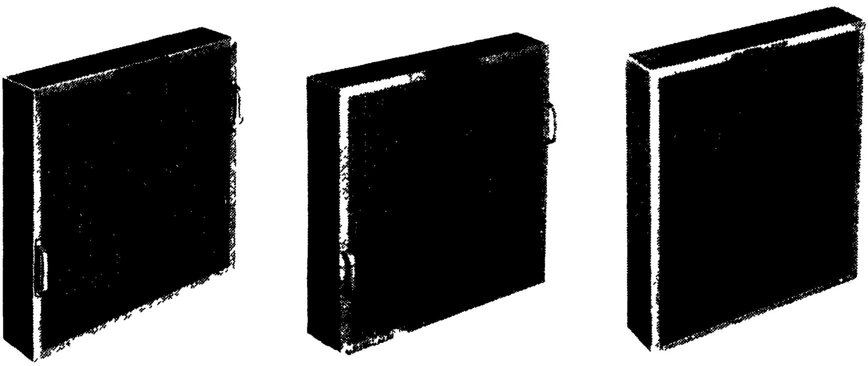
The Air-Maze Corporation

5202 Harvard Avenue, **Cleveland, Ohio**

ENGINEERS AND MANUFACTURERS OF AIR FILTERS EXCLUSIVELY

DIRECT FACTORY REPRESENTATIVES IN ALL INDUSTRIAL AREAS.

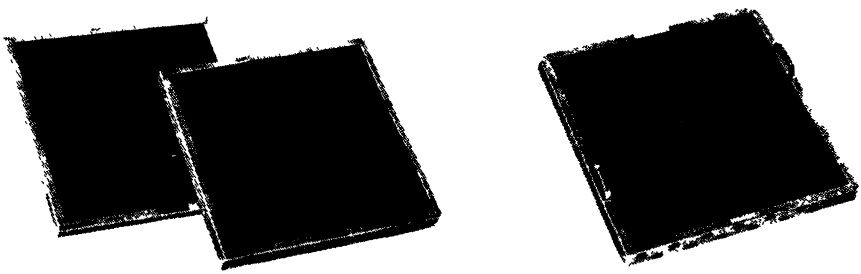
DISTRIBUTORS IN PRINCIPAL CITIES AND TOWNS THROUGHOUT THE UNITED STATES



AIR-MAZE CORPORATION

Air-Maze Corporation manufactures a complete line of viscous impingement type ventilating air filters to meet your requirements within a wide range of limits of pressure drop, velocity, dust-arrestance, weight, dirt-holding capacity and price

These filters are all-metal washable fire retardant.



AMERICAN AIR FILTER COMPANY INC.

673 Central Avenue, **Louisville, Ky.**

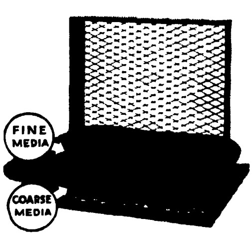
Representatives in Principal Cities

Dust Engineering—Dust Engineering is that branch of applied science which deals with the origin, nature and characteristics of the small solid air-borne particles called "dust," and the development of methods, processes and apparatus for its control or elimination.

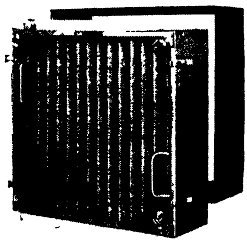
The American Air Filter Company, Inc., has had an important part in advancing the science of Dust Engineering. The efforts of its Research and Engineering Staff for the past twelve years have been devoted exclusively to the study of dust problems and the development of a complete line of air cleaning equipment for modern air conditioning, building ventilation and the control of process dust in industry.

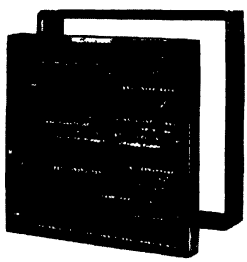
American Air Filter products, therefore, not only embody the knowledge accumulated from years of constant research and the experience gained from designing, building and applying thousands of air filters, but are backed by ample technical and financial resources to insure their outstanding position in the Dust Engineering field.

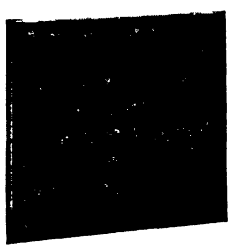
Products—American Air Filters are available for every condition, with operating characteristics and efficiencies to suit specific problems. In general, there are two distinct types based upon the "viscous film" and "dry mat" principles. Each type is made in several styles which differ in method of operation, servicing, space required and initial cost to meet the various conditions encountered in air cleaning problems. A discussion of various filter types will be found in the Technical Data Section under "Air Cleaners."



Renu-Vent Filter


Airmat Type PL-24 Filter


M/W 2 Filter


Throway Air Filter

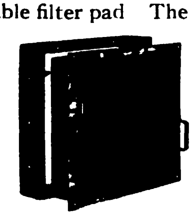
Air filters are generally used for the removal of dust, dirt, bacteria and other foreign matter from the air and are applied to general ventilation, modern air conditioning, process dust control; for air compressors and Diesel Engines; mill motors, turbo-generators and other electrical applications; and for air or gas under pressure to remove entrained oil, moisture and dirt.

Air Filters In Air Conditioning—Filtered air is today recognized as essential in modern air conditioning There are other important factors which contribute to our comfort such as temperature, air movement and humidity, but science today emphasizes the prime necessity of pure air for health and efficiency

Air cleaners have, of course, always been considered an integral part of large central systems These are usually of the fully automatic type such as the American Automatic Self-cleaning filter or the self-cleaning Electro-Matic.

There are now available to manufacturers of unit air conditioners moderate priced unit filters such as the Renu filter, the Throway filter, and other types of filters illustrated on this page.

The Renu filter is an entirely new departure in air filter construction. It consists of a permanent metal frame provided with a removable cover and renewable filter pad The cover


Standard Viscous Unit Filter

is easily removed without the use of tools, and filter pad can be lifted out and replaced with a new one at very small expense.

The Throway filter, as the name implies, is designed to be discarded after it has served its maximum period of usefulness and replaced with a new filter unit. The Filter pad is enclosed in a perforated cardboard container which makes it possible to readily dispose of the dirty filter by burning it.

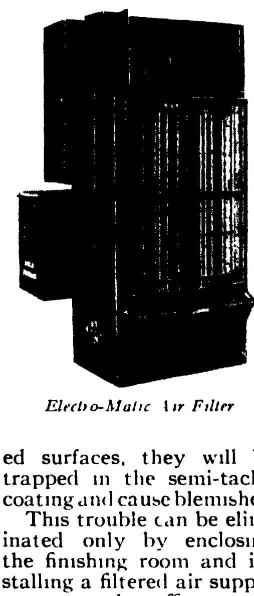
There is probably no single item which costs as little and may mean as much in the design of an air conditioner as air filtration These units are furnished in any dimensions or shapes desired—usually in units handling 400 cfm and from 2 in to 4 in thick. They are usually made in the following sizes—20 x 20 in., 16 x 25 in and 16 x 20 in High cleaning efficiencies can be secured, with a resistance to air flow ranging from 1/16 in to 3/8 in water gauge

The American Multi-Duty Self-Cleaning Air Filter—Is a development of AAF Engineers in which the best features of the former Multi-Panel and Double-Duty filters have been incorporated in a single unit to obtain the combined advantages of both types of construction

The use of thermoplastic finishes for refrigerators, stoves, automobiles, and other metal products has created the need for clean air in finishing rooms This type of finish is hardened by baking, so the product on which it is used must be protected from dust and dirt from the time it is sprayed until it leaves the oven.

Spray booths exhaust large quantities of air, and if this air is drawn from other parts of the plant, it will contain considerable dust and dirt. If dirt and dust particles are permitted to settle upon freshly sprayed surfaces, they will be trapped in the semi-tacky coating and cause blemishes

American Automatic Self-Cleaning Filter



Electro-Matic Air Filter

This trouble can be eliminated only by enclosing the finishing room and installing a filtered air supply system with sufficient capacity to provide a constant supply of clean air in excess of the volume exhausted by the spray booths

High efficiency air filters are needed for this service to minimize rejects and do-overs. The automatic self-cleaning filter has proved the most practical type because of its ability to maintain a constant, uniform air volume with the minimum of attention.

Three Types of Panels—MV, MS and DD—Are available with the Multi-Duty filter. Type MV is the designation for Multi-Panel armored panels with "V" or low resistance spacing, Type MS signifies Multi-Panel armored panels with "S" or high efficiency spacing, and Type DD indicates Double-Duty die-stamped louvre panels.

Electro-Matic Air Filter—Incorporates electrical precipitation as an integral function of an automatic self-cleaning viscous filter to obtain a higher over-all efficiency in the collection of the finer dust particles and smoke. In combination, these two methods of cleaning air not only give the highest efficiency in dust removal but offer operating advantages found only in the automatic self-cleaning filter.

Standard Viscous Unit—The American Unit Air Filter incorporates the time tested unit principle of construction. Each unit consists of a standard steel frame and interchangeable cell equipped with automatic latches to facilitate removal for cleaning and recharging.

Airmat Filter Dry Type—The filtering media in this type is the Airmat sheet, a dry filter mat composed of thin sheets of gauzy, cellulose tissue. The Airmat sheets are supported in screen pockets mounted in a unit frame of box-like construction. These unit frames can be set up to meet any capacity requirement or space condition. Airmat sheets are renewable—their life depending on dust conditions and hours of service.

Airmat filters are used both for comfort and industrial air conditioning. In the latter field they are particularly well adapted for the recovery of valuable dusts and for abating the dust nuisance prevalent in so many industrial plants.

W. B. CONNOR ENGINEERING CORP.

114 East 32nd Street
New York 16, N. Y.

Representatives
in All Principal Cities

Canadian Representative: Arthur S. Leitch Co., Ltd., Toronto, Ont.

Air Recovery DOREX **Equipment**

DOREX Activated Carbon Adsorbers recover the freshness of vitiated air by extracting air-borne odorous, gaseous and vaporous impurities When applied to recirculated air in air conditioning, DOREX Adsorbers convert used, stale, *conditioned* air for ventilation thus reducing to a minimum the volume of *unconditioned* outdoor air make-up otherwise necessary

With DOREX Air Recovery, the cooling and heating capacity of a proposed air conditioning system is decreased by the saving in outdoor air load

With DOREX Air Recovery, an existing air conditioning system will serve a larger space or satisfy a greater conditioning load without increase in cooling or heating equipment or the consumption of more fuel or energy

The highly active, specially processed and impregnated carbon employed will remove from the air passed through it and retain 95 per cent of all entrained gaseous impurities and maintain approximately this efficiency for from 6 months to 2 years depending upon the air contamination Upon exhaustion the carbon may be reactivated for re-use

DOREX Air Recovery equipment is available in several types to suit individual requirements

TYPE H FOR COMPLETE DECONTAMINATION OF ALL AIR PASSED THROUGH IT

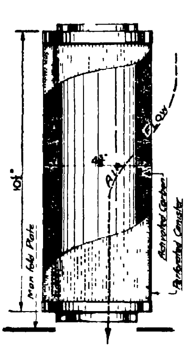


Fig 1

Fig 2

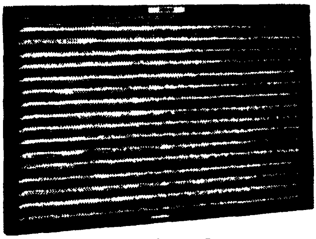
Fig 3

DOREX Type H Equipment consists of light-weight, removable, perforated, activated carbon filled, adsorption canisters. These are mounted in multiple on one or more supporting manifold plates in such manner that all air to be treated will pass uniformly through the granular carbon media The assembly arrangement is flexible to suit the space limitations. The resistance to air flow averages only 0.15 in wg.

Fig. 1 shows a typical canister It is closed at the top and the inner cylinder is open at the bottom, which opening registers with a corresponding hole in the supporting manifold plate Fig 2 is a photograph of a typical arrangement of canisters as installed. In this instance, three manifold plates each support 98 canisters arranged in 7 rows of 14 canisters each. Fig 3 is a sectional view of this arrangement with the side removed and indicating direction of air flow. Each canister decontaminates from 25 to 35 cfm of air.

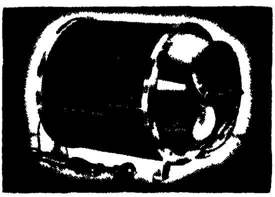
TYPE G FOR CONTINUOUS DECONTAMINATION OF RECIRCULATED AIR

The DOREX Type G Adsorber and Air Recovery Unit is designed to remove accumulated odors and gaseous impurities from air recirculated by air conditioning units and systems Its extremely compact design and the ease with which it can be installed makes it equally adaptable both to existing ventilating systems without necessitating expensive alterations, and to newly designed conditioning systems, particularly where space is at a premium It is also adapted to incorporation in *package* conditioners, unit heaters and cold diffusers



Dorex Type G

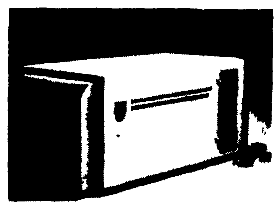
The Type G Adsorber consists of sturdy metal frames forming panels, each housing a battery of exposed perforated metal tubes which contain the granular carbon filter media These panel units are available in stock sizes, suitable for arrangement in air ducts, or for attachment to standard dust filters, outlet or inlet grilles They are designed to expose a maximum of carbon adsorption surface to the air stream with a minimum of resistance to air flow Standard units are of one, two or three tube rows in depth, designated as G-1, G-2 and G-3, respectively



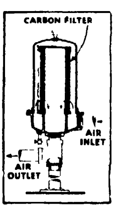
Dorex Type SQ

The DOREX Type SQ Odor Adsorber is an entirely self contained unit suitable for either portable or stationary installation It consists of a cage of closely spaced, perforated, granular activated carbon-filled tubes housing a quiet operating, circulating fan and motor It is furnished assembled, equipped with starting switch and complete with extension cord and plug ready to connect to electric socket **The DOREX Type SQ Unit** keeps the room air in continuous motion while, at the same time, completely decontaminating one-third of the total air circulated It, therefore, provides both air circulation and ventilation without the use of outdoor air

At the right is an illustration of the **Type A-100-B,** smallest of the DOREX self-contained *package* recirculating Odor Adsorbers In its attractive enameled wood cabinet are contained a dust filter, four carbon gas adsorbing canisters, circulating fan and motor It has a host of practical uses in homes, offices, doctor's rooms, walk-in refrigerators, etc



Dorex Type A



Dorex Type PL

The Type PL DOREX Vapor and Gas Adsorber is designed especially to extract all vapors, fermentation odors and other gaseous impurities from compressed air It is the only device which effectively removes air-entrained gaseous odors and impurities not eliminated by commercial filters, separators, after-coolers or receivers

"AIR CONSERVATION ENGINEERING"

A complete and authoritative Text Book on the function, engineering and application of Air Recovery in air conditioning Contains valuable tables and charts Price $2.00. Complimentary copies available to recognized engineers upon request.

AMONG THOUSANDS OF DOREX USERS

Armour & Co	General Electric Co	Remington Rand, Inc.
Baltimore & Ohio RR	Harvard University	Sperry Gyroscope Co.
Bell Telephone Co	Lockheed Aircraft Corp	E. R Squibb & Sons
E. I. DuPont de Nemours & Co.	Pratt & Whitney	Union Carbide Co
Ford Motor Co		F. W. Woolworth Co.

Detroit Lubricator Company

Division of American Radiator & Standard Sanitary Corporation

Detroit 8, Michigan, U. S. A.

Chicago 5, Ill., 816 S. Michigan Avenue — New York 18, N Y., 40 West 40th Street — Los Angeles 13, Calif , 320 Crocker Street

Canadian Representative:

Railway and Engineering Specialties Limited, Montreal, Toronto, Winnipeg

"DETROIT" AIR FILTERS

Detroit Air Filters are of the replacement type. Highly efficient and very low initial resistance. The entire depth of filter is used in cleaning the air—large dust carrying capacity.

The pollen removal characteristics of Detroit Air Filters make them extremely effective in treatment of hay fever Tests indicate efficiency of 99 per cent when used in tandem; over 95 per cent for single filters

OPERATION

Dust laden air entering the filter changes direction 45 deg causing a scrubbing action along the sides of the first cellular passages; then changes direction 90 deg again scrubbing along the sides of the smaller cellular passages.

As dust impinges on the sides of the passages, oil is absorbed by capillary attraction into each dust particle, and that dust particle then becomes the medium for catching other dust particles which, in turn, absorb oil and become the new dust catching medium, and so on, until the filter becomes completely dust-laden

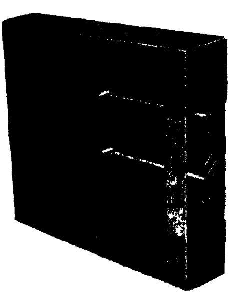
CONSTRUCTION

The Detroit Air Filter is constructed of fiber board and is completely reinforced to make a rigid, non-sagging unit.

In the No 1 Filter the cellular passages formed by the fiber board are the same size on both the inlet and outlet sides of the filter. In No 2 and No. 25 Filters passages on the inlet side are larger than the passages on the outlet side These passages are so arranged that they make a 90 deg angle

COATING

The absorbent coating or dust catching medium is a special odorless compound that remains tacky at minus 10 F and will not drip at temperatures as high as 180 F. It does not contain any substance that will become rancid. This special compound is sprayed on the fiber board before assembling, at a temperature that insures thorough penetration Because of the porous nature of the fiber board the Detroit Air Filter holds many more times the weight of adhesive substance than filters made of non-porous materials



TYPES

No 1. Inlet 3/16 in corrugations Outlet 3/16 in corrugations
For high efficiency applications and pollen removal

No 2 Inlet 5/16 in. corrugations Outlet 3/16 in corrugations.
For ventilating systems or for furnace and air conditioning installations where a portion of air handled is outside air Usually installed in tandem in ventilating systems

No 25 Inlet 1/2 in corrugations Outlet 5/16 in corrugations.
For installations where larger part of air handled is recirculated air When ordering specify type number and size

Type No. 2 Filter 2 in thick furnished on all orders unless otherwise specified

STANDARD SIZES

Type No	Nominal Sizes			Available with 2-pass 3/16 in., 1-pass 3/16 in or 5/16 in Corrugations — Nominal Sizes					
	Wide	High	Thick	Wide	High	Thick	Wide	High	Thick
Nos 1, 2 and 25	16″	20″	2″	16″	20″	1″	20″	20″	1″
	16″	25″	2″	15″	20″	1″	20″	25″	1″
	20″	20″	2″	16″	25″	1″	10″	20″	1″
	20″	25″	2″	16″	23″	1″	10″	10″	1″
	15″	20″	2″						
	10″	20″	2″						
	10″	10″	2″						

The above nominal sizes allow sufficient clearance for easy replacement. Detroit Air Filters can be made in any size and any thickness to meet requirements. Prices supplied upon application.

Research Products Corporation

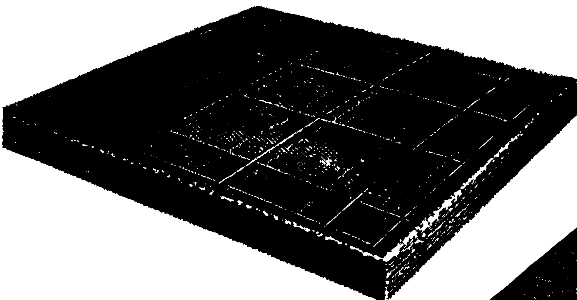
Madison, Wisconsin

RESEARCH AIR FILTERS FOR HEATING AND AIR CONDITIONING

SILICA GEL MOISTURE ABSORBING MATERIAL

U. S. Patent 2070073 — U. S. Patent 2294478

RESEARCH Air Filters With RiP-CLEAN Features



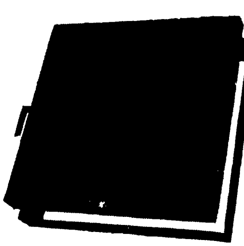
No "200" Series

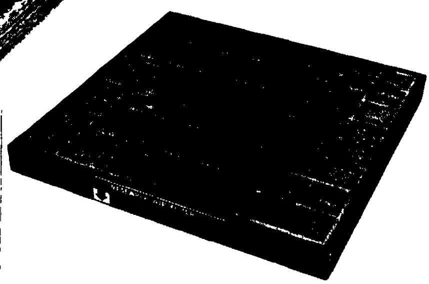
No. "200" Series—When this Re-Fil-Able filter becomes dirt filled, the wire grids are unhooked, the filter pad replaced Pads fit snugly, sealing against air leaks. (Can be Rip-Cleaned)

Exclusive Rip-Clean Feature—When the filter becomes clogged you need only rip off two top layers of filter pad to remove the surface dirt and lint This may be repeated 5 times—extending the useful life of the filter at high efficiency.

Research Filter Banks—Research Air Filters are used extensively in filter banks—both flat and V designs Write for technical data sheets to determine type and size of bank for any installation

Research Washable Filter—This filter is used where a washable filter is specified and also above kitchen ranges to trap grease particles Wash with hot water or steam.





No "100" Series

No. "100" Series—A fiber framed filter is designed so the entire unit is easily replaced by a new one Offers high dust removing efficiency throughout its long life—adaptable to heating, ventilating and air conditioning systems. (Can be Rip-Cleaned)

A Research Air Filter 20 x 20 x 2 in, when tested according to the test code of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has an efficiency of 91 per cent, tested with 80-20 dust. The dust holding capacity with standard code dust is 150 grams per sq ft of filter area, the restriction at this dust load being .2 in of water

Research Silica-Gel—For maintaining dry conditions in storage rooms; ideal for drying refrigerants.

Dollinger Corporation

(Formerly Staynew Filter Corp)

**Air Filters for Building Ventilation,
Air Conditioning, Engine Protection, Etc.**

6 Centre Park Rochester 4, N. Y.

STAYNEW FILTERS

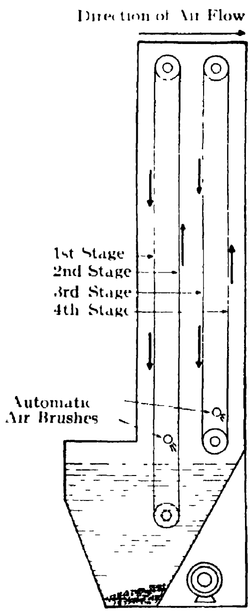
STAYNEW AUTOMATIC FILTER

An improved, fully automatic air filter that maintains a low, constant resistance

Four-Stage air filtration is provided in the form of two endless curtains (see line drawing) arranged to travel in such a direction (counter-clockwise) that they will be thoroughly cleaned after exposure in primary stages and before functioning in secondary stages. After thirty minutes of filtering service in a stationary position, both endless curtains are set in motion simultaneously for a few seconds The time and amount of travel vary according to the height of the unit

In the first stage, the curtain descends into oil bath which releases tension of oil film allowing dirt to settle to bottom of reservoir In subsequent movements, curtain is well drained, pneumatically brushed free from oil globules and moved upward to provide the second stage The third stage serves to collect any residue of oil that might possibly entrain in air stream while passing through first and second stages and also serves as an additional semi-dry air filtering stage In this stage, the curtain proceeds downward into reservoir but does not pass through oil bath, cleaning being accomplished by a second air brush similar to the first After passing over the air brush, the curtain becomes a semi-dry fourth stage which acts as a further safeguard against oil entrainment and as the final stage in the removal of the finest dust particles.

Automatic Filter



Drawing showing features

Specifications:

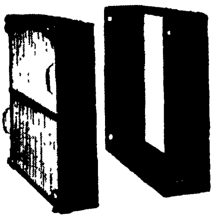
Two standard widths are available,—4 feet, 3 inches and 2 feet, 9 inches in forty-one heights from 4 feet to 14 feet Capacities from 2,000 cfm upward Combinations of standard sizes will fit any required capacity or installation space. Special sizes built to order

A single 1/6 hp motor operates three full-width automatic sections This is made possible by use of ball bearing rollers and a near perfect balance of the curtains, plus a 400 to 1 reduction gear and a further 3 to 1 reduction through silent chain drive and sprockets.

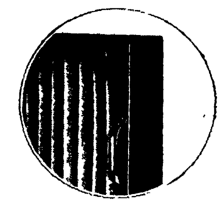
Flow of compressed air to air brushes occurs simultaneously with curtain movement and is controlled by time clock and solenoid valve. A pressure reducing orifice assures delivery of air to the air brushes at the required pressure of approximately 1 pound.

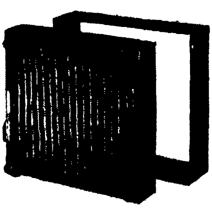
Complete descriptive catalog supplied on request.

STAYNEW PANEL-TYPE FILTERS


Wire-Klad Panel and Frame

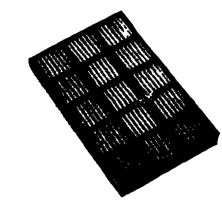
Wire-Klad Model: Flame resistant, highly efficient; cleanable by compressed air, vacuum, or washing. Fins are reinforced on both sides with rust-resisting screen cloth, producing a rigid, long-wearing unit. The Wire-Klad feature is not found in any other filter. When supplied with bonded glass medium, it carries the Bureau of Fire Underwriters Class I rating. Redesigned box-type supporting frame and new style latches and handles (see close-up view) simplify removal for inspection or cleaning. Replacement cost is low, since finned element only need be replaced, and is easily removed from its retaining case. A wide selection of filtering media available to meet varying conditions. Made in standard sizes, 2 in. or 4 in. deep.

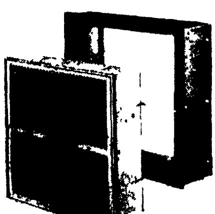

Close-up view of Handles and Latches


Viscous Panel and Frame

Viscous Panel (Model DPV): Medium is viscous-coated galvanized screen cloth and woven copper mesh. Highly corrosion-resistant. Low resistance to airflow (.07 in. W.G.) and extremely high dust-holding capacity.

Multi-V-Type: Low cost dry filter. Frame is of fiberboard. Ideally suited to domestic forced air systems as efficiency is comparable to more expensive industrial types. Cleanable. Maximum efficiency of heating plant is assured by use of this filter and newly-developed airflow indicator.

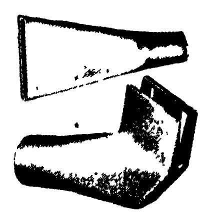

Multi-V-Type


Standard Panel Insert and Frame

Standard Panel: Dry-type Fin Construction, highest possible filtering efficiency. Heavy steel Panel Insert and Frame. Cleanable. Forty-two square feet of filtering area. Thousands of Staynew Standard Panel Units have given satisfactory service for years.

CLEANING PANEL-TYPE FILTERS

All Staynew filters are designed for greatest possible cleaning ease. Vacuum, compressed air or flushing with liquid solvents is recommended for dry-type, depending on kind of service. Special cleaning nozzles are supplied, as well as Staynew Cleaning Compound No. 6, made on a formula developed particularly for this use.


Cleaning Attachments

Representatives in Principal Cities

Complete Information on Request

FILTERS ALSO MADE FOR INTERNAL COMBUSTION ENGINES AND COMPRESSORS, AIR AND LIQUID LINES, ETC.

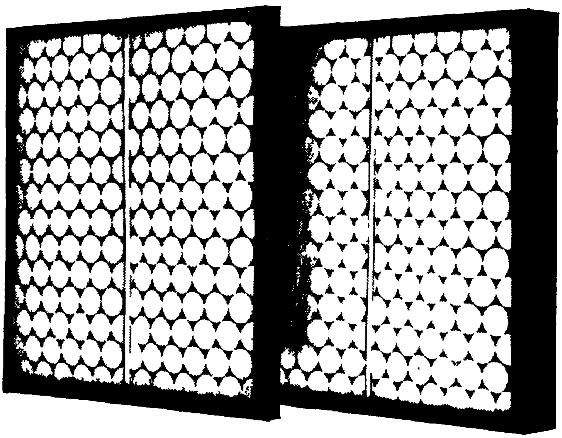
Owens-Corning Fiberglas Corporation

Toledo, Ohio

AIR FILTERS AND FRAMES FOR USE IN RESIDENTIAL, COMMERCIAL *and* INDUSTRIAL HEATING, VENTILATING *and* AIR-CONDITIONING SYSTEMS

FIBERGLAS* DUSTOP* AIR FILTERS

*Trademark Reg U S Pat Off



FIBERGLAS DUST-STOP AIR FILTERS will clean air streams of nuisance dirt, dust, and lint . . . and will do it economically and efficiently. Dust-Stops are made from compressed mats of glass fibers, sprayed with binder to hold the fibers in place. The fibers are then coated with a special dust-catching adhesive.

Available in Two Standard Types—Fiberglas Dust-Stop No. 1 (1 in. thick) is designed for greatest operating economy in commercial and industrial applications. No. 2 (2 in. thick) is recommended for use in unsupervised installations. It permits longer intervals between replacements. Both may be used in domestic applications.

Economical — Dust - Stop filters cost only 1¢ per CFM as original equipment, and less than 1/10th of 1¢ per CFM to replace. Further maintenance savings can be made by reusing filters after gently rapping out or vacuum cleaning excessive surface dirt accumulations. This practice may be repeated once or twice before the filter is discarded.

Engineering Service—Owens-Corning Fiberglas Corporation maintains offices in several metropolitan centers where representatives, qualified to assist in the planning of filter installations, are available for consultation.

Standard Sizes for Equipment*

Standard Sizes (Nominal)	Ratings		Average Resistance Inches Water Gauge Clean
	Cfm	Fpm	
20" x 25" x 1"	1000	300	.062
20" x 20" x 1"	800	300	.062
16" x 25" x 1"	800	300	.062
16" x 20" x 1"	640	300	062
20" x 25" x 2"	1000	300	13
20" x 20" x 2"	800	300	.13
16" x 25" x 2"	800	300	.13
16" x 20" x 2"	640	300	.13

*Other standard and any special sizes available

Literature—Data sheets on all standard Fiberglas products and applications will be furnished to engineers and manufacturers on request.

Owens-Corning Fiberglas Corp. Branches

ATLANTA, GA.	CINCINNATI, OHIO	NEW YORK, N. Y.
BOSTON, MASS	DALLAS, TEXAS	PHILADELPHIA, PA.
BUFFALO, N. Y.	DETROIT, MICHIGAN	PITTSBURGH, PA.
CHICAGO, ILL.	LOS ANGELES, CALIF.	ST. LOUIS, MO.
CLEVELAND, OHIO		WASHINGTON, D. C.

FIBERGLAS* DUSTOP* FILTER FRAMES

*Trademark Reg U. S Pat Off.

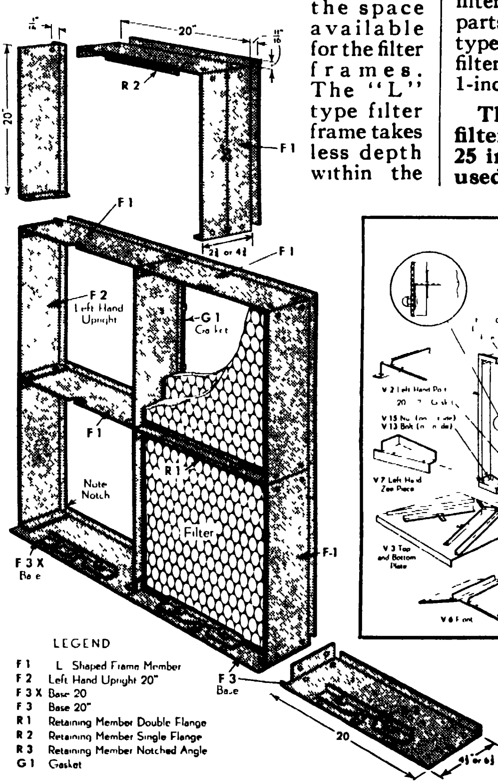
Fiberglas Dust-Stop "L" and "V" Filter Frame Assemblies are specified by engineers of commercial and industrial heating, ventilating and air conditioning. Frame members of heavy steel are assembled in combinations to satisfy any CFM and space requirement.

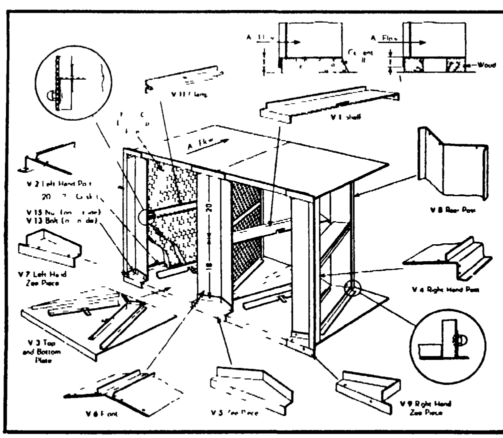
Both types of frames are designed for the convenient and correct handling of Dust-Stop filters They meet all Fire Underwriters' and local Fire Ordinance requirements.

The choice between the "L" type and "V" type frames is determined wholly by the space available for the filter frames. The "L" type filter frame takes less depth within the duct or plenum chamber but requires a larger face area for the same CFM capacity. The "V" type frame requires a face area approximately the same as the cross-sectional area of a duct which will handle the volume of air for which the filters are rated.

Two Depths of "L" Frames—The "L" frame, two filters deep, is designed to hold two Dust-Stop No. 1 filters in each cell. The "L" frame, four filters deep, holds four Dust-Stop No. 1 filters in each cell. The frame that is four filters deep is identical in every way to the frame two filters deep except that the depth of all parts is 2 in. more. When specifying "L" type frames indicate two-filter or four-filter depth. "V" frame is available, four 1-inch filters deep per cell, only.

The "L" frame uses 20 x 20 in. filters only. The "V" frame uses 20 x 25 in. filters only. Filters are always used two or more in series in each cell.





ABOVE—Dust-Stop "V" Type Filter Frame.

LEFT—Dust-Stop "L" Type Filter Frame.

H. J. Somers, Inc.

6063 Wabash Ave., **Detroit 8, Mich.**

Agents in All Principal Cities

SOMERS Heavy Duty Industrial Filter

All Welded Vee Type
Patent No's 2008800, 2130107

Somers Hair Glass Filters provide everything required in an efficient air-cleaning system

Consider These Features:

- **High rating for dust, soot and bacteria separation.**
- **Require no adhesive, coating or impregnation.**
- **Indestructible in normal service.**
- **Minimum low-pressure drop.**
- **Odorless and non-absorptive.**
- **Fireproof.**
- **Washable.**
- **Permanent—Do not rot nor disintegrate.**
- **All welded zinc-plated 20 ga steel frame.**
- **Metal protection strip on apex.**
- **Glass cloth between hot-dipped hardware cloth.**
- **Glass ribbon seal so air cannot short circuit.**

Somers Hair Glass Filters consist of a hot 20 gauge galvanized frame holding galvanized wire cloth packed with hair-spun glass strands The glass strands are flexible, do not break up and cannot be drawn into air stream.

Hair Glass being chemically inert, has no facility of absorption, it cannot rust and lasts indefinitely in service Water either hot or cold may be used to clean it, without impairing its efficiency.

These filters eliminate the necessity, the expense and the inconvenience of periodic replacement.

SOMERS WASHABLE AIR FILTERS All Welded Vee Type Stock Sizes

Frame Size Height and Length	Frame Depth	Filter Surface Sq. Inches	For Average Dry Filter Installations	Wet Application
12 x 12	2 7/8	288	288 C.F.M.	144 C F M.
15 1/2 x 24 1/2	3 1/16	1023	1023 C.F.M	511 C.F.M.
15 1/2 x 24 1/2	3 1/8	1674	1674 C.F M	837 C.F.M.
15 5/8 x 24 5/8	2	480	480 C.F M	240 C.F.M
15 5/8 x 24 5/8	3 1/8	1110	1110 C.F M	555 C.F.M.
16 x 20	2	384	384 C F M.	192 C.F M.
16 x 25	2	480	480 C.F.M.	240 C F M.
16 x 25	2 1/2	624	624 C.F M	312 C.F M.
16 x 25	3	864	864 C.F M	432 C.F.M.
16 x 25	3 1/8	1344	1344 C F M	672 C F M.
16 x 25	3 1/8	1440	1440 C F M.	720 C F M.
16 x 25	3 1/8	1632	1632 C F M.	816 C F M.
16 x 25	3 1/8	1056	1056 C F M	528 C.F M
18 x 18	3 1/16	864	864 C F M	432 C.F.M.
18 x 18	3 1/8	1134	1134 C.F M	567 C.F.M.
19 1/2 x 19 1/2	2	480	480 C F M	240 C F.M
19 1/2 x 19 1/2	3	819	819 C F M.	409 C F.M.
19 1/2 x 19 1/2	3	936	936 C F M	468 C.F M.
19 1/2 x 19 1/2	3	995	995 C F M	497 C F.M
19 1/2 x 19 1/2	3 1/8	1053	1053 C F M	526 C.F.M.
19 1/2 x 19 1/2	3 1/8	1170	1170 C F M	585 C F.M.
19 1/2 x 19 1/2	3 3/16	1696	1696 C F M	848 C F M
20 x 20	1 7/8	360	360 C F M	180 C F M
20 x 20	2	480	480 C F M	240 C F M.
20 x 20	2 1/2	600	600 C F M	300 C F M.
20 x 20	2 3/4	780	780 C F M	390 C.F.M.
20 x 20	3	840	840 C F M	420 C F M
20 x 20	3	960	960 C F M	480 C F M
20 x 20	3 1/16	1020	1020 C F M	510 C F.M
20 x 20	3 1/8	1200	1200 C F M	600 C F M.
20 x 20	3 1/8	1320	1320 C F M	660 C F M
20 x 20	3 1/8	1680	1680 C F M	840 C F M
20 x 25	2	600	600 C F M	300 C F M.
20 x 25	2 7/8	1020	1020 C F M	510 C F M
20 x 25	3 1/8	1560	1560 C F M	780 C F M
20 x 25	3 3/16	1800	1800 C F M	900 C F M
20 x 30	3 1/8	1800	1800 C F M	900 C F M
20 x 30 1/2	3 5/16	2400	2400 C F M	1200 C F M
23 x 20	3 1/8	1656	1656 C F M	828 C F M
23 1/2 x 23 1/2	3 1/8	1621	1621 C F M	1621 C F M.
26 x 34	3 1/8	2652	2652 C F M	1326 C F M
29 x 33 1/2	3 1/8	3045	3045 C F M	1520 C F.M
30 x 15	3 3/16	1800	1800 C F M	900 C F M

Other sizes from 9 1/2 x 30 to and inclusive of 31 in x 23 1/4 also available Send for complete stock size list
Frames zinc plated for 100 hour salt water spray test Refill may be inserted if necessary
Quotations and further engineering data, including master holding frame drawings will be sent on request

Just a **few** users of Somers Filters

Chemical Plants

American Viscose Co
American Zinc & Chemical Co
Davison Chemical Corp
Frederick Stearns Co

Automotive

Cadillac Motor Car Co
Chevrolet Motor Car Co
Chrysler Corp
Fisher Body Corp

Refrigeration and Air-Cond.

Frigidare Corp
Norge Div - Borg Warner
Kelvinator Corp
York Ice Machine Co

Ships

Amer Shipbuilding Co
U S S Saratoga
U S N Lake City, Florida
U S N Dayton Beach, Florida
U S N Vero Beach, Florida
U S N Jacksonville, Florida

Utilities and Municipalities

City of Kenosha
Michigan Consolidated Gas Co
Detroit Edison Co
New York Edison Co
Westchester Lighting Co

Dep't. Stores

S S Kresge Co
S H Kress & Co

Food Processing

Awrey Bakeries
Gilbert Chocolate Co.
Kellogg Co

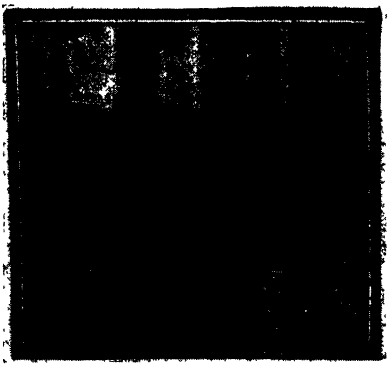
Manufacturers

Buffalo Forge Co
Burroughs Adding Machine Co
Clarage Fan Co
Curtis-Wright Airplane Co
Glensder Textile Co
Hoover Co
International Heater Co
Kearney & Trecker Corp
Killian Mfg Co
National Carbon Co, Inc
Pittsburgh Plate Glass Co
Rockford Machine Tool Co
Sunstrand Machine Tool Co.

Supreme Air Filter Company

126 West 21st Street, **New York 11, N. Y.**

SUPREME WASHABLE DRY TYPE AIR FILTERS
AUTOMATIC OR MANUAL CLEANING



Supreme Air Filter (Before and after cleaning)

Supreme Air Filters are made of a spun glass filter media, annealed under a patented process, and covered with a galvanized or copper wire mesh 25-06, encased in a galvanized or copper frame.

Supreme Filters offer a low resistance to the free flow of air, are light in weight and easy to handle. All cells are equipped with handles and interchangeable in the same bank.

Supreme Filters are made in flat and sawtooth types: The flat filter is encased in a ½ in. frame, for convenience in ordering and lower cost production The sawtooth type is corrugated the depth of the cell frame thereby increasing the filtering surface in a given area.

The spun glass in Supreme Filters is FIRE, ROT and VERMIN proof and not attacked by any chemical other than hydro-fluoric acid; it is approved by the Board of Fire Underwriters

The important factor in rating a filter is generally not the removal of a certain per cent of dust by weight but rather the effectiveness for taking out certain objectionable constituents, which may be either the finest particles present or the coarser ones within a certain range. The lower number per cent of dust particles in buildings and out-of-doors occur just after air washing by rain or snow storm, and the greatest number in busy streets on dry days.

Supreme Filters can be washed in hot or cold water, adding a little soap powder or solvent, and spraying with not over 25 lb. water pressure.

New mats can be supplied for all Supreme cell frames where mats become worn out or defaced, thereby saving the cost of a new frame.

TABLE OF SUPREME AIR FILTER SIZES

Sizes	C.F M	Mat, SQ"	EFFY	Pd, W G	Sizes	C.F.M	Mat, SQ"	EFFY	Pd, W.G
1"					3"				
16" x 20"	480	480			16" x 20"	1500	1500		
16" x 25"	480	480	94.5	14"	16" x 25"	1500	1500	97 1	18"
20" x 20"	480	480			20" x 20"	1500	1500		
20" x 25"	480	480			20" x 25"	1500	1500		
2"					4"				
16" x 20"	800	800			16" x 20"	2000	2000		
16" x 25"	800	800	95 7	16"	16" x 25"	2000	2000	98 0	.20"
20" x 20"	800	800			20" x 20"	2000	2000		
20" x 25"	800	800			20" x 25"	2000	2000		

Some Users of Supreme Air Filters

B Altman Company
Bendix Radio Corporation
Bethlehem Steel Corporation
Christ Hospital
Colgate Palm Olive Peet Company
Commercial Bank & Trust Company
Conde Nast Company
Dime Savings Bank
E I DuPont deNemours & Co , Inc
Fifth Avenue Hotel
Garment Capital Center
Geigy Dye Works

Koss Restaurant
Masonic Temple
Modern Industrial Bank
New Amsterdam Theatre
New York Central R R
New York Telephone Bldg Corp.
New York Trust
Ohio Power Company
Pabst Air Condg Corp
Panama Steamship Line
Posi Print Works
Riegel Paper Corp

Supreme Air Filters have been approved by the Maritime Commission for use on the C-2 ships.

Gulf Shipbuilding Co
Moore Dry Dock Co
Mantowac Shipbuilding Co
U S S A So Africa
International Tel & Tel.
Colonial Trust Co.
Western Electric Company.

Write Supreme Air Filter Company for complete information and prices

Badger CORPORATION

341 East Brown Street, **Milwaukee 12, Wisconsin**

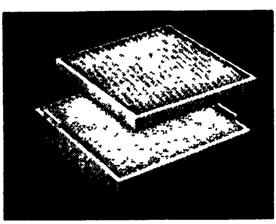
Manufacturers of PERMANENT Filters for Heating, Ventilating, Air Conditioning, Aircraft and Industrial Use

Badger designers and engineers have contributed many of today's most valuable developments in air filtration, dust, grit, and grease elimination in a wide variety of industries BADGER Permanent Type Filters are well known for their high efficiency, low air resistance, greater dust-holding capacity, and reduced maintenance cost on an increasing range of modern applications

BADGER Type P Air Filter . . . For Air Conditioning and Ventilating Systems

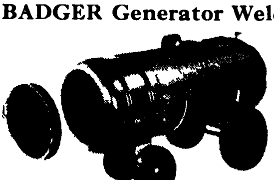
Permanent type, easy to clean, sturdy and strong, with special BADGER filter mesh design that holds dust through complete depth while providing minimum air resistance. Dual purpose expanded metal screen deflects air stream and protects filter media Vents in frame aid in quick cleaning, thorough draining Standard 1 in and 2 in. sizes from 640 C F.M. to 1000 C F M Special sizes to meet all applications. APPROVED BY UNDERWRITERS' LABORATORIES, Inc

BADGER Type HD Industrial Filter for Extra Heavy Duty and Industrial Air Cleaning Services



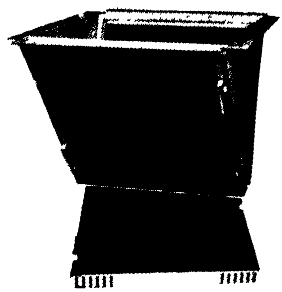
Specially designed for use on industrial applications, Badger Type HD is made in 2 in and 4 in sizes, and capacities from 640 cfm to 1000 cfm Permanent type, with high dust-holding capacity, effective, sturdily constructed filter media protected by deeply crimped layers of expanded metal screen, and heavy arc welded frame

BADGER Generator Welder FILTERS



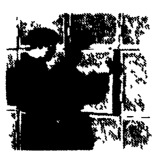
One of the most important recent developments for industry, the BADGER WELDER FILTER stops processing and atmospheric dust which often damage welding equipment Adaptable to most standard Generator Welding units, this filter increases rotor life, doubles brush life, minimizes repair shutdowns, improves welding efficiency, saves its low cost many times over. Permanent type, used on standard Generator Welders

BADGER Grease Filter



A new design in all-metal PERMANENT type filter that is an absolute necessity in kitchens, galleys, and wherever grease-laden air is a fire hazard and a maintenance problem. Filter media assures greater efficiency in air circulation and ventilation, increased working comfort and safety, and protection to motors, brushes and mechanical equipment Available in 2 in thickness, adaptable to a wide variety of applications

BADGER FILTER HOLDING FRAMES



Badger's special "Slip-Groove" holding frame design provides greater filtering capacity with a minimum of space. Compact, sturdy, extremely strong and rigid, this frame construction also simplifies filter removal and cleaning. Available in V-type, in straight banks, or adaptable to special requirements.

Complete bulletins, specifications, test and performance charts may be obtained on all BADGER Permanent Filters, as well as ready engineering assistance on special problems.

Westinghouse Electric Elevator Co.

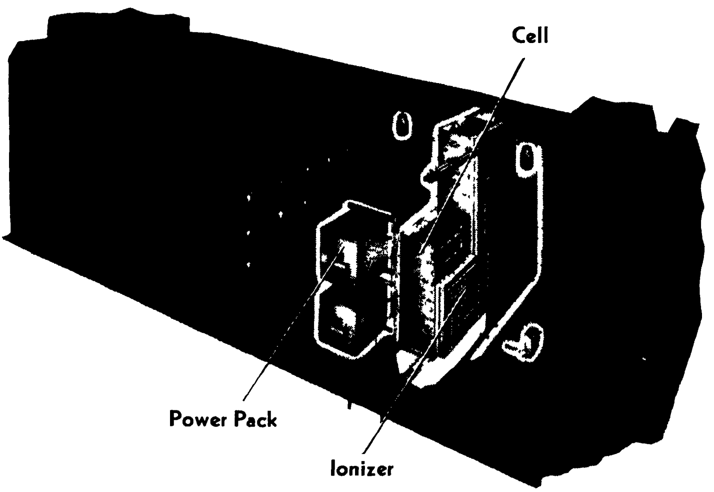
Precipitron Department

150 Pacific Avenue **Jersey City 4, N. J.**

THE PRECIPITRON*

First Commercially Successful Electronic Air Cleaner

Westinghouse Precipitron—the first commercially successful electronic air cleaner—now makes possible greater efficiency in removing dust, dirt, smoke and other foreign particles from the air. The Precipitron cells are simply installed in the forced-air or air-conditioning system and sealed so the air must pass through them (see illustration).



The Advantages of The Precipitron—Five major advantages make the Precipitron the complete answer to mass air cleaning in all buildings using forced ventilation or air-conditioning duct systems

Unmatched Efficiency-The Precipitron removes particles as small as 1/250,000 of an inch in diameter, and removes 90 per cent of all air-borne dust particles in the air stream

Easily Cleaned—New vertical collector plates make it still easier to clean the Precipitron Collected dirt is flushed harmlessly down the drain.

Safe—The Precipitron is approved by Underwriters' Laboratories without installation of additional fire protection apparatus.

Nothing to Clog—Precipitron creates only negligible resistance in the forced-air system under continuous operation

No Moving Parts—Precipitron has no more moving parts than a storage battery

Only Three Major Parts—The unit consists of three major parts, the improved Precipitron cell, the ionizer, and the power pack. Operation is entirely electronic and extremely efficient Dirt, smoke and soot passes through an electrostatic field, receives a positive charge, and is drawn to a collection plate of opposite polarity where it remains until flushed down the drain

Sizes—Precipitron is built in sizes to handle from 1200 cfm (for a single 24 in cell) to any desired volume through multiple cell arrangements Cells are available in two sizes 24 in x 24 in x 26½ in and 36 in x 24 in x 26½ in For 90 per cent efficiency, the 24 in and 36 in cells are rated at 1200 and 1800 cfm respectively For 85 per cent efficiency, ratings are 1500 and 2250 cfm Power packs are manufactured in two sizes Type S for installation up to five 36 in cells and Type L for five to sixteen 36 in cells

Information-For complete information on this new principle of cleaning air electronically, write Westinghouse Electric & Manufacturing Co., Box 868, Pittsburgh 30, Pa or contact your nearest Westinghouse office

*Trade-mark registered in U S A.

W. H. Wheeler, Inc.
234 East 46th Street, New York 17, N.Y.

airkem

TRADE MARK REG U S PATENT OFF

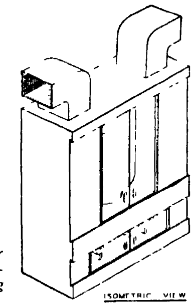
CHLOROPHYLL AIR FRESHENER

Air Quality Control For Air Conditioning By reducing odors and freshening air, AIRKEM chlorophyll air freshener provides indoor air quality control It adapts the principles used by Nature to keep air fresh in fields and forests It is the only product of its kind for industrial and commercial application which uses activated chlorophyll one of Nature's most effective agents for freshening air

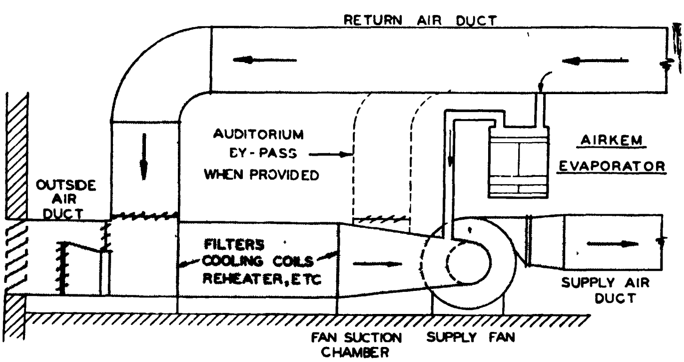
The efficacy of AIRKEM in producing a fresh-air effect has been proved by use in theaters, department stores, industrial plants, banks, hotels, restaurants, offices and in the most exacting of all fields - hospitals In addition, AIRKEM increases the effective capacity of air conditioning systems by allowing the fresh air intake to be substantially reduced, with corresponding reductions in both summer and winter load, and savings in fuel, water and electric power

Easily Added To Systems It is easy and inexpensive to engineer AIRKEM into new systems or add it to present air-conditioning or ventilating systems. Part of the recirculated air is diverted and passed through an AIRKEM Transpar Evaporator, where controlled amounts of AIRKEM are evaporated into this air The Transpar Evaporator has a system capacity of from 1,000 to 36,000 CFM of delivery air, and does not increase the static pressure Both first and operating costs are low compared with the benefits obtained and the savings effected.

AIRKEM is available immediately, without priority Write for full information and distribution opportunities.



Transpar Evaporator, for application to air-conditioning and ventilating systems



The AIRKEM Transpar Evaporator is located so as to reduce odors arising in the system itself, as well as the occupancy odors of the air-conditioned space

American Moistening Company

ESTABLISHED 1888

Providence 1, R. I.

ATLANTA 2, GA. BOSTON 9, MASS. CHARLOTTE 1, N C.

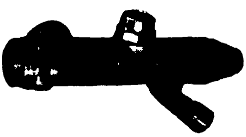
UNIT HUMIDIFYING AND AIR CONDITIONING EQUIPMENT

A few of many AMCO products with a Long Record of Dependable Performance

Self-cleaning Atomizers
Sectional Humidifiers.
Amtex Humidifiers.
Hand Sprayers.
Mine Sprays.

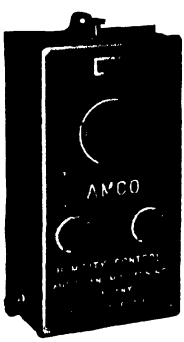
Fabric and Paper Dampeners
Mechanical Psychrometers
Electro Psychrometers
Sling Psychrometers
Hygrometers.

The Amco line of devices for the supply, maintenance and control of humidity is complete in its ability to meet any presented problem of applied humidification Used independently or as an adjunct to Central Station equipment, these devices automatically maintain any required humidity condition in a capable uniform performance.



AMCO ATOMIZER—No. 5

Quality and quantity of spray are maintained even under adverse conditions because this atomizer is automatically self-cleaning. When the compressed air supply is shut off, either manually or in response to a humidity control, *both* air and water nozzles are thoroughly cleaned.



AMCO HUMIDITY CONTROLS

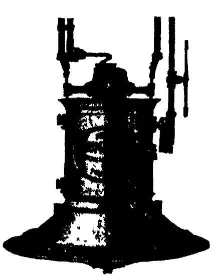
Compressed Air Operated

An extremely accurate and active device operated by compressed air which assures a regulation of humidity within exceedingly close ranges

AMCO HUMIDITY CONTROL

Electrically Operated

Similar in principle to the Compressed Air Type except that the hygroscopic element operates electrical contacts which control the units.



IDEAL HUMIDIFIERS—Senior Type

A high capacity unit for use where conditions require a great amount and good distribution of moisture. Motor driven fan gives wide distribution of atomized spray. Amco heads serve the triple purpose of humidifying, air washing and cooling.

IDEAL HUMIDIFIERS—Junior Type

Similar in construction to Senior Type. Used where medium capacities are required.

April Showers Company

4126 Eighth Street, N.W. **Washington 11, D. C.**

(Trade Mark Reg. U. S. Pat. Office)

AUTOMATIC EVAPORATIVE ROOF COOLING

"Right as Rain" and "The Best Insulation Under the Sun" were phrases Given to APRIL SHOWERS by our customers

Distributors and Dealers in Principal Cities

DEALERS and CONTRACTORS—AN ANNOUNCEMENT of PROFIT

APRIL SHOWERS is the trade name of an EFFICIENT fool proof method of preventing roof heat penetration or solar infiltration causing excessive heat in upper floors and in factories, stores, theatres, shops, etc

City water under normal city pressure is usually adequate to serve your APRIL SHOWERS system. The sun operates it. Roofs LAST LONGER, greater comfort is assured, production during the summer's heat is kept apace, errors through fatigue are avoided, and MANAGEMENT acclaims APRIL SHOWERS a GOD-SEND.

Any established air conditioning engineer, contractor, dealer or any plumber and kindred workmen can obtain APRIL SHOWERS supplies, plans and instructions for erecting APRIL SHOWERS for you or your clients Write for "INTRODUCING APRIL SHOWERS" which is a loose-leaf bound book for Installers and Dealers

Contractors, Dealers and Installers be sure to share in the APRIL SHOWERS post-war prosperity. You are not required to inventory any merchandise. You purchase only such equipment as is needed for each installation You set your own SELLING PRICE You share PROFITS with NO ONE.

- **Solar radiation** is converted to cooling effect, reducing normal heat transmission 70 per cent upward. Entire roof surface temperature is normally held at wet bulb temperature when evaporative factors are favorable. Transmission of solar heat through glass skylights is reduced as much as 85 per cent.
- **Fire Prevention** from external sources is obtained through installation of manual emergency switch. Roof is quickly and completely sprinkled at will, putting out fire-brands, sparks, embers, and maintaining a wet roof.
- **For Cooling Automatically** upper level, floors, lofts, rooms of buildings. For giant stores, theatres, amusement palaces, stadiums. APRIL SHOWERS is self operating by use of an electric thermal control placed upon the surface of the roof in the SUN. Evaporative cooling effect of liquid applied turns system off. Operating cycles repeat as roof temperature calls for cooling. Water may be used from city mains, wells, or waste water from condenser units.
- **Water Consumption** can be adjusted to approximately twenty gallons per day for 1,000 sq. ft.

Air conditioning and plant engineers are agreed that APRIL SHOWERS fills a definite need where its use is indicated.

Hundreds of installations, from Boston to Los Angeles have been made. Write for information and address of nearest distributor.

Inquiries will be answered promptly. Estimates free upon request.

APRIL SHOWERS controlled roof cooling is protected by U S. Patents; beware of imitators or infringers.

Developed in 1933-34.

Lilie-Hoffmann Cooling Towers, Inc.

Exclusive Builders of Cooling Towers for 45 Years

4239 Duncan Ave., **St. Louis 10, Mo.**

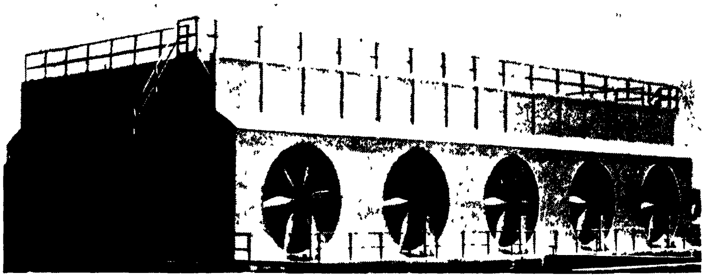
Two Modern Plants—St. Louis, Mo and Plainview, Texas

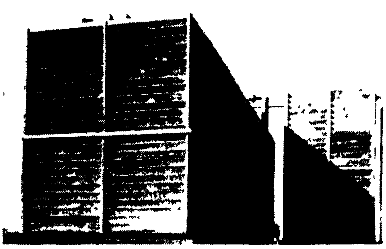
INDUCED DRAFT TOWERS

New type filling reduces static pressure; uniformly distributes air and water over effective tower area. Ring gravity distribution system requires only one riser pipe. Each cell main trough equipped with Weir gate, giving individual cell control. Teco timber connectors develop 100 per cent working stress of members.

FORCED DRAFT TOWERS

Three-tier zigzag pattern spray eliminator system cuts drift loss to a minimum. Fan opening covered by galvanized screen wire Built in single or multiple cells. Continuous design improvement, based on installation studies, presents record of unfailing operation.



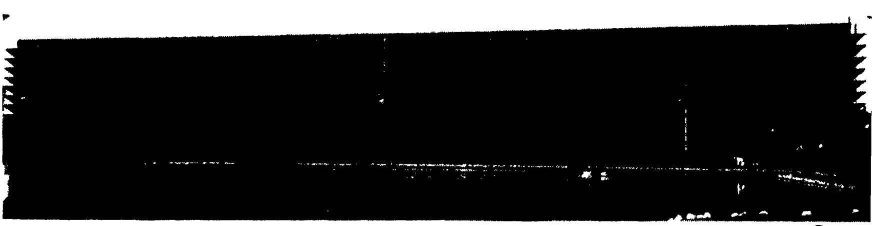


ATMOSPHERIC SPRAY TYPE

Generally offered in capacities up to 300 gpm. Distribution by galvanized pipe header with smaller lateral arms, equipped with non-clogging spray nozzles Louvres fit into mast slots—no nails required Recommended where cost is prime factor, and close approach to wet bulb is unnecessary. Rigidly braced and tied together to prevent warping and buckling.

ATMOSPHERIC DECK

Distribution by either wood trough, or wood pipe header system. Removable deck grids. Special L-H drift baffles on deck supports prevent entrained water loss Generally 12 ft wide—can be made any length or height.

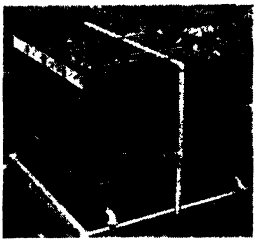


The Marley Company

(Fairfax and Marley Roads,) **Kansas City 15, Kansas**

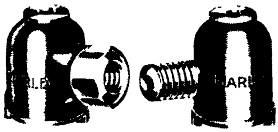
Branches or Agents in Principal Cities

Spray Nozzles and a Complete Line of Water Cooling Equipment

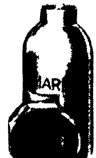


MARLEY NATURAL DRAFT TOWERS

Practically unlimited range of closely graduated sizes, entirely shop fabricated. Minimum initial, maintenance and operating costs Many exclusive MARLEY advantages. **Bulletins 201 and 202.**



MARLEY Small 2-Piece Nozzles for Brine Spraying, Air Washing and Similar uses.



MARLEY 1-Piece Nozzles for Cooling Towers, Spray Ponds, etc

MARLEY Humidifying Nozzle adds moisture to air in open rooms or duct system

MARLEY PATENTED NON-CLOG SPRAY NOZZLES

Made in scores of types and sizes. Practically any metal or alloy the purpose may demand. **Bulletins 101, 102 and 103.**

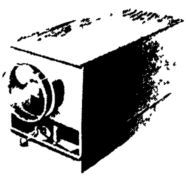
LARGE MARLEY INDUCED DRAFT TOWERS

Both Conventional and Double-Flow types, for heavy duty water cooling services of all kinds Any capacity, with one fan or many, individually engineered to the exact requirements of each installation. MARLEY patents cover a variety of important features for extreme operating flexibility, high efficiency and economy

Redwood is the standard structural material Steel, transite and other special materials require much higher priorities for the war's duration

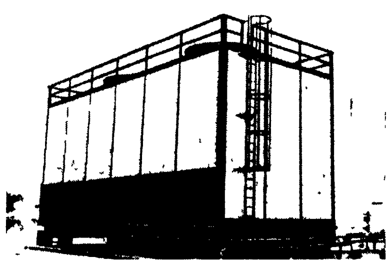
Conventional Induced Draft (pictured above in steel construction; below right, of Redwood) well adapted to most large capacity services **Bulletin 603.**

Double-Flow Induced Draft (below left) for unusually large capacity services **Bulletin 700-A.**



SMALL INDUCED DRAFT TOWERS

Small, self-contained, steel units for 2 to 170 ton service, to go indoors or out Horizontal air flow Smaller sizes shipped all assembled Larger sizes are entirely shop fabricated for fast, simple assembly at the location. **Bulletins 504 and 505.**



Also Many Other Types of Towers, Spray Ponds and Related Equipment

Water Cooling Equipment Company

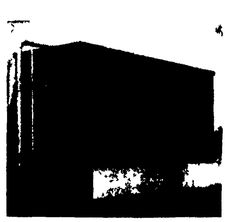
8613 New Hampshire Avenue

Affton Station, St. Louis 23, Missouri

MANUFACTURERS OF MECHANICAL DRAFT AND ATMOSPHERIC COOLING TOWERS

FORCED DRAFT COOLING TOWER

41,500 g.p.m. twenty-two cell, heavy duty, double forced draft cooling tower . . . equipped with twenty-two 12-foot diameter adjustable pitch, propeller type fans.



REDWOOD "WATERFALL" ATMOSPHERIC SPRAY COOLING TOWER

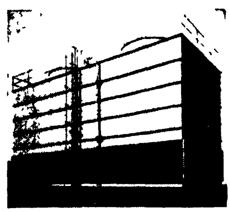
Factory fabricated and shipped knocked-down with all hardware, spray headers and spray nozzles. Complete erection instructions and drawings are furnished to assist in the assembly of the tower.

These towers are portable and can be knocked-down and moved simply by removing the louvres and bolts. Shipment can be made from stock. Write for Bulletin 125-A.

REDWOOD ATMOSPHERIC DECK TYPE COOLING TOWER

PRODUCTS

Heavy duty forced and induced draft cooling towers, standard induced and forced draft towers, induced draft coil towers, atmospheric deck cooling towers, atmospheric spray towers, spray ponds and spray nozzles.

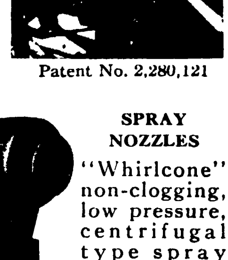


INDUCED DRAFT

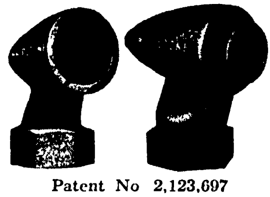
This type of cooling tower is recommended for installations where noise is a prime factor of consideration. The noise of the mechanical items is carried upward and discharged into the air The mechanical draft cooling tower assures a positive cooling of the water to a specified temperature independent of wind velocity.

TIMBER JOINT CONNECTOR

Special cast iron timber joint connector used to develop the full strength of the timber joint, an accomplishment almost impossible to achieve by bolting.



Patent No. 2,280,121



Patent No 2,123,697

SPRAY NOZZLES

"Whirlcone" non-clogging, low pressure, centrifugal type spray nozzles for spray ponds, spray towers and other uses. Write for Bulletin 76-A.

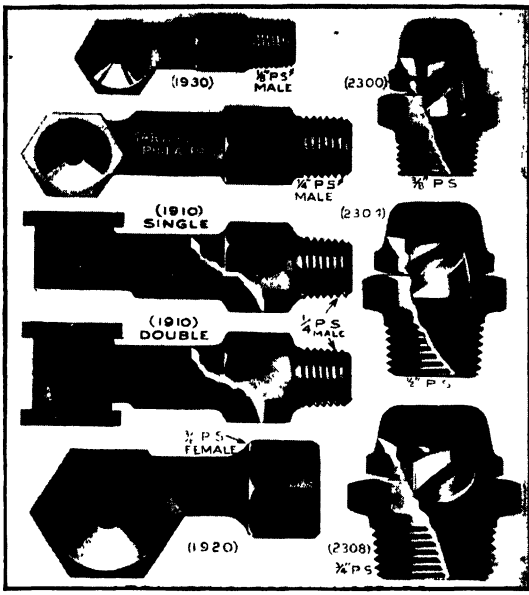
ENGINEERING

Design and construction are based on sound engineering principles to meet specific requirements for cooling performance and structural strength. Redwood, steel or other suitable materials are used.

Jos. A. Martocello & Company

229-31 North 13th Street, **Philadelphia, Pa.**

ATOMIZING SPRAY NOZZLES

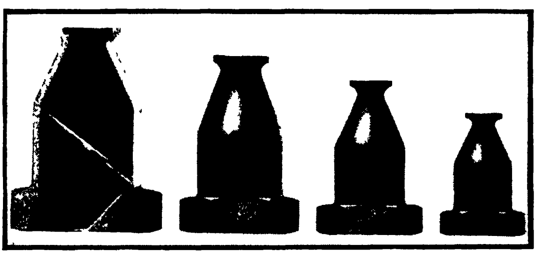


MARTOCELLO

Atomizing Spray Nozzles produce a uniform, good, wide spray with less friction and at minimum pressure requirements.

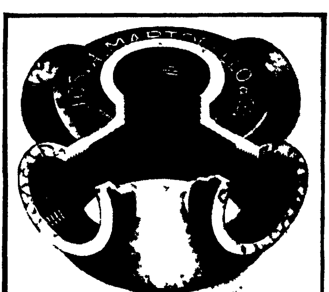
Nozzles illustrated at the right are manufactured with precision in Brass from Forgings and Bar Stock. Their design has been thoroughly tested for results and durability and will give you satisfaction.

Successful, Efficient Results depend largely upon selecting the proper number and type of Nozzles suitable for your job. Therefore we suggest you send us your specifications as we also have several Types other than illustrated and we will gladly assist you to obtain the most efficient application.



MARTOCELLO

Spray Pond Nozzles of a sturdy one-piece construction —cast of High Grade Red Brass with Inlet and Outlet accurately machined, are less clogging, offer less friction and give best overall efficiency.



MARTOCELLO CLUSTER CASTINGS

Sturdy Grey Iron Construction with large area for reduced friction and even distribution and are Hot Dipped Galvanized after fabrication.

Furnished with **Nozzles** and continuous Standard Steel **Long Sweep** Galvanized Pipe **Spray Arms** and **Center Nozzle** Nipple in accordance with layout required.

Sizes Carried in stock for **Prompt Shipment** 1½ in. P S. Outlet, 3 in. P. S. Inlet and 2 in. P. S. Outlets, 4 in. P. S. Inlet Cluster Castings.

WRITE, PHONE OR WIRE

For Bulletin listing Capacities and Prices.
Prompt shipments from stock.

Acme Industries, Inc.

Jackson, Michigan

Representatives in Principal Cities

REFRIGERATION AND AIR CONDITIONING EQUIPMENT

AMMONIA CONDENSERS
FREON CONDENSERS

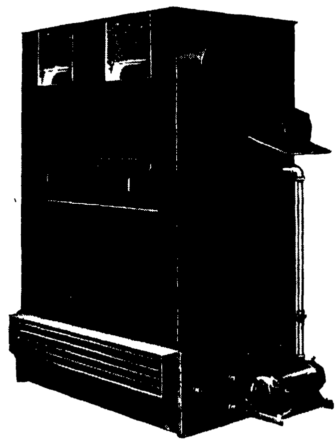
For use with Ammonia, Freon, Sulphur Dioxide or Methyl Chloride Tailor-made to order to meet varying water temperatures available and condensing temperature desired Sizes range from fractional tons up.

For Ammonia—Ask for Catalog No 21
For Freon—Ask for Catalogs Nos 23 and 24

DRY-EX WATER CHILLERS

Water chillers that are especially designed for recirculating systems—air conditioning of indirect type, processing, etc. Refrigerant in tubes—no bends; controlled water velocity; small refrigerant charge.

Ask for Catalog No 54



EVAPORATIVE CONDENSERS

Combine desirable features of cooling tower or spray pond and the water-cooled condenser Heavy galvanized iron casing with bitumastic rust resisting paint coating inside. Rests on heavy sheet steel base. 11 Models—capacities up to 100 tons

Ask for Catalog No 27

ACME HEAT INTERCHANGERS

For use with Freon, Methyl Chloride and other refrigerants Sizes suitable for use with varying imposed loads, refrigerant liquids, and gas temperatures

Ask for Catalog No 12

ACME ALSO MANUFACTURES: FLOODED SHELL AND TUBE COOLERS, PIPE COILS, BRINE COOLERS, HI-PEAK WATER COOLERS, AIR CONDITIONING COILS, FORCED CONVECTION UNITS, OIL SEPARATORS, LIQUID RECEIVERS, ACCUMULATORS, SURGE DRUMS, SPECIALTIES.

Ask for Catalogs

You don't have to choose between quality and price—BUY ACME

AEROFIN CORPORATION

410 So. Geddes Street

Syracuse 1, N. Y.

AEROFIN

Standardized Light-weight Heat Exchange Surface

Branch Offices

CLEVELAND, CHICAGO, NEW YORK, PHILADELPHIA, DETROIT, DALLAS, TORONTO

Aerofin is the modern Standardized Light-Weight Encased Fan System Heating and Cooling Surface originated by *Fan Engineers* to meet the present and future requirements of this highly specialized field All Standard AEROFIN Units are furnished as completely encased Units, ready for pipe and duct connections. The patented casings are built of pressed steel and are exceptionally strong and rigid, protecting the Unit from all the strains of pipe connections and expansion or contraction in service The casings are flanged on both faces, top and bottom, and template punched for bolting together adjacent Units, or for duct connection

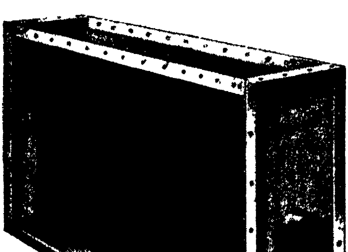


Fig 1

Aerofin Non-freeze heater (Fig. 1) is non-freeze, non-stratifying spiral fin coil built into casing for air conditioning units or for installing in ducts May be installed horizontally or vertically. Used on any two-pipe steam system for preheating or reheating Modulating control on preheaters.

Available in 13 lengths and 3 widths, from net face area of 2 76 sq ft to 26 28 sq ft.

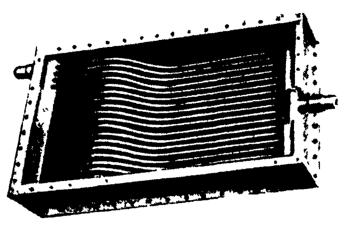


Fig 2

Flexitube Aerofin (Fig. 2) is distinguished from all other developments by its off-set tubes, so arranged as to absorb all expansion and contraction strains

Headers—Steel

Tubing—5/8 in. O D copper, admiralty or aluminum.

Joints—Where admiralty or copper tubes are used together with bronze or steel headers tubes are brazed to headers. Where both aluminum tubes and headers are used tubing is welded to headers.

Casings—Copper, aluminum or galvanized iron.

Design—Constructed with headers on opposite ends making possible installation of units with tubes horizontal or vertical.

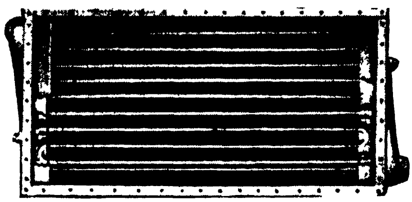


Fig 3

Universal Aerofin (Fig. 3) is distinguished by its "S" bend construction of tubing, units designed with steel headers on opposite ends, the ends of the "S" bends being connected thereto by compression nuts, the bends taking care of the expansion and contraction of the tubing.

Recommended where close control is desired.

Headers—Pressed steel

Tubing—1 in. O.D. Copper or admiralty.

Casings—Copper, aluminum or galvanized iron.

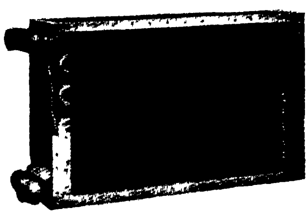


Fig 4

High Pressure Aerofin (Fig. 4) is of continuous tube design, being recommended where extremely high pressures of steam are used.

Headers—Pressed steel.

Tubing—1 in. O D. Copper or admiralty.

Casings—Copper, aluminum or galvanized iron.

Fig 5

Booster Aerofin—straight tube type, single pass construction for pressures from 1 to 200 lb gauge.

Headers—cast bronze.

Tubing—⅝ in. O.D. copper

Casings—copper, aluminum or galvanized iron. Recommended where small coils are needed or to raise the air temperatures in branch ducts

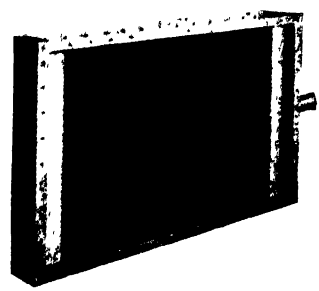


Fig 6

Narrow Width Aerofin: (Fig. 6) recommended for water cooling or for flooded Freon systems. Made in straight tubes only with headers on opposite ends, joints between headers and tubing being brazed Construction similar to Flexitube AEROFIN.

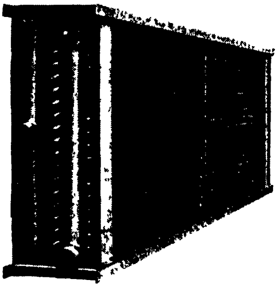


Fig 7

Aerofin Continuous Tube Water Coils (Fig 7) are designed for air cooling by circulating cold water through the AEROFIN and air over extended fin surface. Made for either horizontal or vertical air flow

Tubes and fins are copper, completely tinned with permanent metallic bond between fin and tubes. Headers are made of one-piece cast bronze and casings of heavy galvanized iron or copper

Units tested to 1000 lb hydrostatic pressure.

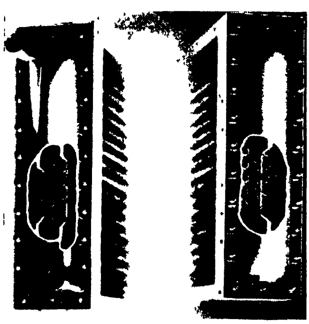


Fig 8

Aerofin Cleanable Tube Units (Fig 8) for cooling only made with headers removable to permit cleaning tubes. Recommended for use where sediment or scale forming chemicals are present in the cooling water

Headers—Cast iron

Tubing—Copper or admiralty.

Casings—Copper or glavanized iron.

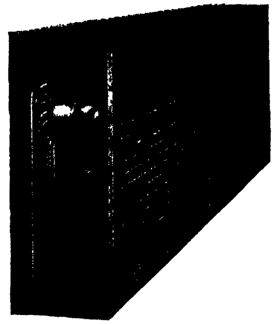


Fig. 9

Aerofin Direct Expansion Units: (Fig. 9) Centrifugal Header Type—Recommended where control of rows in direction of air flow is not required.

AEROFIN Sizes

Flexitube: 13 standard lengths, three widths, one and two rows deep.

Narrow: same as Flexitube.

Universal: 17 standard lengths, two widths, one and two rows deep.

ContinuousTube: 13standard lengths, three widths, 2-3-4-5 and 6 rows deep.

Cleanable Tube: 17 standard lengths, one width, 2 and 4 rows deep.

Direct Expansion: Centrifugal Header -- 11 standard lengths, three widths, 2-3-4-5-6 rows deep

Steel Supporting Legs: 18 in and 24 in high. Punched same bolt hole centers as standard casings Quickly attached. No other foundation required.

Sale: AEROFIN is sold only by manufacturers of nationally advertised Fan System Apparatus List upon request.

Write Syracuse for Heating Bulletin G-32; Direct Expansion Bulletin DE-34 on refrigeration type units; Continuous Tube Bulletin C. T. 34 for Water Cooling Coils; or phamplet on Cleanable Type AEROFIN for cooling.

The G & O Manufacturing Company

138 Winchester Avenue **New Haven, Connecticut**

SQUARE FIN TUBING

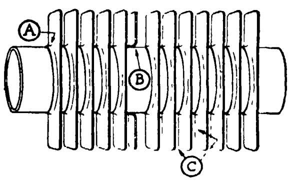
STRAIGHT LENGTHS—U-BENDS—CONTINUOUS COILS

THE use of INDIVIDUAL fins results in high efficiency in heat transfer from primary tube surface to secondary fin surface.

Fins of any size or shape may be obtained giving any desired proportion of primary and secondary surface.

A square fin has about 30 per cent greater surface than a round fin of a diameter equal to one side of the square.

Individual fins permit of any fin spacing; also, of using fins in groups at intervals along tubes.



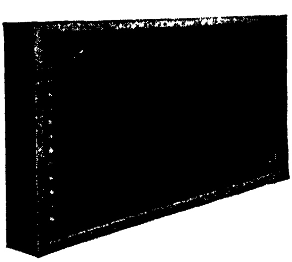
A—Generous Fin Collar provides large contact area between Tube and Fin.
B—Tube expanded against Fin Collar; insures mechanically tight joint, made permanent by bond of high temperature alloy—complete thermal contact.
C—Free air-flow passages; non-clogging

STANDARD SIZES

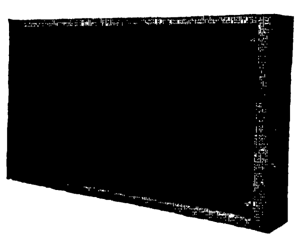
O.D of Tube	Fin Size	Fin Spacing per Inch	Surface per Linear Foot
3/8"	7/8" sq.	6	0 80 sq ft.
3/8"	7/8" r'd.	6	0 60 sq ft.
5/8"	1 1/8" r'd.	6	0 87 sq ft.
3/4"	1 1/2" r'd	6	1 55 sq ft
3/4"	1 5/8" sq	6	2 40 sq ft
1"	2 1/8" sq	6	4 00 sq ft
1 3/8"	2 3/8" r'd	4	2 33 sq ft

RADIATING ELEMENTS FOR ALL HEAT TRANSFER PURPOSES

G&O Finned Radiation Coils for industrial applications are available in a wide range of sizes.



Universal U-102



Standard No 10

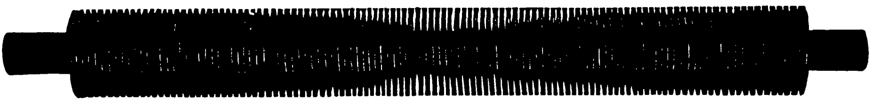
Send for Catalog and Price List

The Rome-Turney Radiator Co.

Rome, N. Y.

ROME

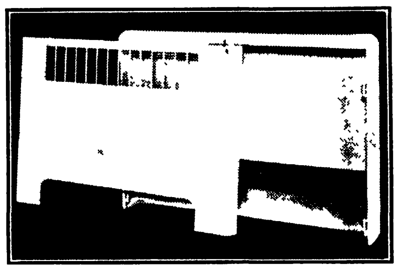
Helical Fin Tubing



Over 100 sizes for all air heating and air cooling equipment straight lengths—U bends—Continuous Coils

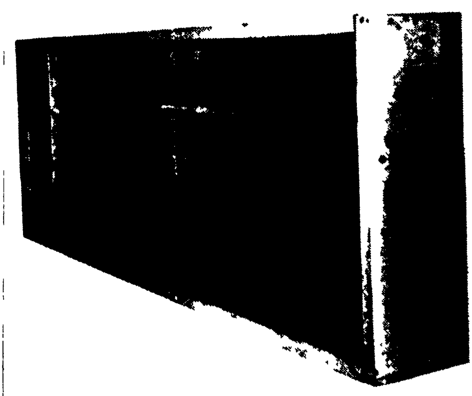
Blast Air Heating and Air Cooling Coils—Complete range of sizes- Heat transfer and Engineering data by recognized authorities Special Construction to suit applications

Helical Fin Tube Core Assemblies for heat transfer equipment

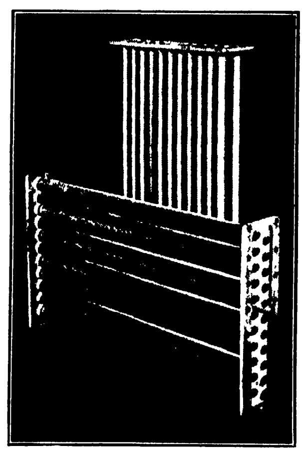


Copper Convector

Rome Copper Convectors for Concealed heating of Homes, Office Buildings and Apartments--many thousands in use Complete line of sizes and cabinet models



Blast Air Heating and Air Cooling Coils



Helical Fin Tube Core Assemblies

Rome products are "backed" by 40 years of experience.

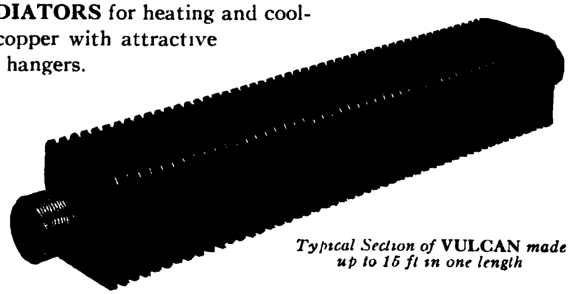
The Vulcan Radiator Company

26 Francis Avenue Hartford 6, Conn.

MANUFACTURERS OF FINNED TUBES FOR TWO DECADES

VULCAN FIN-TYPE RADIATORS for heating and cooling are available in steel or copper with attractive grille covers, wall and ceiling hangers.

VULCAN is used in railroad cars, ships, hospitals, churches, schools, homes and thousands of industrial concerns for the following reasons:



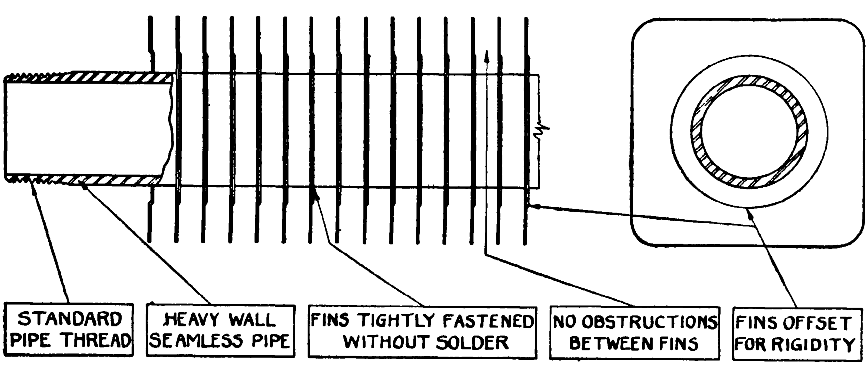
Typical Section of **VULCAN** *made up to 16 ft in one length*

1. It is an *Accepted Product* for performance.
2. Compactness and light weight make installation easy
3. Uniform heat distribution has been proved by tests,—hot and cold drafts are eliminated.
4. Installation costs are reduced since 500 sq ft can be on one continuous line.

Heat Output for 2 in. IPS is 5.25 sq ft per lineal foot with 1 lb steam and air at 70 deg, for 1¼ in. IPS—4 25 sq ft

VULCAN CAN BE INSTALLED 2 or 3 tiers high to secure necessary heat requirements.

CONSTRUCTION DETAILS OF STEEL RADIATOR



Contact Representatives in Principal Cities or send for Catalog

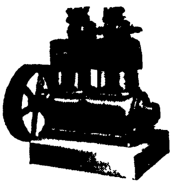
Baker Ice Machine Co., Inc.

SALES AND SERVICE IN PRINCIPAL CITIES

Omaha, Nebraska
1530 Evans Street
Cable Address: BAKERICE

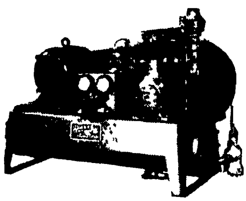
BUILDERS OF DEPENDABLE REFRIGERATION EQUIPMENT SINCE 1905

• BAKER builds compressors, condensing units, compressor units, shell and tube equipment, and cooling units for all commercial and industrial air conditioning applications BAKER engineers are free to select the equipment and refrigerant best suited to your specific requirements Write for information, descriptive literature on units shown below.



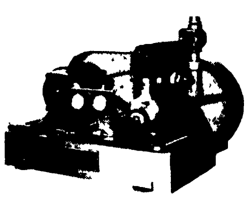
Methyl Chloride or "Freon-12" Compressors

• Four-cylinder type, available in sizes from 3 hp to 60 hp Semi-steel cylinders and pistons. Counter-balanced crankshaft, precision ground. Timken roller main bearings Full force feed lubrication is furnished by gear type oil pump



Methyl Chloride or "Freon-12" Water-Cooled Condensing Units

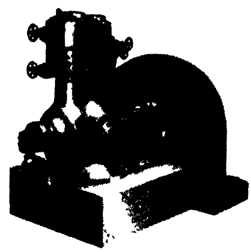
• Complete line of self-contained, automatic units Sizes range from 1 hp to 20 hp capacity. Two- and four-cylinder types Shell and tube condenser-receiver.



Methyl Chloride or "Freon-12" Compressor Units

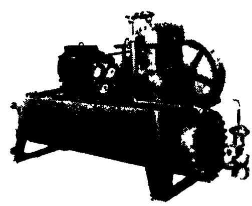
• Arranged for use with evaporative type condenser or separately mounted shell and tube condensers Sizes range from 3 hp to 60 hp. Two- and four-cylinder types Automatic controls.

Ammonia Compressors (3 to 100 hp)



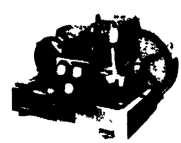
• Vertical enclosed, single-acting type. Can be installed in multiple installations. V-belt drive or direct connection to motors or engines Double suction and capacity reduction in larger sizes

Ammonia Water-Cooled Condensing Units (3 to 15 hp)

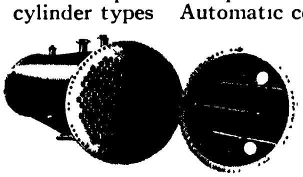


• Excellent for all industrial applications. Shell and tube type condensers with pressure-operated water control valve. Compact design economizes floor space.

Ammonia Compressor Units (3 to 20 hp)



• Designed for quick installation and ready accessibility to all unit parts For use with evaporative or separately mounted shell and tube type condenser Full enclosed automatic motor control with overload and low-voltage protection Two- or four-cylinder compressor, V-belt drive.



Baker Shell and Tube Condensers and Liquid Coolers for Use with Ammonia or "Freon-12" (1 to 250 tons capacity)

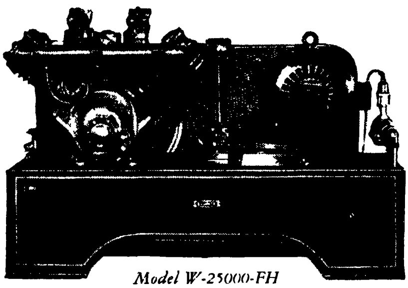
• Made in sizes up to 2500 square feet of cooling surface in single shells. Available for multi-unit installation with special stands to allow compact installation. Supplied in either horizontal multi-pass or vertical type. Heads may be removed quickly and easily for complete cleaning Tubes are spaced to provide even gas distribution throughout the shell. All valves of maximum size for greatest efficiency.

Baker Also Manufactures a Complete Line of Industrial-Type Cooling Units, Ammonia Valves, Screw-End Fittings, Capped Valves, Flanged-Type Fittings

SPECIFICATIONS AND SIZES SUBJECT TO CHANGE AS REQUIRED BY GOVERNMENT REGULATIONS

BRUNNER MANUFACTURING COMPANY

UTICA, N. Y., U. S. A.



Model W-25000-FH

COMMERCIAL REFRIGERATION

The Brunner Line of Refrigeration Equipment includes Air Conditioning models up to and including 25 hp for all types of high temperature applications within their capacity, using either "Freon-12" or Methyl Chloride as refrigerant.

BRUNNER DEPENDABILITY is based on time-proven features of design and manufacture . . . all parts are precision machined with extremely close tolerances . . . bronze bearings throughout . . . extra large fin surface on cylinders and heads . . . bellows seal . . . silent eccentric drive (except on 20 hp and 25 hp models, which employ crankshaft) . . suction and discharge valves in "all-in-one" plate assembly . . . heavy-duty motor with high starting torque . . . adjustable motor base . . multiple V-belt drive. Throughout, Brunner Refrigeration Units are geared to the demands of *heavy-duty* service

SPECIFICATIONS					CAPACITIES Air Conditioning Units Based on 75° F Water Temperature Freon-12 Refrigerant	DIMENSIONS
Model No	H P	Cyls	Bore and Stroke	R P M	B T U per Hr 40° Evap Temp	L W H
W 300-FH	3	4	3¼ x 2¼	260	38547	50″ 24″ 28¾″
W 500-FH	5	4	3¼ x 2¼	420	62270	50″ 24″ 28¾″
W 750-FH	7½	4	4¼ x 3	260	91526	71″ 29½″ 38½″
W 1000-FH	10	4	4¼ x 3	350	123211	71″ 29½″ 38½″
W 1500-FH	15	4	4¼ x 3	525	184815	71″ 29½″ 38½″
W20000-FH	20	4	4¼ x 5	435	255046	73″ 33¼″ 48¾″
W25000-FH	25	4	4¼ x 5	540	316652	73″ 33¼″ 48¾″

Additional air and water cooled models from ¼ h p for commercial and industrial applications

The Brunner Field Sales Organization is available in all parts of the country, backed by outstanding achievements in engineering, and adoption of modern methods and design of air conditioning equipment.

Installation of Brunner refrigerating units is insurance of the finest quality materials and workmanship—plus the highest efficiency possible in modern design and manufacture.

FREE... COMPLETE ILLUSTRATED CATALOG

with large section devoted to ways of selecting the proper units for any application.

Curtis Refrigerating Machine Division

of Curtis Manufacturing Company

1959 Kienlen Ave., **St. Louis 20, Mo., U. S. A.**

ESTABLISHED 1854

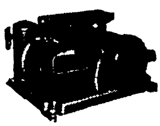
Full Line of Units from 1/6 to 50 hp

CURTIS REFRIGERATION AIR CONDITIONING AND COMMERCIAL

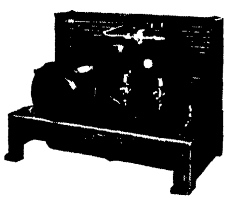
Unit Coolers and Evaporator Coils

PRODUCTS: Refrigerating Machinery; Forced Draft Cooling Units; Cooling Coils, Condensers, Shell and Tube Coolers, Valves, Fittings and Accessories, Complete Refrigerating Equipment for Dairies, Creameries, Ice Cream Cabinets, Ice Cream Making Plants, Cold Storage Locker Systems, Walk-in Coolers, Drinking Water Systems, Commercial and Low Temperature Cooling, Processing and Air Conditioning Installation, Packaged and Remote Types.

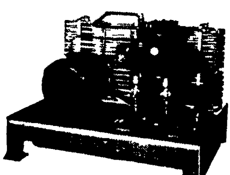
Commercial Refrigeration



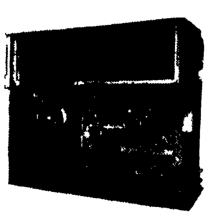
¼ to ½ hp Self-Contained Condensing Unit



1½ hp Air Cooled Condensing Unit Other sizes from ¼ to 5 hp



5 hp Water Cooled (Counterflow) Condensing Unit Other sizes from ⅓ to 5 hp



7½-10-15 ton Remote or Central Type Air Conditioner.

Air cooled condensing units from ¼ to 5 hp, inclusive, and water cooled units from ⅓ to 50 hp, inclusive All models available for either Freon (F-12) or Methyl Chloride Mechanical advantages include Timken Bearings, Centro-Ring Positive Pressure lubrication

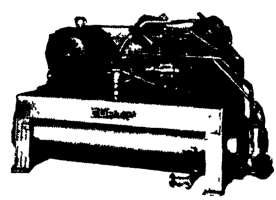
Special models are available for ice cream, frozen food cabinets and for the dairy industry

Saturated Air Condenser

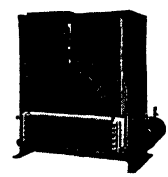
For condensing refrigerant vapors economically and efficiently Saves approximately 95 per cent water cost Used for air conditioning or commercial refrigeration installations up to 5 ton Capacity.

Air Conditioning

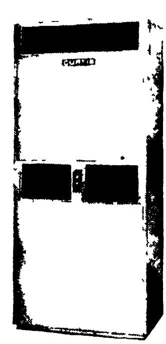
For today's essential Air Conditioning requirements Curtis offers complete packaged, refrigerated air conditioning units, requiring only water and electrical connections to install. Cools, dehumidifies, circulates and filters the air. Eliminates costly installation expense. Adaptable for heating.



15 hp Cleanable Shell and Tube Condensing Unit Other sizes from 3 to 30 hp



Saturated Air Condenser



3 and 5 ton Packaged Type Air Conditioner

ALBANY
ATLANTA
BALTIMORE
BOSTON
BUFFALO
CHARLOTTE
CHICAGO
CINCINNATI
DALLAS
DETROIT
KANSAS CITY

Frick Company

(Incorporated)

Air Conditioning, Refrigerating and Ice-Making Equipment

Waynesboro, Penna.

LOS ANGELES
MEMPHIS
NEW ORLEANS
NEW YORK
OKLAHOMA CITY
PALATKA
PHILADELPHIA
PITTSBURGH
ST LOUIS
SEATTLE
WASHINGTON

Distributors in 150 Principal Cities

AIR CONDITIONING

Ask for Frick Bulletins 503, 504, 505 and 520 on Air Conditioning

Complete Frick systems; also refrigeration for use with equipment supplied by others. Over 1000 installations attest the value of Frick air conditioning systems and those using the Auditorium patents. Successful experience with important Government and industrial war jobs enable us to solve your problems.

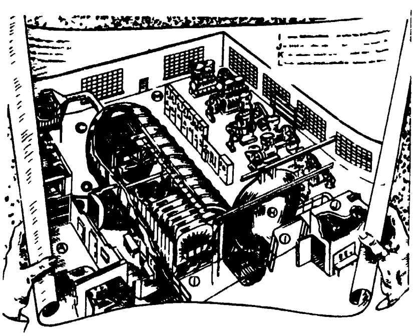
AMMONIA REFRIGERATION

Combined units and vertical enclosed compressors, with two or four cylinders, in sizes from ½-ton up. Widely used for air conditioning, with material savings. Ask for Bul. 503 on this subject.

Pratt and Whitney use Frick Ammonia Refrigeration for World's Most Accurate Large Air Conditioning System

FREON-12 REFRIGERATION

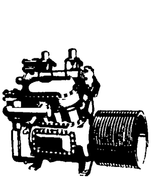
Frick "Eclipse" and larger F-12 compressors form the most complete and efficient line built. Coils, coolers, condensers and controls to suit. Patented Flexo-Seal at shaft, pressure lubrication from submerged pump, capacity controls, and other superior features make Frick machines your logical choice.



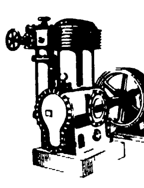
Frick All-Weather Laboratory having 3 Test Chambers and 10 Refrigerating Machines, all under Automatic Control, Built for the U. S. Army. Full Description on Request

LOW-PRESSURE REFRIGERATION

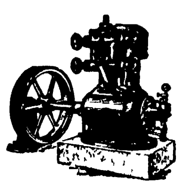
Commercial and industrial units in sizes from ¼ hp up. Charged with either Freon-12 or methyl chloride. Air and water cooled condensers. Coils, coolers, and air conditioners. Get in touch with your Frick Distributor; ask for Bul. 97. Our service includes estimates, layouts, manufacture, installation, maintenance.



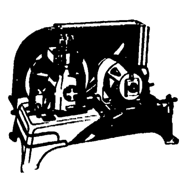
"Eclipse" Freon-12 Compressor. Bulletin 100.



Enclosed Freon-12 Machine. Bulletin 508



Enclosed Ammonia Compressor Bulletin 112



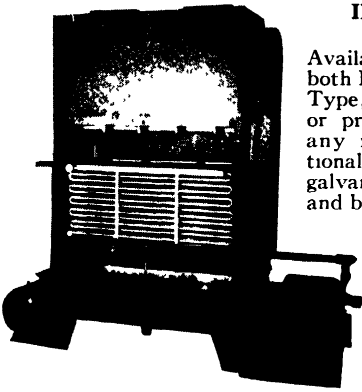
Low Pressure Refrigerating Unit. Bulletin 97

Marlo Coil Co.

6135 Manchester Ave., **St. Louis 3, Mo.**

Manufacturers of Heat Transfer Equipment

Evaporative Coolers—Industrial Coolers—Unit Heaters—Unit Coolers --Evaporative Condensers—Low Temperature Units—Air Conditioning Units—Heating and Cooling Coils.

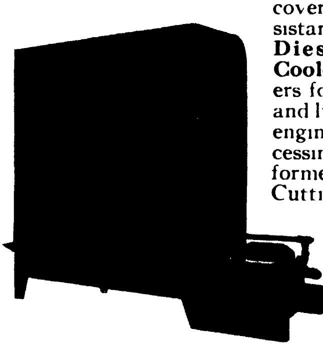


INDUSTRIAL COOLERS

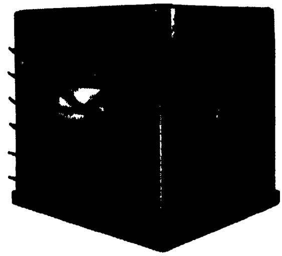
Available in 15 sizes, both Dry Coil and Spray Type, with finned coil or prime surface For any refrigerant. Sectional construction Hot galvanized frames, sump and base

EVAPORATIVE COOLERS

Evaporative Condensers—A complete line of self-contained condensers for any refrigerants having single motor "Unidrive" for fans and pump All prime surface coils. Frame electric welded and galvanized after fabrication All internal surface covered with corrosion resistant mastic

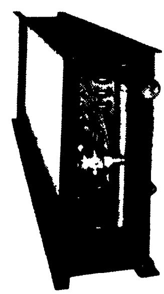


Diesel and Industrial Coolers—Evaporative coolers for cooling jacket water and lubricating oil for Diesel engines and industrial processing; also for Transformers, Quenching Oils, Cutting Coolants, etc.



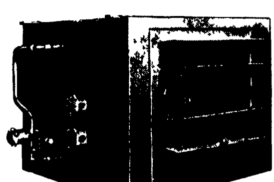
UNIT COOLERS and HEATERS

Sturdily built, for all refrigerants and heating mediums Coolers in "Pull-Through" and "Blow-Through" types.



AIR CONDITIONING COILS—BLAST COILS

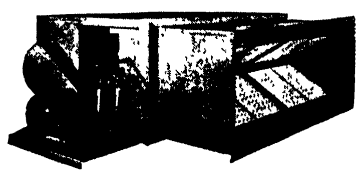
Extended surface—"Ball-Bonded"—coils for all cooling or heating mediums, suitably constructed of any conventional metal—all sizes, duties and capacities



U. S. Patent

LOW TEMPERATURE UNIT

Designed for maintaining sub-zero temperature. Defrosted manually or automatically by electric heater units



AIR CONDITIONING UNITS

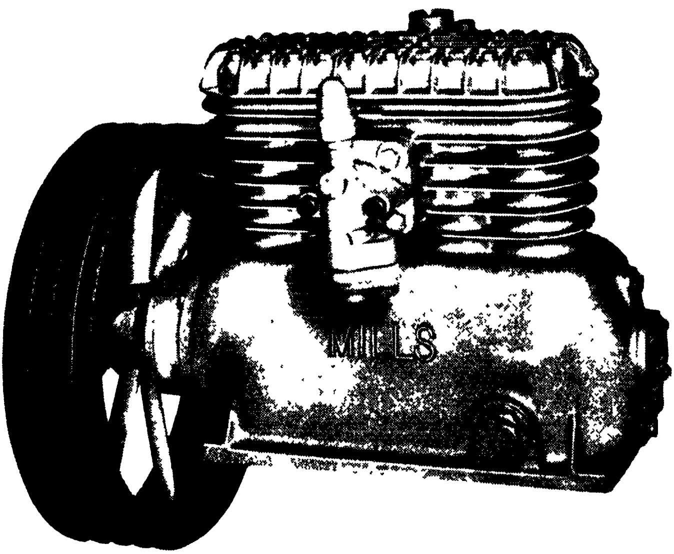
Air Conditioning Units in either ceiling suspended or floor type. Capacities from 900 cu ft to 12,000 cu ft. Sturdily built on welded angle iron frames of sectional design for easy installation.

Mills Industries, Incorporated

Refrigeration Division

4100 Fullerton Avenue, **Chicago 39, Ill.**

MILLS COMPRESSION EQUIPMENT



The Mills line of compression equipment for air conditioning and direct re-refrigeration is complete

The Mills factory representative will furnish technical details These men are available in all parts of the country

COMPRESSORS

COMPRESSORS

RECIPROCATING (1-4 CYLINDER)

CONDENSING UNITS
(Conventional Type Belt Drive)

AIR-COOLED ($\frac{1}{4}$-3 H P)
- Electric Powered
- Gasoline Powered

WATER-COOLED ($\frac{1}{3}$-10 H P)
- Electric Powered
- Gasoline Powered

CONDENSING UNITS
(Motor Speed Direct-Drive)

AIR-COOLED
WATER-COOLED

CONDENSING UNITS
(Mobile Type)

AUTOMOTIVE
AIRCRAFT
RAILWAY

Does the compressor you specify for your vital wartime refrigeration and air conditioning needs have balance and efficiency?

Does the compressor you are thinking about for your clients' postwar refrigeration and air conditioning requirements have balance and efficiency?

The Mills Compressor has both!

Specifying Mills Condensing Units Will Protect Your Reputation and Prestige as a Refrigeration and Air Conditioning Consultant

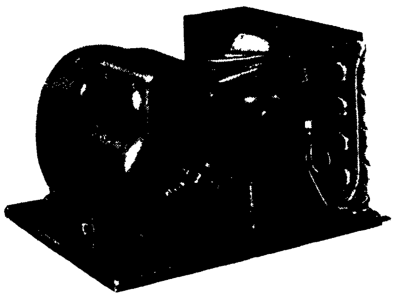
Servel, Inc.

Electric Refrigeration Division

Evansville 20, Indiana, U. S. A.

"POWERED by Servel" is the hall mark of a quality product

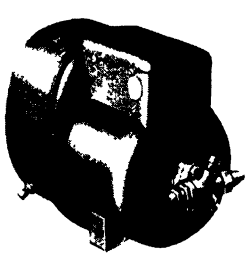
CONDENSING UNIT SPECIALISTS



Servel Supermetic Condensing Unit

The Electric Refrigeration Division of Servel, Inc , has for over 23 years, specialized in producing top quality low pressure condensing units As the first American manufacturer to use oil soluble chlorinated hydro-carbon refrigerants, Servel has pioneered a large part of the most important engineering developments in this field

Servel has maintained its condensing unit production throughout the War Years, adding many important features and developing improved designs and technical processes When production restrictions are removed, Servel will immediately take steps to expand this phase of its business with new tooling and increased personnel This expansion will assure efficient volume production of **Servel Supermetic** units -the most modern line of hermetically sealed condensing units on the market These units, developed before the war, and suspended because of critical material shortages have been refined and improved to provide added performance, efficiency, and dependability.



Steel Cased Supermetic Power Unit Fractional H P

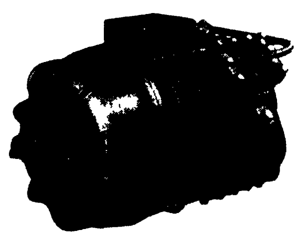
The new line of units will serve the requirements of more than 90 per cent of the refrigeration field from sub-zero **Freezers** up to air conditioning and comfort cooling In addition to these **Supermetic** units, Servel will offer a companion line of belt driven units and compressors for special applications, D C , odd frequency, Gas Engine drive, etc.

FOR DISTRIBUTORS AND CONTRACTORS- Servel sells its condensing units throughout the World through regularly franchised dealers and distributors serving a specific trading area. Full selling, Engineering, and service assistance is extended these customers by Servel's field force and factory staff Servel's National advertising assures acceptance everywhere.

FOR MANUFACTURERS—We especially invite inquiries at this time from manufacturers who contemplate the production of window units, room coolers, store coolers, etc , for post-war sales.

Our Engineering Department will be glad to cooperate in a selection of components and extend testing facilities for completed units Samples of new products will be available shortly after war restrictions are relaxed Our field sales representatives and engineers will be glad to make appointments for consultations on these matters upon request.

Address Servel, Inc , Electric Refrigeration Division, Evansville 20, Indiana.



Cast Case "Field Serviceable" Supermetic Power Unit - 1 H P and above

The Vilter Manufacturing Co.

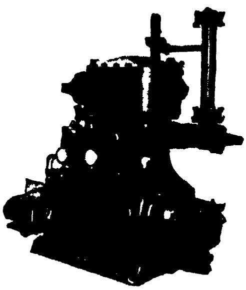
Since 1867

Milwaukee 7, Wisconsin

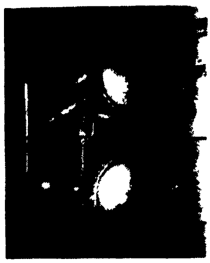
AIR CONDITIONING EQUIPMENT FOR INDUSTRIAL OR COMFORT COOLING

Offices and Distributors in Principal Cities

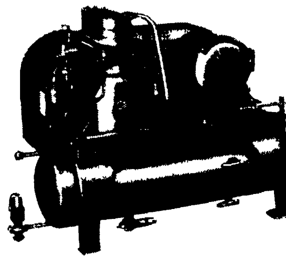
Ammonia and Freon Compressors



Shell and Tube Equipment



Ammonia, Freon, Methyl Chloride Self-Contained Condensing Units



Ammonia Compressors—The result of over seventy years of research, development and experience gained through thousands of installations of all types, in all industries. Famous for high tonnage capacity at low HP and low operating costs. Built in a wide range of capacities from 2 to 100 tons standard A S R E. rating in Vertical Types; up to 750 tons in Horizontal Types

Freon Compressors—Embody many outstanding features that prevent leakage and minimize friction—resulting in extremely low relative HP per ton. Made in capacities up to 150 tons Capacitrols are available, providing flexibility of operation.

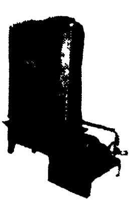
Self-Contained Condensing Units—Vilter self-contained condensing units for Ammonia, Freon or methyl chloride are made in sizes from ¼ to 30 HP Dependable, economical operation

Unit Air Coolers—Available in a wide range of sizes and types for any air conditioning requirement—product coolers, dry coil coolers, spray type coolers, low temperature electric defrosting coolers, and floor or ceiling central system air conditioners.

Water Coolers and Condensers—A complete line of shell and tube and double pipe water coolers, brine coolers and condensers for Ammonia or Freon.

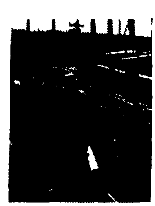
Air Conditioning—All sizes and types of air conditioning coils, evaporative condensers and air washers—and special units for central station comfort cooling systems.

Valves, Fittings and Piping—Ammonia and Freon valves and fittings. Prefabricated piping in all sizes for refrigeration and special process work.



Unit Air Conditioners Spray and Dry Coil Types

Catalog—Valves, Fittings, Accessories Write for Free Copy



Prefabricated Process and Refrigerant Piping

Worthington Pump and Machinery Corporation

Air Conditioning and Refrigeration Division

General Offices: HARRISON, NEW JERSEY

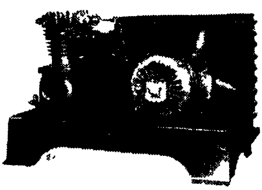
Albany	Buffalo	Detroit	Los Angeles	Providence	Springfield, Mass.
Atlanta	Chicago	El Paso	New Orleans	St. Louis	Syracuse
Baltimore	Cincinnati	Fort Worth	New York	St. Paul	Tulsa
Birmingham	Cleveland	Galveston	Philadelphia	Salt Lake City	Washington, D C
Boston	Dallas	Houston	Pittsburgh	San Francisco	Wilmington, Del.
	Denver	Kansas City	Portland, Ore.	Seattle	

Representatives in all Principal Cities CA3-1

REFRIGERATION SYSTEMS FOR AIR CONDITIONING

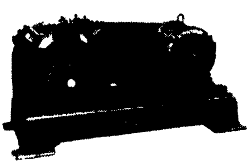
Complete refrigerating systems for use with Freon-11, Freon-12, Methyl Chloride, Ammonia, or Carbon Dioxide, either direct-expansion or water cooling applications. A complete line of refrigeration compressors, permitting impartial recommendations. A nation-wide organization of Distributors in major cities to provide sales and engineering service and plan complete air conditioning systems of the central or unit type. Architects, Engineers, and Contractors are invited to consult with us. Write to Harrison, N. J., or any branch office, for bulletins on these products

Small Self-contained Units



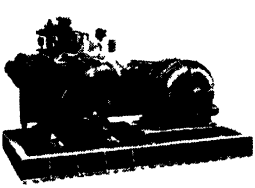
Freon-12 or methyl chloride condensing units; motors ¼ to 2 hp. with air or water-cooled condensers Used in small air conditioning systems, and in commercial refrigeration. Capacities up to 2 tons.

Medium Self-contained Units



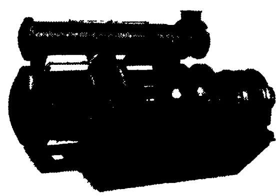
Freon-12 or methyl chloride compressor units for use with "shower" condensers or water-cooled condensers Features: FEATHER (Pat'd.) Valves; automatic capacity control. Capacities 3 to 30 tons.

Large Self-contained Units



Freon-12 or methyl chloride compressor units for use with "shower" condensers or water-cooled condensers. Features: Worthington FEATHER (Pat'd.) Valves; automatic capacity control. Capacities up to 100 tons.

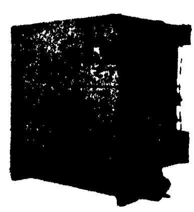
Centrifugal Refrigeration Water Cooling Systems



Freon-11 centrifugal compressor, water cooler and water-cooled condenser in compact unit assembly. Electric motor or steam turbine drive 56 unit sizes . . 150 to 1200 tons

Evaporative Type Jacket-Water Cooler

(With By-Pass Section For Automatic Temperature Control.)



Cooling jacket water for diesel and gas engines, air compressors, etc. Ideal for the cooling of quenching oil for tempering steel products. Also for cooling transformer oil to reduce core loss.

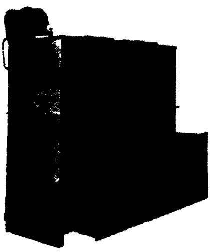
Miscellaneous

High and low side equipment for every purpose.

Worthington Pump and Machinery Corporation

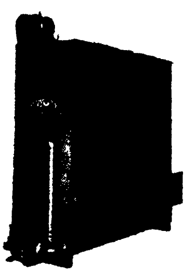
Air Conditioning Units

For Direct Expansion Freon-12 or Chilled Water Circulation



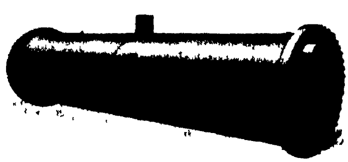
Vertical and horizontal; 500 to 12,000 cfm; large air passages; slow speed, quiet rugged fans; separable sections; readily accessible. The design permits flexibility in installation arrangements.

Shower Condensers



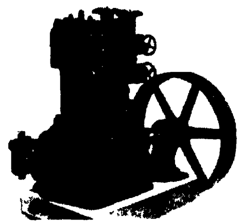
A combined condenser, receiver, and modified cooling tower, in one assembly, for Freon-12 or methyl chloride systems; 2 to 130 tons refrigeration; built in separable sections, all parts easily accessible. Saves 90 to 95 per cent in cost of water.

Horizontal Condensers



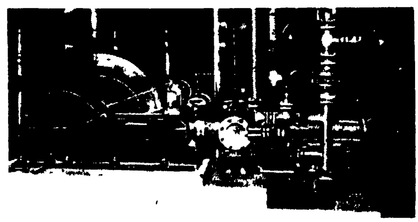
Atmospheric drip type, for warm corrosive waters Double-pipe for closed systems, can be retubed without shutting down. Multi-pass, as illustrated above, for closed systems and space saving.

Vertical Ammonia Compressors



Pressure-lubricated; roller main bearings; safety heads; patented Feather Valves; belt drive, or direct-connected to electric motor, diesel or gas engine; ratings from 2 to 160 tons in one unit.

Horizontal Ammonia Compressors

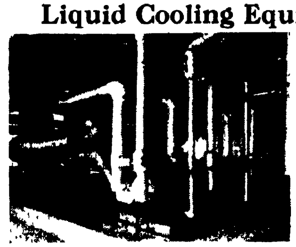


Single and duplex; single-stage and two-stage; belt drive, or direct-connected to electric motor, diesel, gas or steam engine; patented Feather Valves; ratings from 60 to 750 tons. Automatic capacity control features are easily applied. Space requirements vary depending upon type and drive.

Carbon Dioxide Compressors

A series of convenient types and sizes for every requirement is available.

Liquid Cooling Equipment



Various designs of horizontal single and multi-pass types, for a wide range of services; also vertical types. Chillers for oil dewaxing. Single and double-pipe for milk, wort, chemicals, etc. Cold liquid circulating systems.

Air Controls Inc

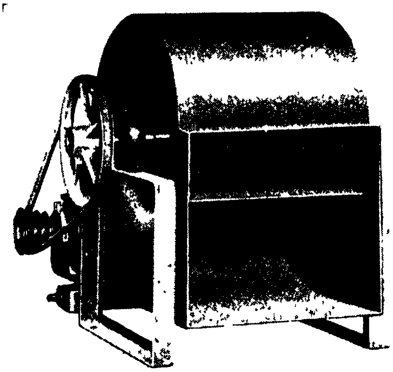
Div. of

The Cleveland Heater Co.

2311 Superior Ave., **Cleveland 14, Ohio**

PIONEER BLOWER MANUFACTURERS

REX BLOWERS



Rex Blower

A complete line of centrifugal blowers well suited for use in the heating, cooling and ventilating field. There are two series of Rex Blowers—the A. C. and H. P. models, ranging in capacity from 400 to 12,000 cfm. Twin units double these volumes. The H. P. series is more compact for use where space requirements are a factor.

WHEELS: Double Inlet, carefully balanced, assuring true running and high efficiency. Forward curved multi-blade type for maximum air delivery at lowest speeds. Each blade is die formed and securely riveted to inlet rings to form a rigid assembly.

HOUSINGS: Made of heavy gauge steel, die formed and electrically welded for maximum air intake. Wide outlet distributes the air evenly.

BEARINGS: Self-aligning to insure smooth operation, mounted on sound and vibration absorbing rubber bumpers. Phosphor-bronze bushings are self-lubricating to supply right amount of oil.

OTHER FEATURES: Motor pulleys on smaller sizes have four steps to provide four speeds. Variable speed supplied on 16 in. size and up. V-Pulleys universally used.

Shafts are accurately ground and polished to 0.0005 in., of heavy weight to prevent whip which causes noise and wear.

Angles of discharge can be varied and construction changed to meet special requirements. REX Blowers are rated in accordance with the test code of the A.S.H.V.E. Performance is quiet, dependable and trouble-free.

For complete information, write for *Catalog No. 222* which contains a complete description of REX Blowers and tables of dimensions, capacities and other data.

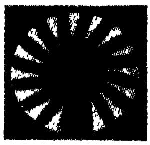
The Allen Corporation

9759 Erwin Avenue, **Detroit 13, Michigan**

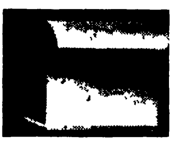
ENGINEERED VENTILATION FOR INDUSTRY

HEAT • MOISTURE FUME AND DUST REMOVAL • • •

Allen "LO-NOIZ-LEVEL" Multi-Blade Fans:



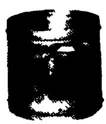
Are precisely balanced units designed for application where minimum noise is permissible. Unit pressures on fan blade areas are low and close blade spacing prevents re-entry of exhausted air even where static pressures are appreciable Construction is heavy and durable. Belt drive keeps motor size down and motor speed up. Proper pulley size enables fan speeds to be kept "right on the button" for maintaining rated capacities



Allen Type "H" Roof Fans:

Offer a simple, compact, light-proof design. Low height means minimum wind resistance. Effluent air blast is deflected to automatically open counterbalanced louvers at either end. Closed louvers when power is off prevent heat loss Small motors exhaust relatively large air volume. Sizes: 30 to 60 inches. Capacities· 5600 to 45,000 c f.m.

Allen "STAXAUST" Fan Sections:

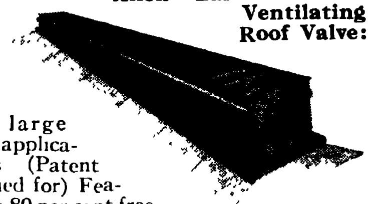


Can be used for conversion of all types of roof ventilators into powerful ventilator exhaust or air supply units Sections are simple, compact, sturdy, efficient Centered fan and motor assembly mounted on inner steel reinforcing band. Wholly enclosed, ball bearing, fan duty motors directly connected to propellors Unit easy of access for inspection and lubrication

Other Allen Fans:

Include Exhaust fans for industrial, commercial and residential use; Remote drive fan sections for corrosive and/or high temperature applications; "Electro-Wind" turbine type ventilators; Winter Supply Units designed to supply make-up air in winter to buildings whose exhaust capacity produces negative static.

Allen "Laminar-Flow" Ventilating Roof Valve:



For large area applications (Patent applied for) Features 80 per cent free area within the limits of the case—a doubling due to Allen's "exclusive" arrangement of effluent air in laminations rather than in a group of bifurcated streams which waste space. Units can be made up to 30 ft wide and of any length. Maximum unit height is 26 inches.

Allen "ELECTRO-WIND" Dual Range Ventilator:

Is a turbine type ventilator utilizing wind actuation and incorporating motor drive for use when no wind is present Exhausts radially around entire circumference of rotor Aided by wind currents from whatever direction. Driving element top-mounted to leave throat free Durable, efficient, economical.

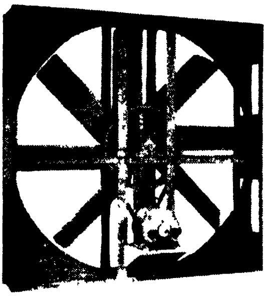
Other Allen Ventilators and Allen Engineering Service:

Standard Stationary . . . "Stream-Flo" . . Type "H" Roof Ventilator . . . "Coni-Vane" Turbine Type . . Type "C" Turbine Ventilator . . . Exhaust Fan Sections . . . Shutters and Bases.

Allen is an engineering organization specializing in ventilation for industry. Allen fans and ventilators have been designed to meet problems of heat, dust, fumes and moisture removal. Inasmuch as Allen is always interested in the most economical solution of any ventilation problem, Allen men are not committed to fan systems but will as readily recommend gravity or combination gravity and fan as fan alone. Ask for an Allen engineering recommendation.

American Coolair Corporation

3606 Mayflower Street, Jacksonville 3, Florida
Manufacturer of COOLAIR Ventilating and Exhaust Fans
A Pioneer Manufacturer of V-Belt Drive Exhaust Fans
Charter Member: Propeller Fan Manufacturers Association



TYPE S—6 TO 9 FEET
Especially designed for ventilating and cooling in industrial plants and shops, power stations, warehouses and other large buildings—the Type S fan has a heavily braced double frame, special pillow-block ball bearings on each side of fan wheel, 8 to 12 reinforced fan blades and up to 10 heavy duty V-belts depending on size of fan and motor. This fan is usually installed in roof bay or gable, penthouse or outside wall

COOLING BY AIR MOVEMENT

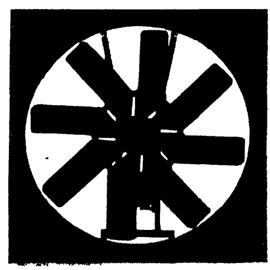
Construction engineers and architects as well as production superintendents know that proper ventilation is necessary for top efficiency in employees to insure maximum production. Not so well known is the fact that when heat and humidity induce high body temperatures and perspiration among workers—it takes from four to eight times the volume of moving air to correct this condition than it does for simple ventilation. Satisfactory cooling requires a complete change of air at least once a minute.

The American Coolair Corporation pioneered in the manufacture of Exhaust Fans for ventilating and cooling. During the past 17 years, Coolair engineers have been directly responsible for many of the developments in this growing field.

In planning a Coolair installation determine the cubic content of the space to be cooled or ventilated and select a fan of ample capacity. Tables of performance data and fan sizes are shown on the facing page—and Coolair's NEW CATALOG, sent FREE on request, contains recommended air changes and suggestions for industrial and commercial installations.

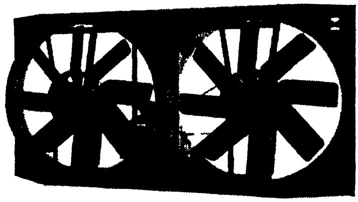
QUALITY FEATURES OF COOLAIR EXHAUST FANS

1. Built-In Springs on Smaller Motored Units—dampen vibration and practically eliminate noise for quiet installations by insulating moving parts from frame. **2. Light, Compact Fabricated Steel Frame**—fits into many openings where bulky metal housings cannot be used. **3. Reversible**—when equipped with reversible motor, fan will blow in or exhaust as desired. **4. Ball Bearings in Fan Hub**—eliminate sleeve bearing chatter and end thrust knock—permit operation in any position (specify ball bearing motor for vertical or angle discharge). Use grease, instead of oil, requiring attention not more than once a year. **5. Eight Large, Slow-Moving Steel Blades**—instead of 4 or less on cheaper fans—up to 12 blades on Type S models. Low tip speeds for quiet operation and steady flow of air. **6. V-Belt Drive**—for high efficiency using small motor for economical operating speed. **7. Long Hour Service Motor**—Nationally known makes of motors. **8. Certified Air Ratings**—in accordance with Standard Test Code of A.S.H.V.E. and N.A.F.M. **9. Full Streamlined Orifice**—(on Type O & OT) avoids "spill-off" at end of blades, reduces power consumption.



TYPE O—26 TO 62 INCHES
Patented built-in springs (on smaller motored sizes) and streamlined orifice combine to make this type quiet in operation. Designed for reversible operation it is equally efficient for vertical or angular discharge when equipped with ball bearing motor. Usually installed in full or section of window opening, outside wall, partition, skylight opening, penthouse—often used in a battery of units for most efficient cooling of large buildings. Covered by U. S. Patents 1992112 and 2191418

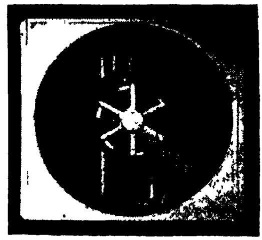
TYPE OT—TWIN UNIT
This unique Coolair Twin-Unit is two fans of Type O specifications mounted side by side in one frame and operated by a single motor. Widely used where limited headroom or vertical wall space will not permit the use of a single fan large enough for the job Especially adapted for installation in partitions, outside wall and on end can be fitted into existing window and door openings Covered by U. S. Patents 1992112, 2108738 and 2191418



Dimensions in Inches

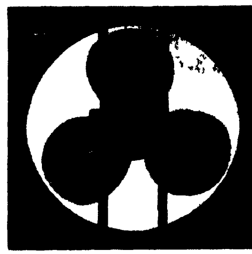
Fan Size	Overall Height	Overall Width	Overall Depth (Approx)
28-W	31	33	15
2-O	30⅝	30⅝	18
2½-O	36⅝	36⅝	18
3-O	42⅝	42⅝	18
3½-O	49	49	19
4-O	55⅛	55⅛	19
4½-O	61⅛	61⅛	19
5-O	67⅝	67⅝	19
6-S	75¼	75¼	28
7-S	86⅞	86⅞	34
8-S	99	99	38
9-S	111⅛	111⅛	38
2-OT	30⅝	61¼	18
2½-OT	36⅝	73¼	18
3-OT	42⅝	85¼	19
3½-OT	49	98	19
4-OT	55⅛	110¼	22

28-W FAN WITH SAFETY GUARD



Coolair's lowest priced belt-drive fan is equipped with built-in springs, adjustable diameter motor pulley and safety guard. This fan can be easily and quickly installed in upper or lower half of any standard window of work rooms or offices where proper ventilation is necessary for health, comfort and efficiency of workers. See tables for data.

DIRECT DRIVE FANS



Four sizes 16 to 24 inches in diameter. General purpose exhaust fan for most commercial and industrial uses. Data on sizes, performance and dimension furnished on request.

COOLAIR AUTOMATIC CEILING & WALL SHUTTERS

Precision-built all-steel shutters that open and close automatically when fan is turned on or off. Ceiling shutters eliminate need of ceiling grille and trap door. Wall shutters give weather protection for fans discharging directly to open air.

Performance Data—Coolair V-Belt Drive Fans

Fan Size		Horse Power	Fan R. P. M.	Cubic Ft. Air Per Min
28-W	L	¼	580	6000
2-O	L	¼	570	6200
	D	⅓	630	6800
2½-O	B	¼	411	8000
	C	⅓	454	8800
	D	½	522	10100
	D	¾	600	11700
3-O	B	¼	312	10000
	C	⅓	345	11000
	C	½	398	12700
	D	¾	450	14400
	D	1	500	16000
3½-O	B	⅓	261	13000
	C	½	300	15000
	C	¾	345	17000
	D	1	380	19000
	D	1½	440	22000
4-O	L	½	258	19000
	C	¾	317	22000
	D	1	353	25000
	D	1½	405	28000
4½-O	B	½	224	22000
	C	¾	255	25000
	C	1	276	27000
	D	1½	319	32000
	D	2	355	35000
5-O	B	½	200	27000
	C	¾	225	30000
	C	1	245	33000
	C	1½	282	38000
	D	2	310	42000
	D	3	355	48000

Fan Size		Horse Power	Fan R P. M	Cubic Ft. Air Per Min.
6-S	B	1	155	35000
	C	1½	177	40000
	C	2	195	45000
	D	3	224	50000
	D	5	270	60000
7-S	B	2	170	58000
	C	3	195	67000
	D	5	230	76000
	D	7½	270	85000
8-S	B	3	170	85000
	C	5	200	100000
	D	7½	230	112000
	D	10	259	124000
9-S	B	5	150	110000
	C	7½	170	125000
	D	10	190	138000
	D	15	215	154000
2-OT Twin	B	¼	455	9800
	C	⅓	500	10800
	C	½	570	12400
2½-OT Twin	B	⅓	359	14000
	C	½	411	16000
	C	¾	470	18200
3-OT Twin	B	½	312	20000
	C	¾	360	23000
3½-OT Twin	B	¾	272	27800
	C	1	300	30700
4-OT Twin	B	1	258	38000
	C	1½	317	44000
	D	2	345	48000

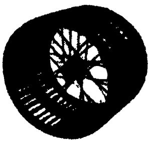
B—Very Quiet (Homes, Theatres, Hospitals, etc.).
C—Quiet (Stores, Offices, Restaurants, Barber Shops, etc.).
D—Industrial (Laundries, Factories, Canneries, Bakeries, Pressing Clubs, Garages, etc.).
L—Has adjustable diameter motor pulley for Very Quiet and Quiet performance

Bayley Blower Company

1817 S. Sixty-Sixth Street **Branches in Principal Cities** **Milwaukee 14, Wis.**

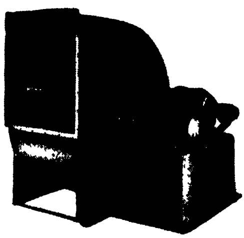
Builders of Heating, Ventilating, Cooling, Purifying, Humidifying and Air Washing Equipment; Exhaust and Drying Apparatus, Mechanical Draft and Blast, Fans and Blowers of all Types

TYPE "F" PLEXIFORM FANS AND TYPE "AP" AEROPLEX FANS



Plexiform and Aeroplex Fans are both designed for heating, ventilating and air conditioning service. The **Type "F"** fan is using a multiblade wheel (see cut at left) while the type **"AP"** fan is built around a wheel of high speed design (see cut at right.)

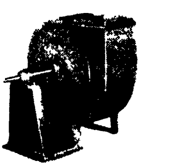
The wheels determine the character of the performance or the fan "Characteristics," thus the Type "F" is a slow speed fan having a rising power curve, while the **"AP"** is a high speed design and has self-limiting power characteristics Both fans are highly efficient and can be built single inlet, single width, or



double inlet, double width in ten different arrangements of drive and eight directions of discharge The standard fans range in capacity from 1200 to 300,000 Cfm and these fans are readily available in Class I or II designs for various types of duty Class III and IV can be furnished on special application

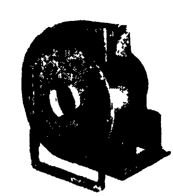
Type "EX" Fans

For all exhaust problems, pneumatic conveying, fume exhaust etc. **"EX"** is highly adaptable, feasible and serviceable fan It is sturdy in construction, easily installed and maintained The design is reversible and standard sizes Nos 15 to 80 inclusive are available



Type "H" Fans

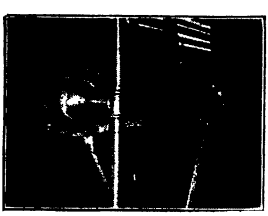
For pressure applications from 6 in. to 42 in for belt drive or direct connection to motor, **Type "H"** is a most practical design Standard sizes Nos 25 to 80 are listed but the design is suitable for modification of wheel diameters to meet various motor speeds



Both **"EX"** and **"H"** fans can be designed to fit specific applications The design permits easy modification of details and the fans can be used for induced, forced draft, cupola service, primary or secondary air supply to oil burners, etc Special details, outlet and inlet, or mounting can be designed to suit your special assembly

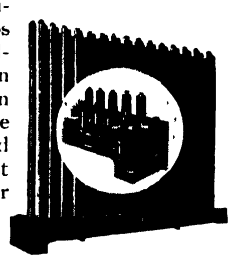
Turbo Spray

The Turbo Spray for industrial and comfort conditioning, cooling and humidifying, is a practical design as the atomization is by mechanical means, thus there can be no clogging or interruption of service even in atmosphere heavily laden with fibers or other foreign material Single or multiple bank washer, one or two stage designs can be furnished from 2500 to 100,000 Cfm capacity. To fit space requirements width and height can be modified to suit



Chinook & Chinookfin

Both **Chinook and Chinookfin Coils** are based on the tube-within-a-tube design, originated by Bayley The principle involved permits single header with all tube ends free expanding, thus eliminating the likelihood of freeze ups and cracking usually due to expansion and contraction strains. Both type coils are designed for usual hot blast heating and cover a wide range of capacities and applications.

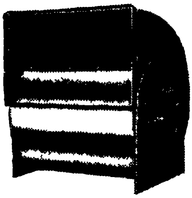


The Bishop & Babcock Mfg. Co.

4901 Hamilton Ave. **Cleveland 14, Ohio**

Branch Offices in Principal Cities

MASSACHUSETTS HEATING AND VENTILATING PRODUCTS

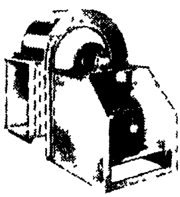


BLOWERS

Squirrel Cage Type—Slow speed suitable for all air conditioning and industrial applications.

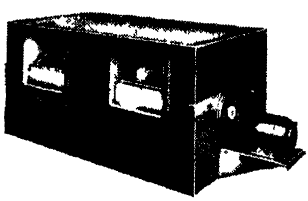
Backward Curve Blade Type—Power fixed self-limiting horsepower characteristic

Both squirrel cage and backward curve blade blowers available in complete range of sizes and capacities, single width and double width



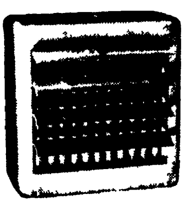
BELT DRIVEN VENTILATING SETS

Complete packaged units shipped assembled with motor and drive ready for operation Housings are convertible type permitting change in discharge after shipment from factory Quiet, compact, available for capacities 200 to 10,000 cfm

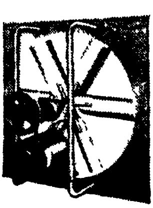


UNIT HEATERS

Blower Type—10 sizes for ceiling and floor mounting All units V-belt drive with motor Can also be used as ventilating and air conditioning units by addition of filter and damper section



Propeller Type—10 sizes in horizontal type with capacity range of 28,000 to 278,000 Btu 9 sizes vertical projection type, capacity range 144,000 to 600,000 Btu, heating elements copper construction



PROPELLER FANS

Direct drive type sizes 12 in. to 36 in. belt drive type 24 in. to 48 in Complete line of accessory items consisting of shutters and fan houses for all sizes.

Buffalo Forge Company

450 Broadway, Buffalo, N. Y.

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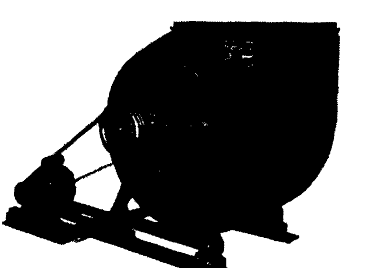
TUCSON, ARIZ—Tidmarsh Engrg Co. 23 North Main St
WASHINGTON 5, D. C—Buffalo Forge Co, G S Frankel, 512 Woodward Bldg.
WILKES-BARRE, PA—Power Engineering Co, 517 Brooks Bldg

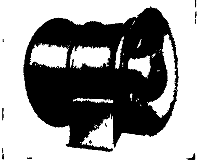
PRODUCTS: Heating and Ventilating Equipment including: Unit Heaters, Multiblade Fans, Pipe Coil Heaters, Buffalo Air Washers, Buffalo Unit Air Washers, Buffalo Unit Coolers, Drying Equipment, Mechanical Draft Fans, Air Preheaters, Exhaust Fans, Blowers, Dust Collectors, Disc Fans, Spray Nozzles.

Buffalo Limit-Load Fans



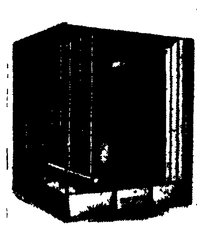
Buffalo Limit-Load Fans for ventilating embody several improvements to deliver stepped-up efficiency under practical conditions. Durably built for years of service. Dynamically balanced. Quiet, economical to operate. Non-overloading characteristic prevents motor from overloading and burning out, regardless of fan load.

Axial Flow Fans



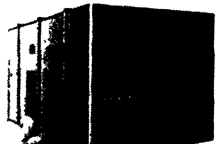
This non-overloading high pressure fan—with directional guide vanes—propels the air stream in a true axial direction. Energy losses are reduced to a minimum with a marked power saving.

Buffalo Air Washers



Type "A" Washer

Buffalo Air Washers and Humidifiers are built for the most efficient washing, purifying and tempering of air under varying atmospheric conditions. Designed and constructed for low cost installation and maintenance Careful attention to structural details insures long service-ability with a minimum of attention. Several types and models to meet specific plant requirements.



Wet Filter Washer

Fans for Every Ventilating Need

Buffalo Fans represent over 60 years of specialization in the design and construction of fans for practically every ventilating and air-handling application from small kitchen fans to rugged fans for boiler draft. Complete information on request.

Buffalo Unit Heaters

"Highboy" and "Lowboy" and "Breezofin" Units for floor, wall or ceiling installation. Sizes for all buildings of all types.

Champion Blower & Forge Co.

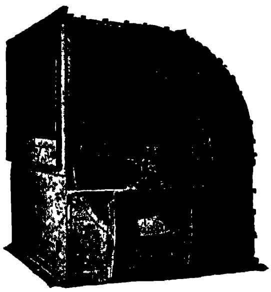
Manufacturers and Engineers

Plant and Offices: Lancaster, Pa.

Address Correspondence to Div. 9

Manufacturers of Blowers, Ventilating Fans and Exhaust Fans for Air and Material; and Blast Gates

Representatives in Principal Cities



Type S Forward curve ventilating fans, single and double width, as well as direct motor drive

Ventilating Fans—For air conditioning systems and mechanical draft. Manufactured in the forward curved type for slow speed, and extremely quiet operation; also in the backward curved type with its flat horsepower curve characteristics and higher speeds suitable in the smaller sizes for direct connecting to synchronous speed motors. Ventilating Blowers manufactured in sizes up to 60 in. diameter wheel, in single and double width. Belt driven blowers equipped with either ball or high-grade babbit bearing. Direct motor drive can be equipped with any type or characteristic motor desired, in any arrangement.

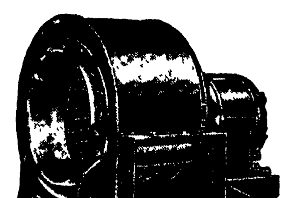
Disc Fans—Super Ventilating Fans made in direct connected type up to 36 in. diameter, totally enclosed, ball thrust type motors. Slow speed motor belt driven type manufactured in sizes up to 48 in. for attic exhaust work and wherever large volumes of air are to be moved against low static pressures. All disc fans are quiet in operation. Decibel ratings on all fans are available.

Super Ventilating fans, direct motor drive up to 36 in diameter. Motor belt drive up to 48 in size.

Forced Draft Fans—All sizes for use on the smallest to largest boilers. Fans can be furnished with inlet or outlet adjustable louvers for controlling air volume.

Blast Wheels—We are well equipped to manufacture single and double width blast wheels in forward or backward curve type for oil burner and stoker manufacturers, as well as manufacturers of air conditioning units and other ventilating equipment.

Vibration Dampener Sub-Bases—For blower and ventilating equipment. Made with heavy channel iron and rubber vibration eliminator pads to suit size and weight of fan or blower.



Type BC Backward curve ventilating and exhaust fans, single and double width; belt driven and direct connected electric.

Special Fan Equipment—We are in position to engineer and build fans, blowers, or exhausting equipment to meet customers' special needs. A card addressed to Div. 9 will bring you complete catalog data or information on any particular problem confronting you.

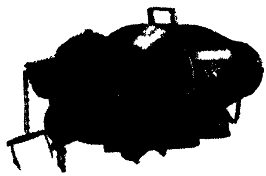
Chelsea Products, Inc.

1206 Grove St., **Irvington 11, N. J.**

FANS

Trade Mark Registered

Charter Members Propeller Fan Manufacturers Association



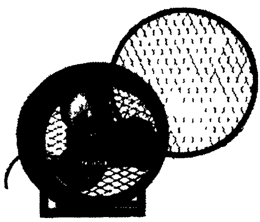
Type OPJ

Type CP

FEATURES

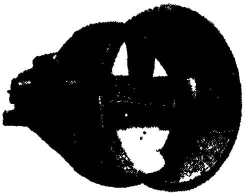
(On Comfort Coolers)

1. **New Streamline Venuri Orifice**—greater volume
2. **Certified Ratings**—as per the A S H V E method
3. **Efficient Fan Duty Motors**—low current consumption
4. **Die Stamped Blades**—precision balance and Alignment.
5. **Moving Parts Rubber Cushioned**—quietness.
6. **Ball Bearing on Fan Shaft**—long life and little attention.
7. **Faultless All Steel Construction**—stability.

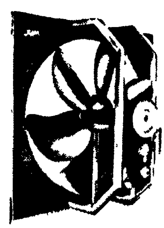


Type DUB

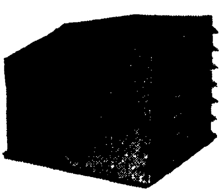
Type 11



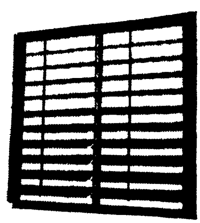
Type DXB



Type IND



Type PH



Type LWL

Chelsea Fans are Known All Over the World

Bulletins and Engineering Data furnished on request Personnel includes many skilled operators—employees who have been identified with the air conditioning industry for over 30 years.

DeBothezat Fans Division

American Machine and Metals, Inc.

Main Office and Factory—East Moline, Illinois

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PRESSURE FANS

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NON-OVERLOADING POWER CHARACTERISTICS CERTIFIED RATINGS—GUARANTEED PERFORMANCE

Ventilating Fan Axial Flow

Axial Flow Ventilating Sets

A complete series of volume and pressure axial-flow fans of high mechanical and static efficiencies with a non-overloading power characteristic. These fans offer savings in space, weight and power. Axial-Flow Ventilating Sets are available in a wide range of capacities in sizes 8 in. through 10 ft. in diameter, and may be had arranged for direct motor drive or belt drive.

Bifurcators

Designed for handling corrosive or high temperature vapors with direct motor driven fan. Motor is located in chamber open to atmosphere but isolated from gases handled by fan. Installed as integral part of duct system, in any position.

Bifurcator

Multi-Stage Impeller Blowers

Units can be furnished in 2, 4, 6 or 8 stages. Direct motor or belt driven, producing high capacities and static pressures, with non-overloading power characteristics

"Power-Flow" Roof Ventilators

Designed to provide positive ventilation at all times regardless of temperature, humidity and wind velocity Guaranteed performance ratings. Equipped with high-efficiency Axial-Flow Pressure Fan, these "Power-Flow" Roof Ventilators possess the greatest air moving capacity per horse power! Low fan tip speeds permit unusual quietness of operation Work efficiently against resistance of duct systems Have non-overloading power characteristic available in a wide range of sizes, speeds, and for all standard electric current. Hinged top gives easy access to motor, fan and shutter.

"Power-Flow" Roof Ventilator

The above is only a partial list of the ventilating units DeBothezat builds. Our engineers will be glad to give you expert assistance in your ventilating problems—offering you a solution in space, weight and power saving equipment. Catalog on all products sent on request.

Hartzell Propeller Fan Co.

DIV. OF CASTLE HILLS CORP.

Piqua, Ohio

Sales-Engineering Offices in Principal Cities

Manufacturers of Propeller-Type Fans and Blowers For Every Industrial Use

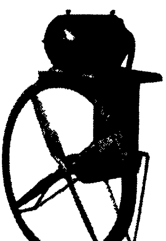
SINGLE PROPELLER FANS

Sizes, 12 to 144 in.; free air deliveries 590 to 270,000 cfm. NEW curved orifice overlaps propeller tips, forming Hartzell Airseal (patented), giving extra air delivery, minimum turbulence. Send for *Bulletin 1101.*

MULTIBLADE FANS

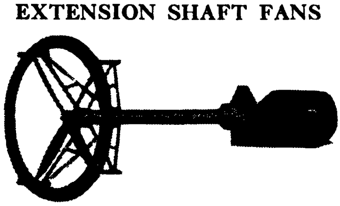
Sizes, 12 to 144 in.; free deliveries 1,200 to 280,000 cfm. (Up to 200,000 cfm against 1.00 in. static pressure) Long life, low upkeep. NEW curved orifice and patented Hartzell Airseal. Send for *Bulletin 1101.*

BELT-DRIVE FANS



Sizes, 12 to 144 in.; air deliveries 590 to 280,000 cfm. Available in one-propeller, two-propeller and multiblade styles. For use in fire-hazardous locations such as paint spray booths Grease-sealed, dust free ball bearings. Send for *Bulletin 1401.*

EXTENSION SHAFT FANS



Sizes, 12 to 55 in.; types, one-propeller, two-propeller and multiblade. Designed for use in ducts through which corrosive elements, excessive heat or abrasive dust passes. Send for *Bulletin 1401.*

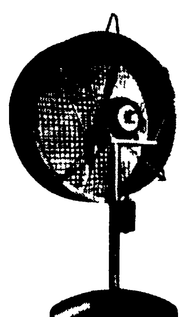
TWO-PROPELLER FANS

Sizes, 16 to 144 in.; free air deliveries 2,400 to 275,000 cfm. (Up to 200,000 cfm against 0.34 in. static pressure) NEW curved orifice and patented Hartzell Airseal mean maximum air. Send for *Bulletin 1101.*

BLOWERS

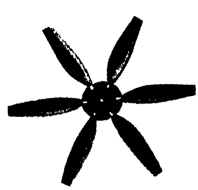
Sizes, 18 to 48 in.; free air deliveries 3,300 to 50,500 cfm. (Up to 32,200 cfm against 2 in. static pressure) Built in both belt drive (shown) and direct drive models Easy installation. Send for *Bulletin 1601.*

COOL BLAST FANS



Models: Portable, Stationary and Heavy Duty Cool Blast Fans (stationary shown), sizes 24 to 55 in, for large-volume air movement; Industrial Stand Model, up to 36 in. size, with substantial but light-weight high stand; Utility, up to 36 in. size, extremely versatile, with simple, close-to-the-ground mounting. Send for *Bulletin 1201.*

COOLING TOWER FANS



Sizes, 3 to 16 ft; four and six blades, adjustable pitch. 10 to 16 ft sizes are special Hartzite plastic, steel hub. Smaller sizes are aluminum alloy. Send for *Bulletin 1501.*

SPECIAL EQUIPMENT

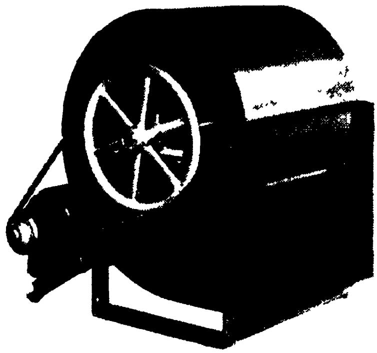
There is a Hartzell fan or blower to meet your needs or we will design and build one for you. Write. All Hartzell air deliveries are in accordance with the Standard Test Code for Centrifugal and Axial Fans.

The Lau Blower Company

2007 Home Avenue, **Dayton 7, Ohio**

Engineers and fabricators of general Air Handling Equipment
Single Inlet and Double Inlet Blowers • Propeller Fans • Accessories

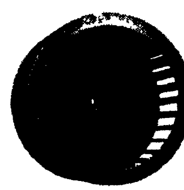
THE WORLD'S LARGEST MANUFACTURER OF FURNACE BLOWERS



Blower Assemblies

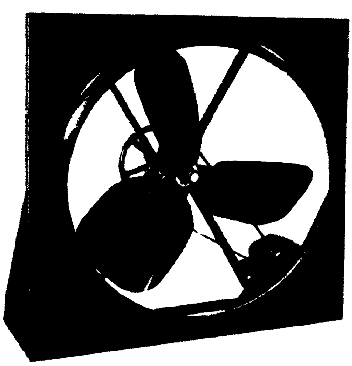
For easy adaptation to modern installation problems wherever it is necessary or desirable to move air—Lau Blower assemblies offer efficient air delivery with low power consumption. Developed through years of experience and in accordance with approved principles of fan design. Double or single inlet. Top, side, or rear motor mountings. Dynamically balanced wheels. Self-aligning bearings. Widest range available. Greatest selectivity of air delivery. Every size tested and rated for performance in accordance with A S H V E and *N A F M* Codes

Blower Wheels



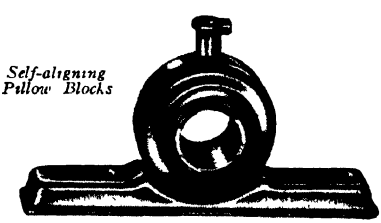
Lau can supply both single and double inlet blower wheels in wide range of diameters and lengths. Also for both clockwise and counter-clockwise rotation Back plate (or center disk) die-formed for greater strength Blades are precision cut and formed for maximum air contact Wheels deliver highest rated performance for space required Dynamically balanced. Operate quietly and economically

Propeller Fans

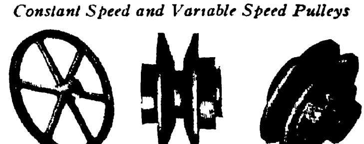


For providing fresh air in confined enclosures; removing stale, excessively hot, or dust-laden air, fumes, smoke, etc Three broad, deep-pitched blades provide maximum volume air with minimum power consumption Venturi-type entrance housing reduces air "drag" and turbulence—eliminates most common cause of "air noise." Five sizes.

Self-aligning Pillow Blocks



Constant Speed and Variable Speed Pulleys



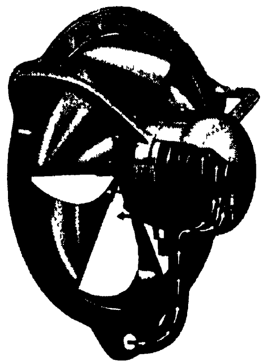
Catalogs, performance data, specifications, and prices available on request on above and other air handling equipment. Inquiries solicited for any application. Our engineers will gladly assist you *Write us regarding your requirements.*

ILG Electric Ventilating Co.

2832 North Crawford Ave., **Chicago, Ill.**

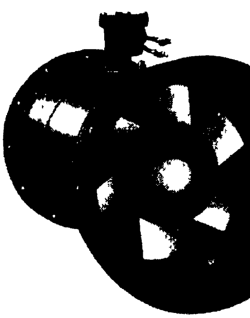
Offices in 38 Principal Cities

Propeller Fans, Axial-Flow Fans, Blowers, Unit Heaters, Unit Coolers



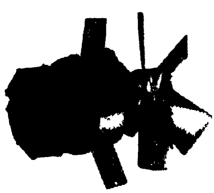
ILG Self-Cooled Motor Propeller Fans

Used for exhaust of stale air, fumes, heat, dust, odors, etc. Self-cooled motor combines protection of enclosed motor with low operating cost of open motor—constantly cooled by fresh, clean air, circulated internally—never "gums-up" from contact with foul air—saves 5 to 10 per cent on power costs. Rugged, heavy-duty framework. Dynamically balanced fan wheel, direct-connected to motor. Smooth, quiet, effortless operation—low cost, long lived. "One-Name-Plate" Guarantee Certified ratings. Sizes, 8 in to 72 in Get Catalog No. 142.



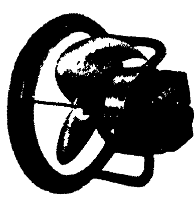
ILG Axial-Flow Fans

Heavy-duty, extremely compact design, as developed by ILG for use aboard ship by the U. S Navy and Maritime Commission Suitable for duct and trunk mounting for vertical and horizontal operation All steel housing flanged at both ends Removable streamlined inlet cone Guide vanes straighten air flow, reduce turbulence. In ILG design, vital working parts are quickly, easily accessible for installation, lubrication, or servicing Proven efficiency and dependability "One-Name-Plate" Guarantee 10 in. to 46 in sizes Get new data sheets



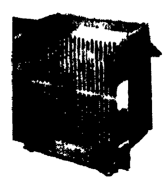
"UN" TYPE FANS

For vertical or horizontal mounting anywhere except in ducts. Several models with various mounting arrangements. Get data sheet.



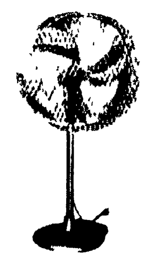
"CQ" TYPE FANS

For mounting outside of ducts. Wide range of sizes and models for various mounting arrangements. Get data sheet.



KITCHEN VENTILATORS

Built-in becomes permanent part of kitchen wall. Quiet, high capacity, efficient, equipped with ILG Self-Cooled Motor. 3 sizes. Also models for window installation Get Catalog No. 142.



NIGHT COOLING FANS

For permanent installation in attic, use ILG Self-Cooled Motor Propeller Fans (top of page) Portable models for use at downstairs windows.

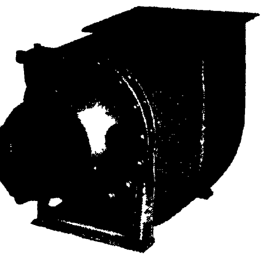
FOR ILG UNIT HEATERS AND COOLERS, SEE PAGE 878

DIRECT-CONNECTED BLOWERS

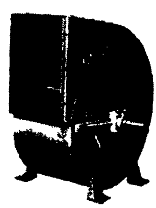
"BC" Type Load-limiting type with backward curved blades Motor load remains constant over wide range of air volume and change in static pressure Wheel mounted directly on motor shaft with motor partially recessed in side of blower. No motor base required Unobstructed inlet Also available for belt-drive 11 sizes Get Catalog No 241

"B" Type—Motor recessed in side of blower, eliminating separate base Multiblade, forward-curved wheel direct-connected to motor shaft. Also available for belt drive. 12 sizes Get Catalog No .241.

Type "B" Volume Blowers— Small volume, low pressure, quiet running. Multiblade wheel direct-connected to motor shaft. Cast-iron base. Universal discharge. 12 capacities. Get Catalog No. 142.



"CC" Type—All steel, heavy gauge, hot dipped galvanized, welded housing—no angle iron braces. Flanged inlet and outlet connections. Non-overloading type wheel mounted directly on motor shaft, with motor partially recessed in housing to save space—no motor sub-base required. Unobstructed inlet 11 sizes. Get Data Sheets.



"BW" Type—Housing similar to ILG Direct-Connected Blowers Wheel with forward curved blades. Full universal discharge, right or left hand drive. Wide range of sizes Get Catalog No 241.

Type "P" Volume Blowers—For exhausting dust, fumes, removal of steam, vapors. Four discharge positions to avoid friction in short bends. 7 capacities. Get Catalog No. 142.

Type "6S" Utility Blowers—Particularly suitable for building into apparatus which requires ventilation or air movement. Extremely flexible in arrangement, furnished with or without inlet flange, outlet flange, stand, etc., providing a practically custom-built unit on basis of quantity manufacture. Wide pressure range. Low power input. Get specifications, including dimensions and performance graph.

LA-DEL CONVEYOR & MFG. COMPANY

ENGINEERS OF

LA-DEL

AIR MOVEMENT

NEW PHILADELPHIA, OHIO

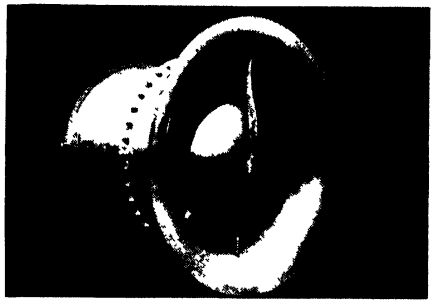
ORIGINATOR AND PIONEER
MANUFACTURER OF AXIAL FLOW FANS

ARMY E NAVY

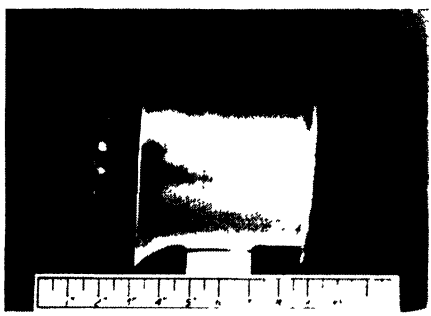
LA-DEL AXIAL FLOW FANS

Axial Flow the air flow principle selected by the U S Navy, Merchant Marine and private ship builders as the modern answer to the problem of providing top-efficiency in marine ventilating systems—merits the attention of everyone interested in *complete air control* for all types of installations

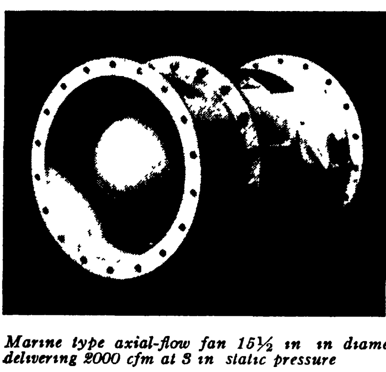
Illustrated on these two pages are a few of the developed sizes of La-Del Axial Flow Fans—the fan that truly propels air along the axis of the motor shaft, *straight in its course as the probing finger of a radar beam!* Models are manufactured ranging from 1/8 hp to 1000 hp; from 1/4 in static pressure to 45 in. static pressure; from 6 in diameter to 11 ft in diameter, from 400 cfm to as much as 600,000 cfm.



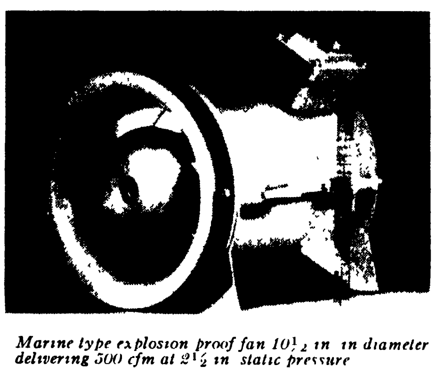
Marine type adjustable pitch axial-flow fan with flared intake



Compact Aviation type axial-flow fan only 6½ in in diameter by 6 in long delivering 400 cfm at 6 in static pressure



Marine type axial-flow fan 15½ in in diameter, delivering 2000 cfm at 3 in static pressure



Marine type explosion proof fan 10½ in in diameter delivering 500 cfm at 2½ in static pressure

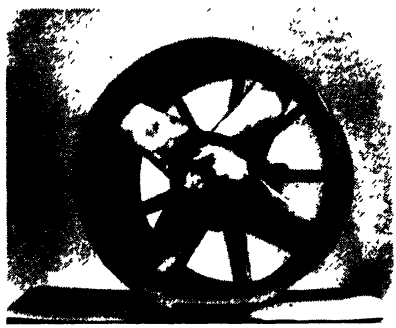
EFFICIENT AIR PROPULSION METHOD

Performance, economy and increased operating range possible with La-Del Axial Flow Fans guarantee widely increasing use And in keeping with these new demands, La-Del's research-engineering department has developed many additional sizes, designed to meet marine, aviation, industrial and domestic requirements with real economy and utmost efficiency

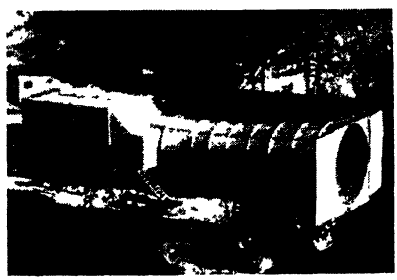
Models with fixed pitch blades, or with La-Del's *exclusive* adjustable or controllable pitch blades, broaden the useful operating range of a fan of any given size by fully 75 per cent Such fans are already designed, tested and serving with distinction in wartime installations

POST-WAR VENTILATION

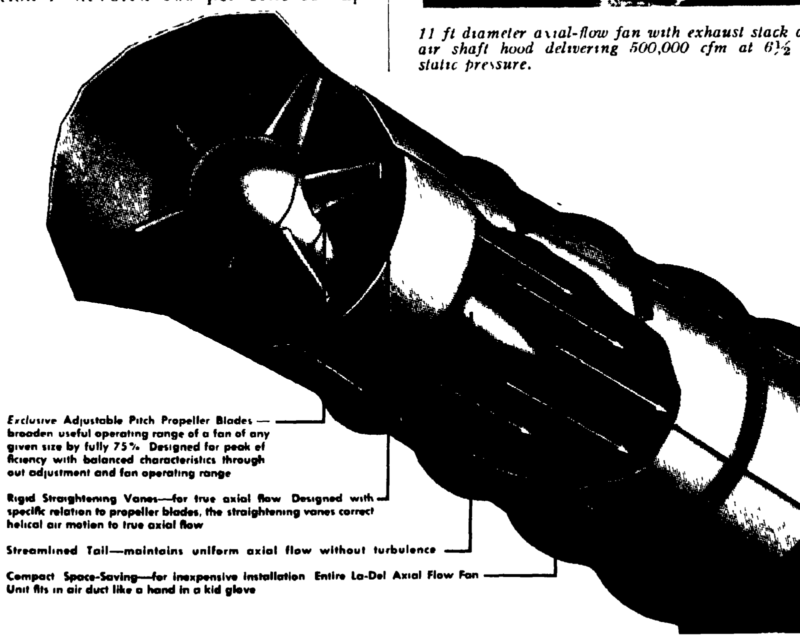
Industrial, domestic, marine and commercial post-war ventilation requirements will undoubtedly bring many new demands for additional sizes of these compact, efficient La-Del Axial Flow Fans La-Del is prepared to meet these specific needs with quiet operating, space-saving equipment designed to reduce power consumption costs and to bring freedom from maintenance troubles Efficient production planning, expert engineering and modern manufacturing make LA-DEL a name to remember in post-Victory ventilating For the duration, La-Del's production is devoted 100 per cent to supplying our Armed Forces, but your inquiry regarding specific applications are always welcome.



All aluminum Aviation type axial-flow fan 70 lbs. complete with 2½ hp motors, 4300 cfm at 2 in static pressure



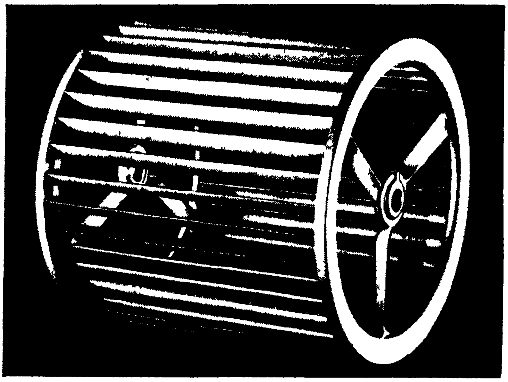
11 ft diameter axial-flow fan with exhaust stack and air shaft hood delivering 500,000 cfm at 6½ in. static pressure.



MORRISON PRODUCTS, INC.

East 168th St.
& Waterloo Rd.

Cleveland 10, Ohio



Airstream

BLOWER WHEELS

for MANUFACTURERS OF WARM AIR HEATING AND AIR CONDITIONING

Every manufacturer whose products employ blowers will welcome the AIRSTREAM Blower Wheel—an outstanding advancement in the art of fan design and manufacture. The AIRSTREAM Blower Wheel is a definite stride forward in aerodynamic design.

• ONE-PIECE BLADE CONSTRUCTION

All blades in every AIRSTREAM Blower Wheel are made from one strip of soft steel on progressive die equipment, eliminating any possibility of loose blades.

• THREE-PIECE BALANCED ASSEMBLY

Two pressed end rings with integral hubs and a one-piece blade assembly constitute the complete blower wheel When spotwelded together, these three pieces form a strong, rigid, true running blower wheel.

• EQUALIZED WEIGHT DISTRIBUTION

End-mounted AIRSTREAM Blower Wheels distribute their weight closer to the bearings. Smaller shafts and bearings are therefore permissible Shaft whip is eliminated and shaft deflection reduced.

• 40-60 PER CENT LIGHTER IN WEIGHT

AIRSTREAM'S unusual design reduces weight to a minimum. Integral pressed hubs eliminate heavy cast or screw machine parts. Lightness affects motor loads—makes for quieter operation and reduces starting torque.

AIRSTREAM Blower Wheels are furnished in standard diameters of 10 in., 12 in., 14 in., 16 in. and in a wide range of widths. Special diameters and widths can be furnished if required in large quantities. **Engineering data upon request.**

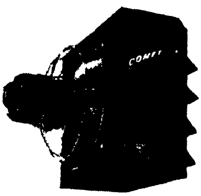
The NEW YORK BLOWER COMPANY

GENERAL OFFICES
32nd STREET & SHIELDS AVENUE • CHICAGO 16
FACTORIES at LA PORTE, IND and CHICAGO ILL

Representatives in Principal Cities

FANS • BLOWERS • UNIT HEATERS • AIR WASHERS
HOT BLAST HEATING SURFACE

Comet Unit Heaters


Comet Unit Heater

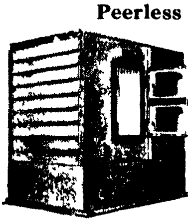
Heavy duty, welded steel, fin-and-tube heating element. Suitable for continuous heating service on steam pressures up to 150 lb or more 10 sizes with capacities from 31 Mbh to from 31 Mbh to 300 Mbh *Bulletin 431.*

Comet Exhaustair


Comet Exhaustair

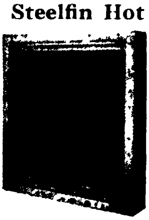
Delivers large volumes of air at low resistance and low current consumption All wheels are machine balanced for smooth, vibrationless operation Made in three types and ten basic sizes Wheel diameters from 12 in to 60 in Direct or belted drive Capacities from 400 cfm to 33,500 cfm *Ask for Bulletin 442*

Peerless Air Washers


Peerless Air Washer

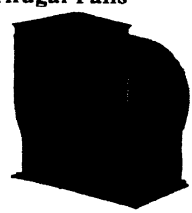
A completely engineered line of Air Washers for Cleansing, Conditioning, Cooling Complete with Single-and-Double-bank Atomizing Spray Systems, Marine Type Doors, Eliminators, Entering and Back Spray Louvers, Water Strainers, Pumps and Motors, and with or without Humidity Flooding Provisions Sizes and capacities from 3,600 cfm to 76,000 cfm

Steelfin Hot Blast Heating Surface


Steelfin

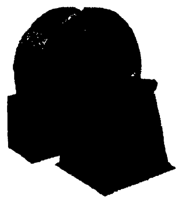
Extra heavy duty, fin-and-oval tube, all-steel, welded construction. A hot dip metallic coating over all, including headers, affords perfect bonding and conductivity. Suitable for continuous heating service on steam pressures up to 150 lb

Type ME Centrifugal Fans


Type ME Fan Fixed Housing

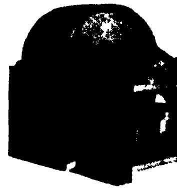
Quiet operating slow speed wheels for heating, ventilating and air conditioning systems, or high speed wheels with non-overloading horsepower characteristics for industrial applications Wheel diameters from 15 in to 80 in, with any speed or discharge required Class I, II or III construction Capacities up to 108,000 cfm

Type SSP Industrial and Heat Fans


SSP Heat Fan

For dust and gas removal, conveying of materials and handling hot gases in industrial processes Housings, drives, both pulley and direct, and discharge arrangements to meet any requirement. Wheel diameters from 14 in to 66 in Capacities from 450 cfm to 60,500 cfm *Details and engineering data in Bulletin 432.*

General Purpose (Jeep) Fans


General Purpose Fan

Portable, self-contained units for Class I industrial and ventilating applications Recommended for ease of installation, low maintenance and space saving features. Made in three types and eight basic sizes Capacities 400 cfm to 12,000 cfm. *Bulletin 443*

Certified Ratings

All New York Blower Company products are laboratory tested, accurately rated and fully guaranteed in strict accordance with Standard Codes.

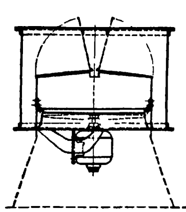
Member of National Association of Fan Manufacturers and Industrial Unit Heater Association.

PROPELLAIR Inc. 1944 CLARK BLVD. SPRINGFIELD, OHIO

VENTILATING SPECIALISTS IN ALL PRINCIPAL CITIES

Certified Ratings PFMA

Designed by Propellair to deliver maximum air with minimum horsepower. Air flow is even over all of blade—the *whole* fan works, not just the tips! Non-overloading—from free air to block-off, horsepower is virtually constant as long as motor speed is constant. Number of blades, angle and shape, depend on use.

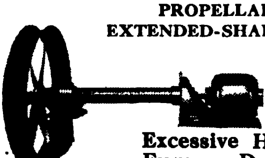


PROPELLAIR VERTI-STACK

For Roof Ventilation

A fool-proof, automatic ventilator installed easily on roof. Butterfly dampers, within wind guard, offer virtually zero resistance. High velocity, vertical discharge throws heat and fumes far above the roof and prevents rain from entering when the fan is in operation. Dampers close gently and positively when fan coasts to a stop. 12 to 60 in. diameters, with capacities from 2100 to 68,000 cfm.

PROPELLAIR EXTENDED-SHAFT FANS



For Handling Excessive Heat, Moisture, Fumes, Dust and Gases

Type "CE" extended-shaft fans for severe conditions involving excessive heat, moisture, harmful fumes, explosive gases, etc.—where a motor should not operate within air stream. Drive shaft housed and sealed within steel tubing. Special formed-steel ring and support assembly. Fan sizes 12 to 60 in., capacities 2000 to 68,000 cfm.

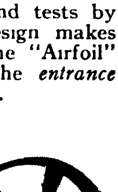
PROPELLAIR DRUM-TYPE

Belt-Drive Fans for Heat, Acids, Alkalis, Fumes, Gases or Dust

"CS" and "CSV" drum-type, belt-drive fans for severe, heavy-duty jobs involving elements listed for "CE" type. Sturdy, welded drum construction. Motor completely isolated from fan, driving belt protected in steel tube. 20 to 48 in. duct sizes, capacities 2400 to 43,000 cfm

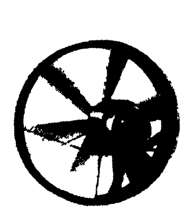
CURVED ENTRANCE RING

An Important Propellair Feature



In addition to serving as a sturdy support assembly, it reduces tip loss and enables Propellair Fans to deliver *maximum air per horsepower.* Introduced in 1930 as a result of exhaustive experiments and tests by Propellair engineers, this design makes possible the utilization of the "Airfoil" air-movement principle in the *entrance ring* as well as in the *propeller.*

PROPELLAIR DIRECT-CONNECTED FANS



For Duct, Wall or Hood Installation

Type "CD" direct-connected fans for wide range of jobs, from free air conditions such as window or panel mountings, to installation in roof ventilators, hoods, or duct work involving medium to relatively high resistance. Successful wherever a motor can operate directly within air stream. Sizes 12 to 60 in., capacities 800 to 85,000 cfm.

B. F. STURTEVANT COMPANY

Air Conditioning, Heating, Ventilating, Dust Control and Fume Removal Equipment, Vacuum Cleaners, Dryers, Compressors, Motors, Turbines, Mechanical Draft Equipment

Main Office and Works

Hyde Park, Boston, Mass.

Sales Engineering Offices

Akron, Ohio				St. Louis, Mo.
Albany, N. Y.				Salt Lake City, Utah
Atlanta, Ga				San Francisco, Calif.
Baltimore, Md.				
Boston, Mass.	Denver, Colo	Jacksonville, Fla	Newark, N J.	Seattle, Wash.
Buffalo, N. Y.	Des Moines, Iowa	Kansas City, Mo.	New York, N. Y.	Spokane, Wash.
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Cincinnati, Ohio	Greensboro, N C.	Memphis, Tenn.	Pittsburgh, Pa.	Tampa, Fla
Cleveland, Ohio	Hartford, Conn.	Milwaukee, Wis	Portland, Ore.	Toledo, Ohio
Columbus, Ohio	Indianapolis, Ind	Minneapolis, Minn	Richmond, Va	Washington, D C

PLANTS Located at HYDE PARK, BOSTON, MASS., LASALLE, ILL.; CAMDEN, N. J.; BERKELEY, CALIF.; and GALT, ONT.

B. F. STURTEVANT COMPANY OF CANADA, LTD , GALT, ONT.; Sales Offices in Toronto and Montreal, and representatives in principal Canadian Cities.

COOLING & AIR CONDITIONING DIV. of B. F. Sturtevant Co. is Organized to Engineer and Install Complete Industrial Air Conditioning Systems.

OFFICES. Atlanta, Ga ; Hyde Park, Boston, Mass ; Camden, N J.; Cleveland, Ohio; Greensboro, N. C.; Los Angeles, Calif.; New York, N. Y.

HOW STURTEVANT ENGINEERING SERVICE CAN HELP YOU

As the world's largest concern engaged in the manufacture of Air Handling Equipment, backed by more than 80 years' experience, B. F. STURTEVANT COMPANY is exceptionally qualified to help you attain the most efficient and economical solution of any air handling problem. Skilled Sturtevant engineering experts, located in many leading cities are prepared to render the following 5-point service: (1) Analyze your problem. (2) Recommend the solution. (3) Specify the equipment (4) Supervise the installation. (5) Check the operation.

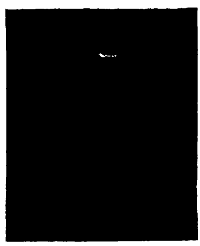
Whether requirements call for a single unit of apparatus or a complete engineered system, Sturtevant engineers can recommend the solution best suited to fulfill your individual needs. Do not hesitate to call the Sturtevant representative nearest you for assistance. Needless to say, no obligation is incurred.

As a preliminary aid, we list on the following two pages the major types of equipment manufactured by us, together with their general applications and the reference numbers of catalogs available.

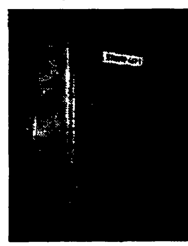
ILLUSTRATIONS OF SOME OF THE MAJOR LINES OF STURTEVANT EQUIPMENT

Axiflo Fan (25)

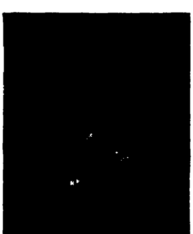
Propeller Fan Design 7 (29)



Multivane Fan Design 6 (30)



Speed Heater (40)



Centrifugal Compressor Design 14 (14)

Filterwasher Showing Filter Spray Nozzles (6)

Planovane Fan Design 3 (32)

Centrifugal Compressor Design 9 (12)

STURTEVANT EQUIPMENT INDEX AND OTHER ILLUSTRATIONS ON NEXT PAGE

B. F. STURTEVANT COMPANY

TABULAR VIEW OF STURTEVANT EQUIPMENT AND APPLICATIONS

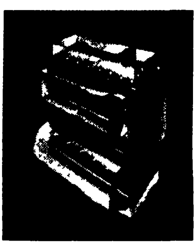
This table shows at a glance types of equipment manufactured by B. F. Sturtevant Company and their general application, together with catalog numbers on specific equipment. The numbers shown under "General Application," in the left column refer to the Index of Products. If no catalog number is given, write to B. F. STURTEVANT COMPANY, main office, or to the nearest branch office, stating specific requirements, and complete information will be sent immediately.

General Catalog No. 500 gives brief descriptions, Capacities, etc. of all Sturtevant Products

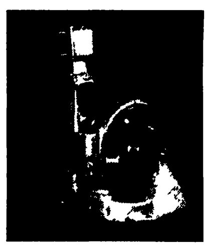
GENERAL APPLICATIONS
AIR CONDITIONING
1, 2, 3, 4, 6, 7, 15, 25, 30, 31, 33, 39, 43, 45, 47, 51, 57
DUST AND FUME REMOVAL
7, 8 9 10, 11, 12, 13, 14, 17, 18, 25, 27, 31, 32, 45, 52, 53, 54, 55, 56
HEATING
7, 25, 30, 31, 33, 40, 41, 42, 43, 49, 50, 57
INDUSTRIAL DRYING
9, 16, 17, 25, 28, 30, 31, 32, 42, 43, 45

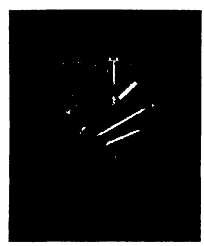
	STURTEVANT EQUIPMENT INDEX	TRADE NAME OR DESIGN NO.	CAT. NO.
1	Air Blenders	"Air Blenders"	
2	Air Conditioning Apparatus	Sturtevant	425-2
3	Air Conditioning Systems, Industrial	Sturtevant	
4	Air Conditioning, Railway	"Railvane"	401-1
5	Air Heaters	Sturtevant	450
6	Air Washers	"Filterwasher"	453
7	Blowers, Ventilating	"Rexvane Vent Sets"	400-9
8	Blowers, Small Portable	"Big Midget"	F1266
9	Blowers, Pressure	"Steel Pressure"	297-1
10	Collecting and Conveying Systems	Sturtevant	291-2A
	Compressors, Centrifugal		
11	—vols 35-75 c f.m , 12-20 oz press	Design 7	408-2
12	—vols to 5,500 c f m ½-3 lbs press	Design 9	386-1
13	—vols 35-650 c f m , press to 5 lbs	Design 1	431
14	—vols to 60,000 c f m press to 5 lbs	Design 14	458
15	Cooling Coils (Extended Surface) Water and Direct Expansion	Sturtevant	
16	Drying Systems	Sturtevant	461
17	Dust Collectors, Industrial	Sturtevant	
18	Dust Collecting Systems, Fly Ash	S A I D.	463
19	Economizers, Cast Iron	Sturtevant	405
20	Engines, Vertical Steam	Sturtevant	
21	Engine Generator Sets	Sturtevant	
22	Exhausters (Material Handling)	"Planovane," Des 3	410-3
23	Exhausters, Slasher (Textile)	Sturtevant	432-1
24	Fans—Attic Ventilating	"Atticvane"	400-9
25	—Axial Flow, Pressure Type	"Axiflo"	444-1
26	—Engine Cooling	Engine Cooler	418-5

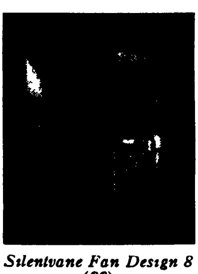
CONTINUED ON NEXT PAGE


Air Heater (5)

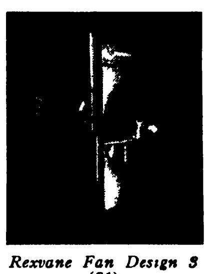

Steel Pressure Blower (9)

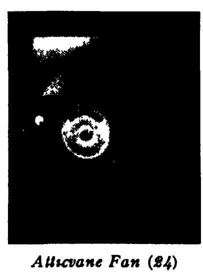

Engine Generator Set (21)


Engine Cooler for Gas, Diesel Engines, etc. (26)


Silentvane Fan Design 8 (33)


Silentvane Fan Design 7 (32)


Rexvane Fan Design 3 (31)

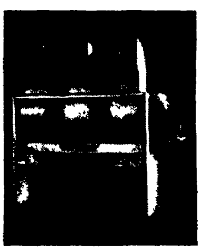

Atticvane Fan (24)

B. F. STURTEVANT COMPANY

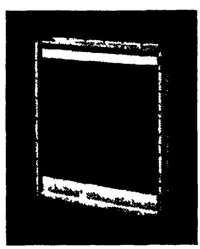
B. F. STURTEVANT EQUIPMENT—Continued

GENERAL APPLICATIONS		STURTEVANT EQUIPMENT INDEX	TRADE NAME OR DESIGN NO.	CAT. NO.
	27	—Fume and Dust Removal, Materials Handling	"Planovane," Des 3	410-3
MECHANICAL DRAFT	28	—High Temperature		
	29	—Propeller, Motor Driven	Design 7	400-9
5, 9, 12, 13, 14, 19, 25, 32,	30	—Low Speed, General Purpose	"Multivane," Des. 6	271-4
34, 35, 36, 37, 38, 45, 48, 57	31	—Medium Speed, General Purpose	"Rexvane," Des. 3	414-4
	32	—High Speed, Industrial	"Silentvane," Des. 7	449
	33	—High Speed, Heating and Ventilating	"Silentvane," Des 8	457
PNEUMATIC CONVEYING	34	—Mechanical Draft, Duplex type (Combined Forced and Induced)	Duplex	436
9, 10, 11, 12, 13, 14, 22, 27, 44, 45	35	—Mechanical Draft, Low Speed, Induced Draft, Abrasion-resistant	S P I D., Des. 2	447
	36	—Mechanical Draft, Medium Speed, Large Volume, Forced and Induced	M.V M.D., Des 6	409
	37	—Mechanical Draft, High Speed, Forced Draft	T V.F D., Des 9	448
PRIME MOVERS	38	—Mechanical Draft, High Speed, Induced Draft	T V.I D., Des 2	445
	39	—Theatre Ventilating	Theatre Fans	424-1
20, 45, 48	40	Heaters, Unit, Directional Flow Type for Wall and ceiling mounting	"Speed Heater"	396-9
	41	Heaters, Unit, Downblast Type, for ceiling mounting	"Downblast"	454
	42	Heaters, Unit, Large Capacity, for floor, wall, ceiling mounting	"Multivane"	452
	43	Heating Coils (Extended surface)	Sturtevant	462
VACUUM CLEANING	44	Melt-Recovery Units (Welding)	Sturtevant	438-1
	45	Motors, Electric	Sturtevant	433-1
12, 13, 18, 45, 52, 53, 54, 55, 56	46	Roof Ventilators	"Roofvane"	422
	47	Surface Dehumidifiers		
	48	Turbines, Steam, Helical Flow Type	Sturtevant	426
	49	Unit Ventilators, Schoolroom Type	Sturtevant	377-1
	50	Unit Ventilators, Auditorium Type	Sturtevant	S377-1
	51	Ventilating Sets, Direct Connected Motor	"Rexvane" Vent Sets	406
VENTILATING	52	Vacuum Cleaners, Portable	"Vortex"	413-2
	53	Vacuum Cleaners, Portable Furnace type	"Vortex"	373-6
7, 24, 25, 26, 29, 30, 31, 33, 45, 46, 49, 50, 51	54	Vacuum Cleaning Systems, Commercial Buildings	Sturtevant	397-2
	55	Vacuum Cleaning Systems, Industrial	Sturtevant	368-3
	56	Vacuum Cleaner Attachments	Sturtevant	387-1
	57	Vane Control (Fan)	Sturtevant	446

Downblast Heater (41)



Multivane Heater (42)



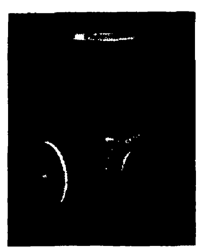
Extended Surface Heating Coil (43)

Sturtevant Electric Motor (45)

Sturtevant Steam Turbine Design 12 (48)

Melt Recovery Unit Portable Type (44)

Rexvane Vent Set (51)



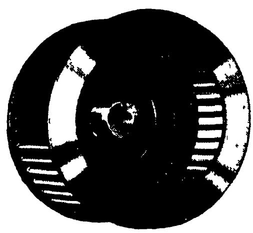
Vortex Portable Vacuum Cleaner (52)

The Torrington Mfg. Co.

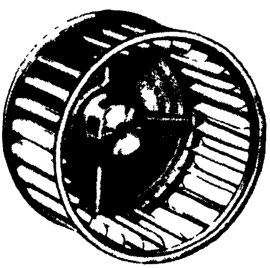
50 Franklin Street, **Torrington, Conn.**

Manufacturers of Blower Wheels and Propellor Type Fan Blades.

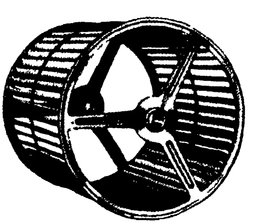
AIRISTOCRAT Quiet Propeller Fan Blades AIROTOR Blower Wheels AUTOCRAT Fan Blades



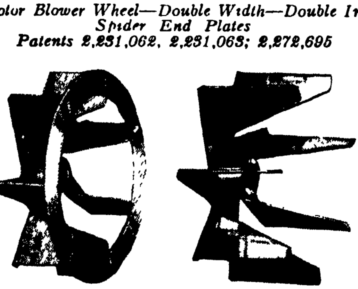
Single Inlet Aluminum Blower Wheel
Patent 1,700,017



Airotor Blower Wheel—Single Width—Single Inlet
Patents 2,231,062; 2,231,063; Des 126,043; 2,272,695



Airotor Blower Wheel—Double Width—Double Inlet Spider End Plates
Patents 2,231,062, 2,231,063; 2,272,695



Autocrat Paddle Type Blower Wheels

Torrington Aluminum Blower Wheels produce the smooth, quiet performance which is essential in modern heating and air conditioning units because the unique patented construction breaks up resonance and minimizes noise. Made of aluminum, they resist corrosion and their light weight facilitates quick starting—saves power. Every wheel is statically balanced. Sizes 3 in. to 15 in. diameter in all standard widths, which may also be made up as double width, double inlet wheels.

Torrington Airotor Blower Wheels are light, sturdy and inexpensive—incorporate new principles of design and construction, which insure rigidity and concentricity. **Single Width—Single Inlet** wheel is of simple four-piece construction. **No rivets or welds are used;** concentric rib serving as backing for blade strip is formed at same time as hub socket, insuring trueness of wheel. Rigid radial ribs prevent deflection by thrust. Three thicknesses of metal in rims make for maximum strength. Excellent for many heating and ventilating uses. Manufactured in both aluminum and steel in $1\frac{1}{2}$ in., 2, 3, $3\frac{5}{8}$, $4\frac{1}{2}$, 5, 6, $7\frac{1}{2}$, 9 and $10\frac{1}{2}$ in. diameters. Clockwise or counterclockwise rotation. Same sizes available in DA type double width, double inlet wheels.

Torrington Airotor Blower Wheel—Double Width—Double Inlet—Spider End Plates. Has blades punched and formed in a single strip, rigidly held by flanged single piece end rings. Hubs are rigidly mounted by peening. Wheels of $3\frac{5}{8}$ in., $10\frac{1}{2}$, 12 and 16 in. diameter are available at present.

AUTOCRAT Paddle Type Blower Wheels are suitable for automobile defrosters, electronic cooling applications, small hair dryers, and any other purposes where small air deliveries and inexpensive wheels are required. Reenforced construction is sturdier, operates at higher speeds. Available in 2, $2\frac{1}{2}$ and 3 in diameters.

New Blower Wheel Catalog charts capacities, gives detailed dimensions, suggests housing dimensions for all above wheels. Valuable "Tips" for designing air impelling units are also included.

AIRISTOCRAT Quiet Propeller Fan Blades are widely recognized for their all-around excellent performance. The unique, patented construction embodies entirely new principles in the art of fan design—produces a blade unsurpassed for quiet operation, rugged construction and attractive appearance. Every **Airistocrat** unit is carefully built and the blades are hand gauged for correct contour and alignment. Statically balanced, these blades deliver full air volume with a minimum of noise. Aluminum alloy blades and steel spiders are standard except where otherwise noted.

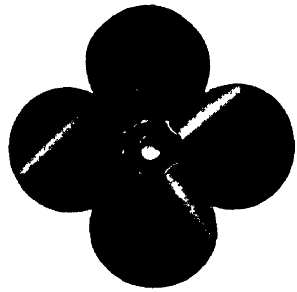
Available in the following finishes: 1. Plain—no finish on blades, spiders or hubs. 2. Blades with no finish; spider and hub with cadmium plate or black lacquer. 3. All black lacquered, with or without center button. 4. Buffed and lacquered blades, black lacquered spider and hub, with or without center button. Catalog gives detailed dimensions and guaranteed performance curves recorded under *NEMA* and *NAFM* code tests at various speeds for each of the **Airistocrat** models described below.

"Standard" Series—Has blades mounted on a steel spider. Sturdy, attractive steel or aluminum blades which have withstood extreme laboratory breakdown tests. Sizes 8 in., 10, 12, 14, 16, 18 and 20 in. diameters in a variety of pitches to meet every need.

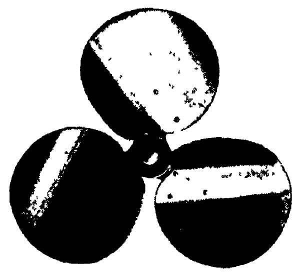
Three Blade "Y" Series—The design of this blade is the result of two years of laboratory experiment to produce a better air circulator blade. At recommended speed these blades produce a high velocity air stream effecting deep penetration with unusual quietness. Sizes 10 in., 12, 14, 16, 18, 20, 24 and 30 in diameters, steel or aluminum blades.

Pressure "P" Series—Similar in construction to "Standard" Series but with blades especially designed for higher pressures. Sizes 10 in., 12, 14, 16 and 18 in. diameters.

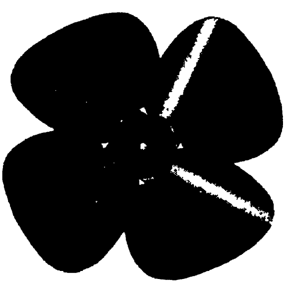
Pressure "U" Series—Two and four blade models of steel designed for pressure operation. Sizes 20 in , 22, 24, 26, 28 and 30 in. diameters.



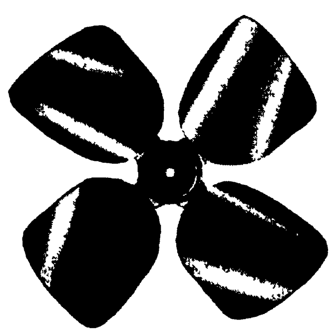
Airistocrat "Standard" Series
Pats 2,072,322 and 2,021,707



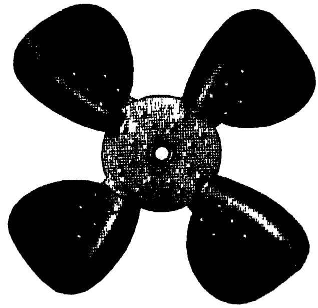
3-Blade Airistocrat "Y" Series



4-Blade Airistocrat Pressure Fan "P" Series

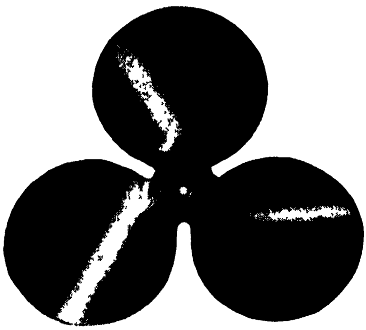


4-Blade Airistocrat Pressure Fan "U" Series
(also made in two blade model)



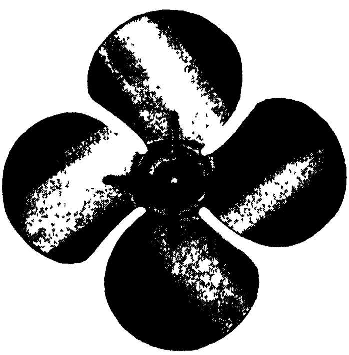
4-Blade Airistocrat Attic Fan "B" Series (also made in 3 and 5 blade models)

AIRISTOCRAT "B" Series Attic Fan Blades represent a complete redesign of the popular "A" Series. Efficiency has been still further improved. The same proportions, proved aerodynamically correct, have been carefully adhered to in all diameters New larger center disc and heavier spider arms increase strength and a new blade shape adds to the appearance of this carefully designed product. Flexibility of parts allows assembly with either 3, 4 or 5 blades in standard diameters 24, 30, 36, 42 and 48 in All steel construction Available in the following finishes: 1 Plain 2 Colored lacquered blades, black lacquered spider and hub 3 All one color lacquer



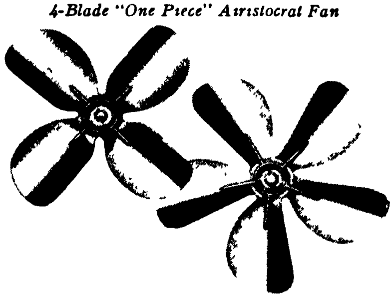
Airistocrat 3-Blade "One Piece" Series

3-Blade "One-Piece" Series Propellor Fan—An attractive, inexpensive one-piece blade incorporating the **Airistocrat** features for quiet operation Available in both steel and aluminum. Sizes 8 in and 10 in diameters.



4-Blade "One Piece" Airistocrat Fan

4-Blade "One-Piece" Series Propellor Fan—An exceptionally rigid model blanked from one piece of metal with four wide blades. Quieter than narrow blade types Made in both steel and aluminum. Clockwise rotation only (viewing air delivery side) Sizes 8 in., 9, 10, 12 and 16 in. diameters. Available in the following finishes: 1. Plain 2 Lacquered. 3 Nickel or cadmium plated (steel only).



Autocrat Fan Blades

AUTOCRAT Fan Blades—For auto heaters, windshield defrosters, electric heaters, etc Have been standard ever since these devices were first marketed. Made in sizes 3 in , 4, $4\frac{1}{2}$, 5, $5\frac{1}{4}$, $5\frac{1}{2}$, 6, $6\frac{1}{4}$, $6\frac{1}{2}$ in , all four blades, also $5\frac{1}{2}$, 7, 8, 9, 10 in. 5-blade, in one piece of cold rolled steel or aluminum with brass hubs, complete with set screw. Either clockwise or counter clockwise rotation (expressed when looking at air delivery side of fan) White nickel is standard finish for steel blades.

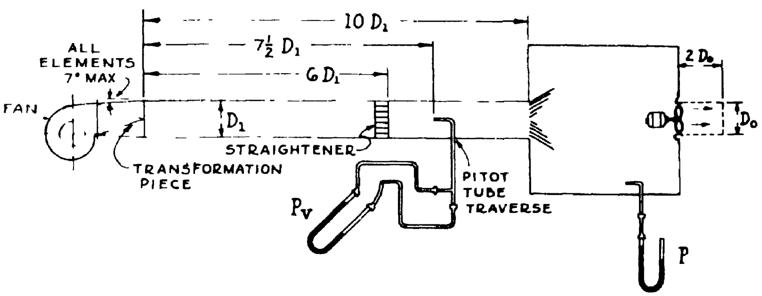
PFMA

Propeller Fan Manufacturers Association

2-255 General Motors Bldg., **Detroit 2, Mich.**

When you see the **PFMA** Certified Rating Insignia, you know that the company displaying it certifies that the performance of its products is in strict accordance with the Standard Test Code.

Air tunnel design for Standard Test Code measurements is illustrated below, the propeller fan under test being mounted at the discharge end



Exhausters without Duct

Supply Duct
Chamber
Fan Outlet

Supply Duct: Velocity pressure, p_v, Differential reading between impact tube and static tube **Chamber:** p by impact tube **Fan Outlet:** P_v; Correct p_v to P_v, the velocity pressure D_0 corresponding to the area A of the minimum ring diameter P_t, add p to P_v P_s; Subtract P_v from P_t Correct all pressures to Standard Air Conditions All additions and subtractions mentioned above are arithmetical

Testing equipment and methods are prescribed in detail by the Standard Test Code, from which the accompanying air tunnel dimension diagram is reproduced to indicate the exactitude which is required in PFMA testing

Copies of the complete Standard Test Code are available for 25 cents through the Association

The **PFMA** insignia shown at the right, is your assurance of a dependable basis to judge fan performance Guesswork is eliminated and data listed are reliable and based on sound engineering technique, permitting you to evaluate comparative performances

Where fans are rated by Standard Test Code procedure, volume, pressure, power input, mechanical efficiency and all related factors, are measured under precisely controlled conditions

Look for the PFMA insignia
Insist upon Certified Fan Ratings

Utility Fan Corporation

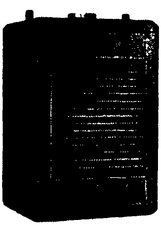
4851 S. Alameda St., **Los Angeles 11, Calif.**

Utility Gas-Fired Heating Equipment, Evaporative Coolers, Blowers and Fans

FORCED AIR FURNACES

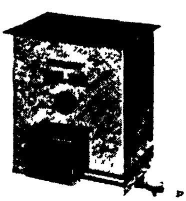
Basement and closet types . . . Compact design . . . Multiple-fin element with air-cooled hollow baffle . . . Element guaranteed against burnout forever . . . Automatic controls . . . Filters . . . Built-in motor overload protection.

UNIT HEATERS



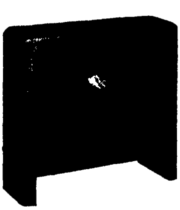
Compact heat exchanger with no inside baffles—Individual burners for each element section—Silent, disc-type fan—All-welded cabinet—Directional grilles—Built-in draft diverter—Temperature limit control. Four sizes.

FLOOR FURNACES



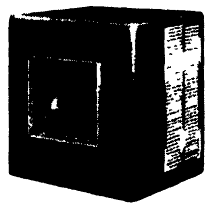
All-welded construction . . . Die-stamped grilles . . . heavy cast-iron burner . . . removable inside jacket . . . interlocking gas valve. Floor or dual registers. 25,000, 37,000 and 50,000 Btu input.

CIRCULATING HEATERS



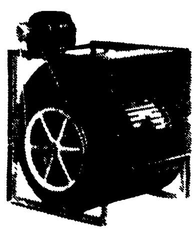
Fan sends stream of air through nozzle-shaped outlet to hold warm air in living zone. Built of heavy furniture steel . . . all die-formed and electric welded. Vented and unvented models.

EVAPORATIVE COOLERS



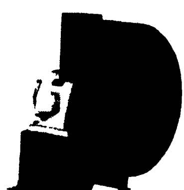
Comfort cooling—residential, commercial, industrial. Exclusive feature—Uni-flowmeter (Pat. Pend.) for uniform water distribution. Patented No-Sag cooling pads. 14 Models.

STANDARD BLOWERS



Dynamically balanced, multiple-vane centrifugal blowers. Four-side angle iron frame increases rigidity, eliminates vibration—permits installation with any of four discharge positions.

EXHAUSTERS



High flow and high pressure designs in four drive arrangements, for exhausting many materials. High efficiency . . . decreased sound. Available with acid-proof plastic surfacing.

CLOVERLEAF TYPE FANS

Propeller Fans are available with Cloverleaf (over-lapping) or Streamline type blades, with direct or belt drive for mounting into windows, openings, or on steel or wood panels

Utility Blowers are tested in accordance with the A.S.H.V.E. Code.

Write for complete information, catalogs and prices.

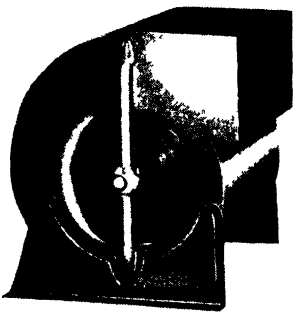
The Viking Air Conditioning Corporation

5600 Walworth Avenue, **Cleveland, Ohio**

Viking products, which include blowers, fans, and humidifiers, offer top economy and efficiency to furnace and air conditioning manufacturers because Viking ***specializes*** **in equipment for air conditioning. The economies of mass production due to specialization are thus passed on to all Viking customers.**

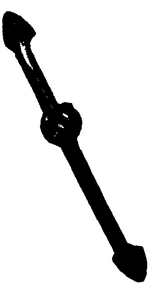
BLOWERS AND BLOWER PARTS FOR

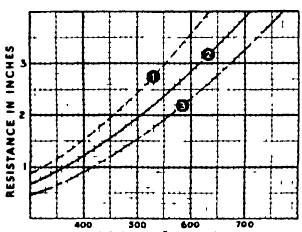
Air Conditioning Manufacturers



COMPLETE ASSEMBLY

Viking's blower assembly embodies several entirely new features, among which are the streamlined bearings, the new-type blower wheel, the lower outlet, and the smaller unit size overall. Space savings alone in 10 in. blowers run 2½ in. in height and 1½ in. from front to back. Extremely specialized equipment plus high production of 9 in., 10 in., 12 in., 14 in., and 16 in. blowers result in completely assembled blowers at practically the cost of parts only.

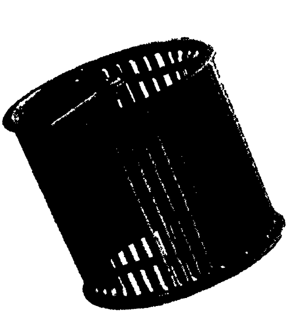




1 *Conventional Bearing Assembly*
2 *Viking Streamlined Bearing*
3 *Inlet Free—no obstruction*

STREAMLINED BEARINGS

Important innovation in bearing design is this unit Viking Streamlined Bearing. Note the graph at left for comparative record of Free Areas. You see that whereas a typical bearing allowed 77 per cent Free Area for air intake, the Viking Streamlined Bearing permits an 85 per cent Free Area, resulting in improved overall performance of the entire blower assembly.



BLOWER WHEEL

Viking blower wheels are produced by a continuous stamping process and consist basically of only 6 parts—2 blade sections, 2 spoke ends and 2 hubs. Larger inlets help provide increased air delivery. The wheels are designed for the higher pressures of modern air conditioning requirements. Costs are lower than for any pre-war wheel.

Send For More Information

Though parts alone are available if desired, the majority of standard requirements can be met most profitably with a Viking assembly. As specialists in making air conditioning blowers, we invite your inquiries. Write for our Special Manufacturers Brochure, which explains Viking facilities and products. Address Desk A.

GENERAL ELECTRIC

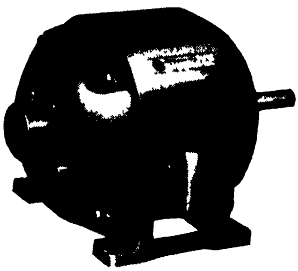
Schenectady, N. Y.

Apparatus Sales Offices, Warehouses, Service Shops, and Distributors in Principal Cities

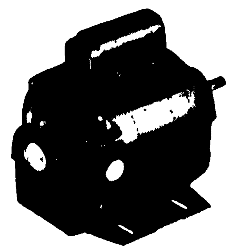
MOTORS FOR HEATING, VENTILATING, AND AIR CONDITIONING

General Electric offers a complete line of motors for compressors, fans, and pumps from which you can select easily the motors with electrical and mechanical characteristics best adapted to your equipment. Many of the most common applications are listed below. Complete information on other types of motors—vertical, enclosed, etc., with various electrical and mechanical modifications—can be obtained at a G-E office near you.

For additional information, ask for Motor Catalog GEA-4281.



Tri-Clad induction motor, Type K, polyphase*



Fractional-horse-power capacitor-motor Type KC

SOME G-E MOTORS AND THEIR USES

Application	Speed	Type Winding	Type	Horsepower Range	Classification
Fans and Centrifugal Pumps	Constant or Adjustable	Shunt	B & CD	1/8-200	Direct-current
Reciprocating Pumps and Compressors		Compound	B & CD	1/8-200	
	Constant	High-torque Capacitor	KC & KCJ	1/4-3	Single-phase Alternating-current
Small Direct-connected Fans	Constant	Resistance Split-phase	KH	1/40-1/3	
		Shaded-pole	KSP		
	Constant or 3-speed	Low-torque Capacitor	KCP	1/50-1	
Belted Fans Centrifugal Pumps	Constant	General-purpose	KC	1/4-3	
		Capacitor	KC	1/8-3	
		Repulsion-induction	SCR	5-10	
Reciprocating Pumps and Compressors	Constant or Multispeed	Squirrel-cage	K or KB	1/4-1000	Polyphase, Alternating-current
		High-starting-torque	K & KG	1/4-5 5-100	
Pumps, Compressors, Fans	Constant or Adjustable-varying-speed	Wound-rotor	M & MB	½-1000	
	Constant	Synchronous	TS	25-2000	

Types of Enclosures: Open (dripproof)—protected from falling objects or dripping liquids. Splashproof—where wetness is a factor. Totally enclosed—for complete protection. Explosion-proof—for inflammable gases. Dust-explosion-proof—for combustible dusts.

For code wire, conduit products, wiring material, insulating materials, etc., address APPLIANCE & MERCHANDISE DEPARTMENT, BRIDGEPORT, CONN

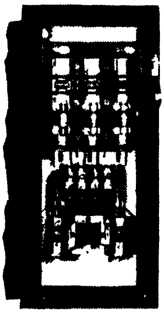
*Reg. U. S. Pat. Off.

GENERAL ELECTRIC

Schenectady, N. Y.

Apparatus Sales Offices, Warehouses, Service Shops, and Distributors in Principal Cities

CONTROL FOR HEATING, VENTILATING, AND AIR-CONDITIONING MOTORS

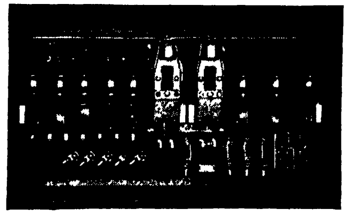


The General Electric line of standard control offers manual or automatic equipment for compressors, fans, or pumps driven by any type motor which you require, providing full protection for your motor, especially those listed on the preceding page.

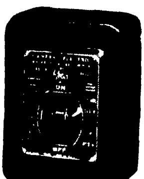
For special applications, General Electric controllers can be designed to meet your exact requirements.

For additional information, ask for Control Catalog GEA-606.

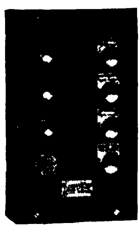
CR7008 combination a-c magnetic full-voltage starter Provides short-circuit protection, as well as overload and undervoltage protection



CR7107 a-c magnetic controller for use with multi-speed squirrel-cage motors, driving pumps, fans, or blowers

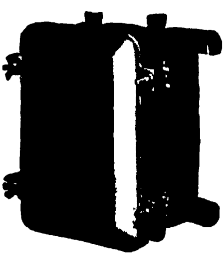


CR1062 manual full-voltage starter for small polyphase motors Available in 2-or 3-pole forms, with temperature overload protection

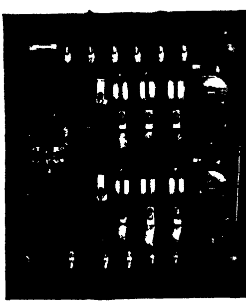


CR2940 indicating-light push-button station Used with magnetic controllers to indicate speed of fan or blower

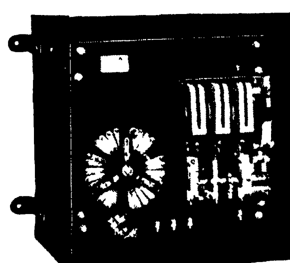
CR1061 manual starter for fractional-horsepower motors, a-c or d-c Available in 1- or 2-pole forms, with temperature overload protection



CR7006-D61FS quiet magnetic switch For full-voltage starting of squirrel-cage motors up to 5 hp, 220 volts For applications where quiet operation is required, such as on fans or domestic air-conditioning systems



CR7896 throwover panels To transfer motor or lighting load to emergency source of power if normal source fails Retransfers load when power returns to normal source



CR7764 a-c speed-regulating controller for wound-rotor induction motors For controlling the speed of motors driving ventilating fans and blowers Provides undervoltage and overload protection

A complete line of accessories, including pressure governors, pressure switches, float switches, electrically operated valves, and indicating selsyns, is available

The General Electric Company will gladly assist in the solution of any electrical problem in relation to heating and ventilation.

Established in 1891

Wagner Electric Corporation

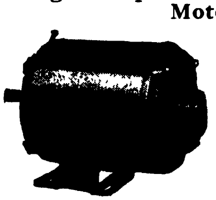
Sales Offices in 29 Principal Cities

6464 Plymouth Avenue, **Saint Louis 14, Mo., U. S. A.**

Wagner motors embody the very latest improvements in motor design, which contribute toward such essentials as simplicity, ruggedness, and dependability They successfully meet all requirements for good starting performance, quiet and cool operation, high efficiency, good power-factor and long life.

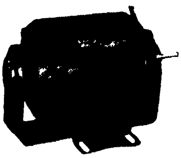
Wagner engineers will be pleased to assist your engineers in the selection of the correct motor for newly designed machine tools and other equipment.

Wagner Repulsion-Start-Induction Motors



Adaptable to heavy-duty applications such as **Stokers, Compressors, Pumps,** etc., all standard frequencies and voltages, sleeve and ball bearing; open, totally-enclosed and drip-proof; horizontal and vertical; rigid and flange mounted; small sizes resilient mounted (1/8 to 15-hp).

Wagner Unit Heater Motors

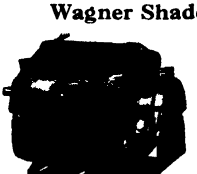


Particularly adaptable for **Unit-Heater** applications These motors are totally-enclosed to prevent entrance of dust or moisture; ball-thrust bearing on front end to take care of end-thrusts imposed by fans; small sizes resilient mounted for ultra-quiet operation. (1 to 3/4-hp)

Wagner Split-Phase Motors

For Unit-Heaters, Oil-Burners, and endless variety of machinery used in homes, factories and offices All standard frequencies and voltages; sleeve and ball bearings, open, drip-proof and totally enclosed, rigid, resilient and flange mounted. (1/20 to 1/3-hp).

Wagner Shaded-Pole Motors

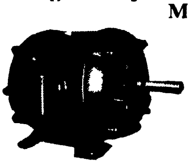


Ideal for **Fan and Blower drives,** or for applications requiring very small horsepower or low starting torque in which fans or blowers are mounted directly on the motor shaft Totally-enclosed and open-type; rigid and resilient mounted. (1/125, 1/80, 1/40, and 1/30-hp).

Wagner Capacitor-Start-Induction-Run Motors

For **Refrigerators, Oil-Burners, Stokers, Household Air-Conditioners** and other appliances. All standard frequencies and voltages; sleeve and ball bearings; open, totally-enclosed, and drip-proof, horizontal and vertical; rigid, resilient, and flange mounted (1/20 to 3/4-hp)

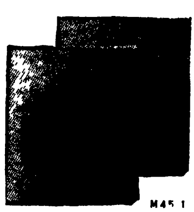
Wagner Open-Type Squirrel-Cage Motors



More widely applied than any other type of polyphase motor. All standard frequencies and voltages, sleeve and ball bearings, open and totally-enclosed; horizontal and vertical Made in several electrical types varied as to torque and current characteristics (2- and 3-phase, 1/6 to 400-hp).

Wagner-Direct Current Motors

For direct-current service wherever repulsion-start-induction, split-phase, capacitor, or squirrel-cage motors would be used for alternating current service Built in two types Appliance type up to 1½-hp, Industrial type, up to 3-hp.



Send for BULLETINS

Write for *Wagner Bulletin MU-183* on single-phase motors and *Bulletin MU-182* on polyphase motors.

Anemostat Corporation of America

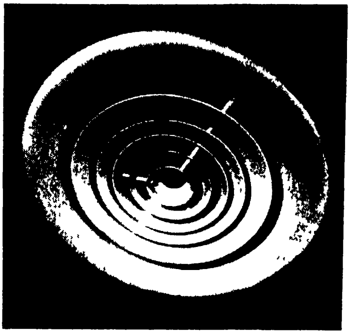
10 East 39th Street, **New York City 17, N. Y.**

The Anemostat High Velocity Air Diffuser

THE ANEMOSTAT PRINCIPLE

ANEMOSTATS produce unparalleled results because they are the only air diffusers which operate on the following interdependent principles:

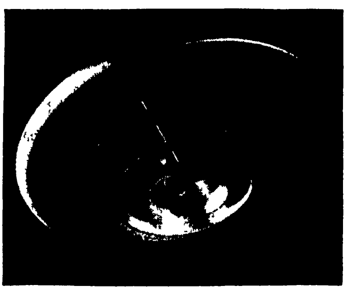
1. Air expansion within the device, which reduces velocity instantly.
2. True Aspiration, which causes room air equal to 30 to 35 per cent of the supply air to be drawn into the device where it is mixed with the supply air. The percentage of aspiration depends on the type of Anemostat used.
3. Creation of a multiplicity of air currents and counter-currents at low velocities, which causes slow but adequate secondary air motion.



Anemostat Type "AC"
For supplying, extracting or returning air

Type "AC" Anemostat is a combination device for supplying air and either extracting it or returning it to the conditioner. Designed to extract or return 75 cfm of room air for every 100 cfm of supply air. This percentage of extract or return may be increased or decreased by varying the extract velocities. It furthermore has an aspiration effect of 30 per cent. May be used with velocities up to 2500 fpm, and wherever both supply and return, or extract are required through the same unit. Should not be used with ceiling heights exceeding 14 ft.

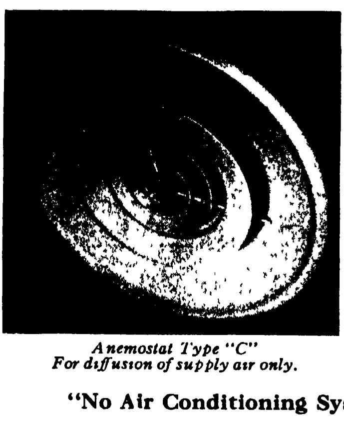
Type "AR" Anemostat is a diffusion device, for supply air only for use in applications where a relatively high rate of air change (more than 12 per hour) must be maintained. It has 35 per cent aspiration. It is particularly suitable where obstructions such as beams or columns are close to the diffusion unit and restrict the normal radius of diffusion



Anemostat Type "B"
For diffusion of supply air

Type "B" Anemostat is a diffusion device for supply air only. It has 35 per cent aspiration. May be used with velocities up to 4000 fpm and is suitable for industrial and commercial installations. Can be used on either exposed or concealed duct work.

Type "C" Anemostat is a diffusion device for supply air only. It has 35 per cent aspiration. May be used with velocities up to 2500 fpm. Must be installed flush with ceiling and cannot be used on exposed duct work.



Anemostat Type "C"
For diffusion of supply air only.

Type "HU-3" and "HU-4" Anemostats have been developed to obtain draftless, economical heat distribution from vertical discharge unit heaters, and uniform heat coverage of floor areas. Type "HU-3" and "HU-4" Anemostats may be combined with practically all sizes and types of Vertical Discharge Heaters on the market. A number of unit heater manufacturers now supply "HU" Anemostats as a part of their equipment. "HU" Anemostats may be used on duct work for general heating installations.

"No Air Conditioning System is better than its Air Distribution"

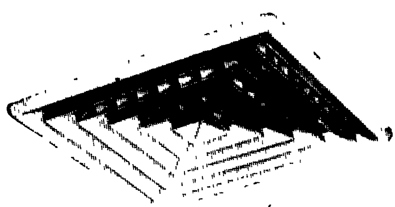
Air Devices, Inc.

Air Diffusers ● Exhausters ● Air Filters
Filter Holding Frames ● Hot Water Generators

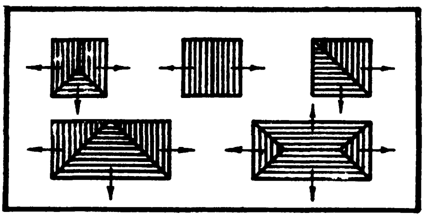
17 East 42nd St.
New York 17, N. Y.

Agents in All
Principal Cities

AGITAIR DIFFUSERS



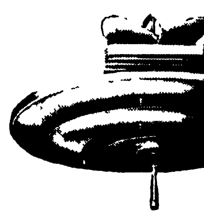
4-way



Type R AGITAIR is the only diffuser that offers the higher efficiency of the square or rectangular shape in diffusing air quietly, draftlessly and with rapid temperature equalization throughout the average square or rectangular room Its patented construction can be assembled into patterns which discharge the required amount of air in one to four directions

Each side delivers a quantity of air proportional to the areas served Thus the engineer or architect can select an attractive diffuser that fits his design - rather than make his design fit the diffuser.

Type A 125



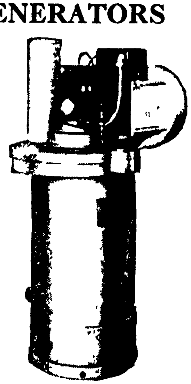
Model CSF with Type B Damper

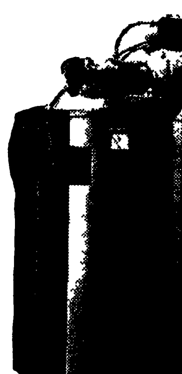
Circular AGITAIRS combine beautiful design with finest operating features to give rapid temperature equalization and draftless diffusion of air Model CSF is quiet, easy to install Pressure losses at minimum Type A diffusers are smaller, weigh less, are easiest to install Type CM is for marine use In all sizes for all types of mounting and with lighting combinations Dampers are provided where needed

The AGITAIR Diffuser Data Book, available to architects and engineers, will help you design and install air distribution systems Consult our engineers

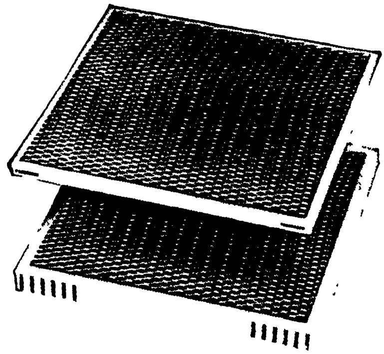
OIL BURNING HOT WATER GENERATORS

Ideal for hot water or heating service in domestic, apartment house, industrial and marine applications because they are light in weight and can be installed without special foundations or floor coverings The AGITAIR is down-fired to conserve floor space and facilitate servicing of the burner Burner is fully automatic, with simplified design to reduce weight, cost of installation and possibility of failure Special stacks are not necessary Type A, weight 300 lb, 400 to 1000 sq ft radiation. Type B in larger capacities from 1000 to 4500 EDR. *Ask for bulletin.*





AGITAIR AIR FILTERS



Permanent, cleanable AGITAIR Air Filters give more effective filtering with greater dust holding capacity and exceptionally low resistance. The filter media divides the air stream into myriads of streams which impinge upon and deposit dirt and dust on the viscous coated surfaces of the media. Construction is sturdy arc-welded steel, with protected faces. Agitair Filters are easier to clean. *Write for Bulletin AF44-1.*

Type P AGITAIR Filters, for heating, ventilating and air conditioning systems Capacity 2 cfm per sq in at 288 fpm Initial resistance: Type P1, 0 06 in. w g ; Type P2, 0 065 in. w g Sizes up to 1200 sq in. *Bulletin FP44-1.*

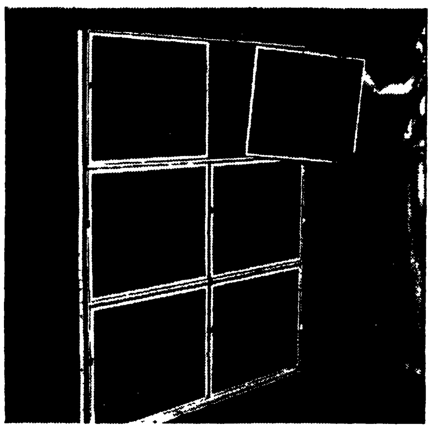
Industrial, heavy duty Permo-Aire filters. Capacity 2 cfm per sq in. at 288 fpm. Initial resistance: 2 in thick, 0.09 in w g , 4 in. thick, 0 10 in. w g Sizes up to 2400 sq in. AGITAIR engineers have solved many industrial air filtering problems, such as protection for heavy blowers and other equipment. Call on our experience

Grease Filters for kitchens, etc Capacity 1½ cfm per sq in with fan system, 1 cfm per sq in. on gravity exhaust Initial resistance. 0 07 in w g. at 216 fpm, 0 02 in w g at 72 fpm. Standard sizes up to 25 x 30 in ; others to 2400 sq in. Complete assemblies are available

FILTER HOLDING FRAMES

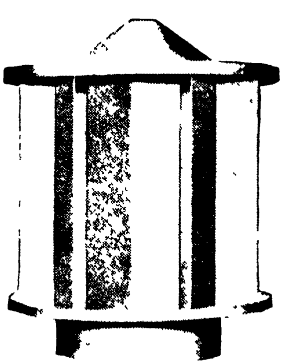
Adaptable to all types of installations Saves time and labor, eliminates bolts and rivets in frame, and minimizes air leakage Made of heavy steel, assembly is sturdy, neat. Available for all standard and special sizes of AGITAIR filters

V-Type assemblies of AGITAIR filter holding frames are available to increase filter area where space is limited.



Typical Filter Holding Frame

AGITAIR WIND-ACTUATED EXHAUSTERS



Provide proper ventilation regardless of wind direction, and with positive elimination of down-draft. Functions at peak efficiency at average low wind velocities. Will not restrict the flow of air or gases when there is no movement of outdoor air across the head.

Available in sizes from 4 in to 48 in. neck diameter, with capacities shown at right. For calculation of natural ventilation requirements, and for proper design principles, refer to Chapter 42 of this GUIDE. Fan-equipped units also for higher ratings. *Bulletin EX44-1.*

Capacity—Wind-Actuated

Size of Vent O D Neck	Neck Area Sq Ft	Nominal Capacity cfm
S 4"	0.0873	38
S 6	0.1964	86
S 8	0.3491	154
S 10	0.5454	240
S 12	0 7854	346
S 14	1.069	470
S 16	1.396	614
S 18	1.767	777
S 20	2.182	960
S 22	2.640	1162
S 24	3.142	1382
S 28	4.276	1881
S 32	5.585	2457
S 36	7.069	3110
40	8.727	3840
44	10.56	4626
48	12.51	5500

Barber-Colman Company

Rockford, Illinois

ENGINEERED AIR DISTRIBUTION OUTLETS

VENTURI-FLO
Overhead Air Diffusers

Venturi-Flo overhead air diffusers have flow characteristics similar to those of the well known fluid-flow measuring device—the Venturi Meter. The relationship between the neck area of the unit proper and the Venturi-Flo throat area is so proportioned as to create a slight back pressure in the neck at all times, thereby automatically insuring uniform distribution around the entire periphery of the unit.

Three types of overhead diffusers are available, the recessed, the surface, and the thermostatically controlled types. A wide range of sizes permits handling air volumes up to 15,000 cfm per unit. Fittings for attaching any standard light fixtures to the outlets may be obtained for all designs. The surface and recessed types can also be furnished as combination supply and exhaust units and with adjustable dampers.

Uni-Flo Grilles and Registers are especially designed for air conditioning applications. They are engineered and prefabricated with directional flow fins for each individual application. Proper air distribution is assured and the necessity for adjustment after installation obviated.

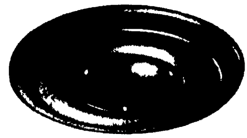
Uni-Flo Grilles and Registers can be furnished in a variety of shapes and sizes for plain and curved surfaces.

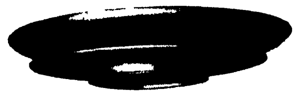
Registers are similar in construction to grilles but with the addition of spring loaded positive closing fan or key-operated dampers.

Electroplated Finishes: Gunmetal, brushed bronze, plain zinc, and satin copper; also available in gray prime coat and satin aluminum.

Uni-Flo Air Distribution Accessories: Include the Volocitrol and the Elturn. The Elturn is a scientifically designed and highly efficient air turning member. The aero-dynamically correct vanes reduce losses caused by eddies, reverse air flow and low pressure areas at the turn to a minimum. Elturns are prefabricated and are available in units of one piece construction in sizes up to and including 48 in. x 48 in.

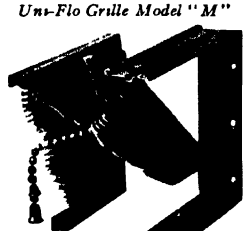
The Volocitrol has been designed to noiselessly provide positive and adjustable control of air volume, pressure and distribution across a supply outlet. They are available in two models, one for use with side wall and the other for overhead systems of air distribution.

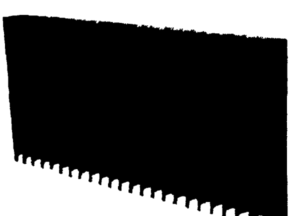

Venturi-Flo-Recessed Type

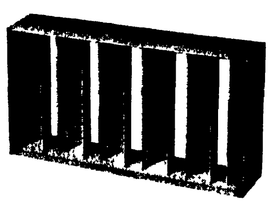

Venturi-Flo-Surface Type

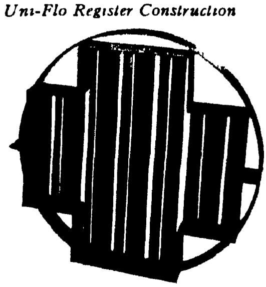

Venturi-Flo-Thermostatically Controlled Type


Uni-Flo Grille Model "M"


Uni-Flo Register Construction


Elturn


Volocitrol


Circular Volocitrol

SEE OUR COMPLETE CATALOG IN SWEETS

W. B. CONNOR ENGINEERING CORP.

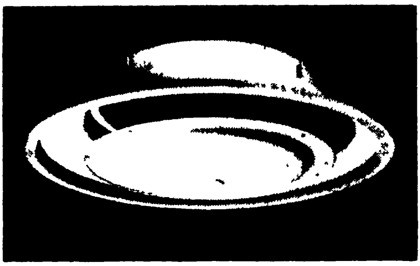
114 East 32nd Street, New York 16, N. Y.

Offices in All Principal Cities

Canadian Representative: D & D Air Conditioning Co., Montreal, Canada

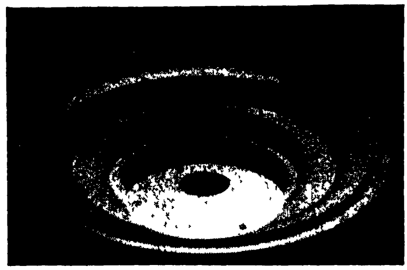
Manufacturers of KNO-DRAFT Adjustable Ceiling Air Diffusers

KNO-DRAFT Adjustable Air Diffusers insure efficient air distribution, maximum secondary air induction, noiseless and draftless diffusion, and uniform temperature throughout the occupied zone, *regardless* of season or ventilation requirements. Every KNO-DRAFT unit is easily and quickly adjustable for individual or seasonal requirements. The same KNO-DRAFT Diffuser is equally effective to expel chilled air parallel to the ceiling or eject heated air downward to prevent stratification.



(*Patents Pending*)

MODEL K KNO-DRAFT Diffuser—For Supply Air—attractive—light, yet sturdy—for high or low ceilings or attachment to exposed duct work. Anti-smudge rim prevents streaked ceilings Sizes 4 in. to 40 in neck diameter for capacities from 50 cfm to 20,000 cfm per unit



(*Patents Pending*)

MODEL SR KNO-DRAFT Diffuser–For Combination Supply and Return air to simplify duct work Sizes 6 in to 24 in supply-air neck diameter for supply capacities from 50 cfm to 9,000 cfm per unit, with central, return neck area 75 per cent of supply neck area

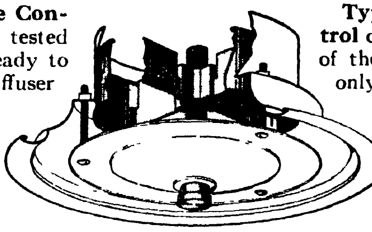
The KNO-DRAFT Diffuser will effectively distribute large volumes of air—premixing room and supply air It permits the use of higher duct velocities—resulting in smaller ducts and lower costs. Duct designs are simplified and fewer outlets are required. KNO-DRAFT Diffusers blend well with any architectural treatment. They are simple in construction, light in weight, and easily installed.

KNO-DRAFT Diffusers are geometrically proportional, size for size, insuring like resistance at like neck velocity for any size—and balanced static pressure throughout the system.

THE TYPE D AIR VOLUME CONTROL is designed for application exclusively to KNO-DRAFT ADJUSTABLE CEILING AIR DIFFUSERS. It is furnished already assembled within the diffuser and requires neither field assembly nor attachment to ducts, angle rings or other external appurtenances

Type D Air Volume Control is adjusted and tested before shipment and ready to function when the diffuser is installed

Its operation is entirely independent of the air direction adjustment which is part of all standard KNO-DRAFT Air Diffusers



Patented

Type D Air Volume Control complements the function of the air diffuser. It varies only the *quantity*, not the *characteristic* of the air distribution. With it, a series of diffusers may be balanced without affecting the air diffusion efficiency.

With KNO-DRAFT ADJUSTABLE CEILING AIR DIFFUSERS equipped with TYPE D AIR VOLUME CONTROL, the desired air pattern for any room or zone is AT YOUR FINGER TIP.

Hart & Cooley Manufacturing Co.

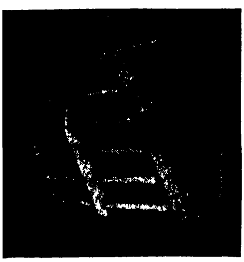
Established 1901

Air Conditioning Registers and Grilles - Warm Air Registers
Damper Regulators - Furnace Regulators - Pulleys - Chain

Holland, Mich.

NO. 75 DESIGN—FLEXIBLE FIN TYPE with TURNING BLADE VALVE to provide DOUBLE DEFLECTION. Also without Valve as Grille or Intake

CONTROL OF AIR FLOW IN TWO PLANES



Instant Adjustment of Air Flow
(Up, Straight or Down)

Is obtained by turning the regulator on the register face to the proper setting with a key furnished with each register. When the valve is opened, as shown at the left, the individual valve louvres automatically stop in position to provide the proper air flow—Up (Fig 1) for cooling systems to avoid drafts; Straight (Fig. 2) for ventilating systems; Down (Fig. 3) for heating systems to prevent stratification. When the valve is closed, as shown at the left below, it completely stops the flow of air.

Air Flow Can be Quickly Adjusted Sideways

No. 75 Design has a flexible fin-type face. Each fin may be twisted individually with a wrench furnished with each register or grille to provide any desired sideway deflection of the air flow.

Greatly Reduced Turbulence and Resistance

Figs. 1, 2, and 3 show the air flow with No. 75 Design; Fig. 4, with the conventional register. Compare the turbulence in the stackhead of the latter with the smooth flow obtained with No. 75 Design. So efficient is No. 75 Design that there is actually less resistance with this register, using a standard stackhead, than if no register at all were used.

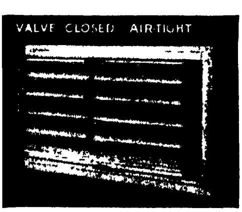


Fig. 1

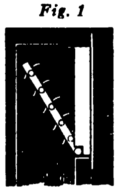


Fig. 2

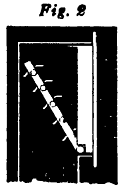


Fig 3

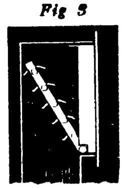
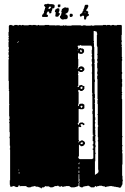
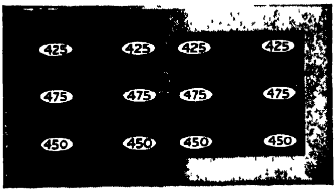
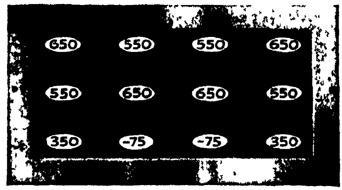


Fig. 4





Velocities with No. 75 Design



Velocities with Conventional Register

EVEN DISTRIBUTION OF AIR OVER ENTIRE FACE

The turning blade valve distributes the air evenly with a uniform velocity over the entire face, as shown in Figs. 1, 2, and 3 on the preceding page. Note how the air rushes through the upper part of the face with a conventional register, as shown in Fig. 4. Since the entire face of No 75 Design register is utilized for discharge of air, smaller and in some cases fewer registers can be used without causing excessive velocities.

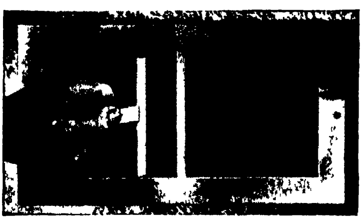
Prevention of Streaked Ceilings—With either UP, STRAIGHT, OR DOWN deflections the air does not strike the ceiling immediately in front of the register; streaked ceilings are thus avoided.

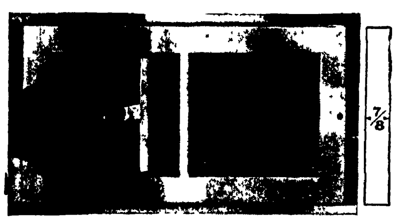
Excellent Concealment of Duct—The depth and close spacing of the vertical bars, combined with the valve, provide almost complete concealment of the duct, adding considerably to the pleasing appearance of the register face.

Special Settings—No. 75 Design functions equally well when located at the end of a horizontal duct or, by installing it upside down, when the air is delivered to it from above.

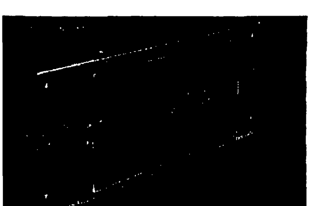
AVAILABLE IN FOUR TYPES

With Turning Blade Valve—No. 751 Register (Left) has Sponge Rubber Gasket and 3/16 in. turndown. No. 754 Register (Right) is similar except has 7/8 in. projection.





Without Valve—No. 750 Grille (Left) has Sponge Rubber Gasket and 3/16 in. turndown. No 757 Intake (Right) has 7/8 in. projection.



FOUR TYPES OF INSTALLATION FRAMES AVAILABLE

No. 75 Design items can be used with or without installation frames No. 3 Sidewall Stud Frame (illustrated), fastens directly to stud, forming a solid, streak-proof foundation for register. No. 8 Frame is similar for baseboard use. No. 5 Baseboard Stack Frame provides inexpensive, streak-proof installation. No. 2 Band Iron Frame provides for connecting register to stackhead.

CATALOG 42 showing the complete H & C line, available upon request.

Hendrick Manufacturing Company

Hendrick Perforated Metal Grilles

48 Dundaff Street, **Carbondale, Pa.**

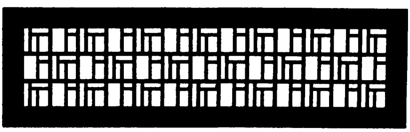
SALES OFFICES IN PRINCIPAL CITIES—CONSULT TELEPHONE DIRECTORY

PRODUCTS—Hendrick Perforated Metal Grilles; Mitco Open Steel Flooring; Mitco Armorgrids; Mitco Shur-Site Treads.

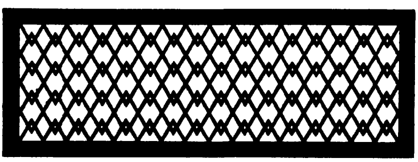
HENDRICK PERFORATED METAL GRILLES

To architects, engineers, contractors and others who buy or specify grilles, Hendrick offers literally hundreds of designs from which to select the pattern or patterns best suited to specific applications.

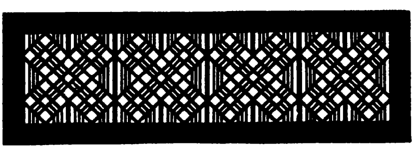
In addition to those popular designs which have been specified so consistently that they are today regarded as standard patterns, Hendrick offers a number of exclusive designs, many of them covered



Musak



M No. 9



La Crosse

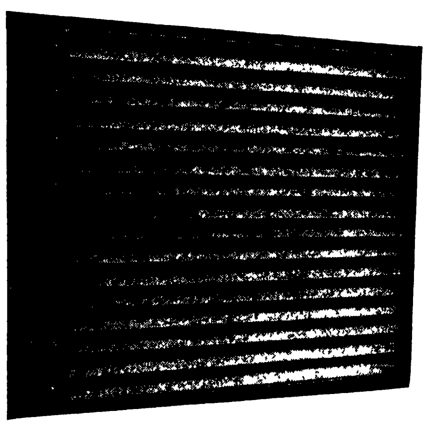
by design patents. Originally designed to meet specific requirements, these Hendrick patterns are available, without premium, to those who seek something that is distinctive as well as different.

All Hendrick Grilles are characterized by clean-cut perforations and fine finish. In addition, Hendrick grilles are put through a special flattening operation which insures easy installation.

HENDRICK FIXED LOUVRE GRILLE

One of the most popular grilles in the Hendrick line is a door grille, developed originally for hotels and hospitals but equally ideal for bathroom doors in residences.

Hendrick Fixed Louvre Grille is built up of a series of strips bent to a fixed angle and rigidly fastened into a band frame, a construction permitting free circulation of air but preventing vision through the grille from any angle. Easily installed in any door.



Hendrick Fixed Louvre Grille

Regularly furnished in No. 18 U. S. Gauge Steel; also obtainable in other commercially available metals.

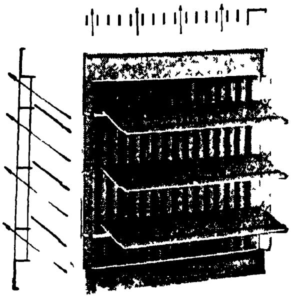
Write on your letterhead for a copy of 194-page handbook, "Hendrick Grilles."

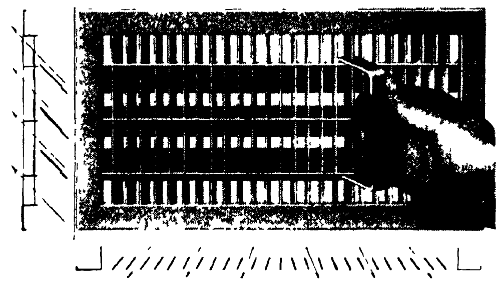
The Independent Register Co.

ESTABLISHED 1898

3747 East 93rd Street, **Cleveland, Ohio**

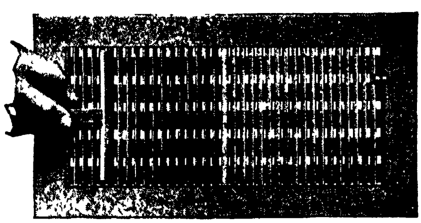
AIR CONDITIONING REGISTERS AND GRILLES



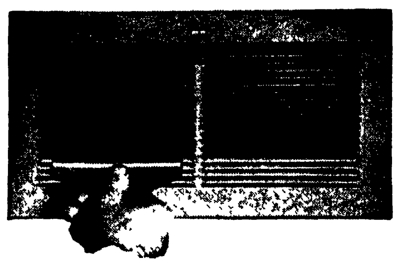


Rear View
Showing Adjustable Deflecting Vanes

No. 321A Grille with Deflecting Vanes—With vertical grille bars and horizontal deflecting vanes. The grille bars may be individually adjusted to direct air flows to right or left; and the vanes are made individually adjustable to deflect air flows up or down.



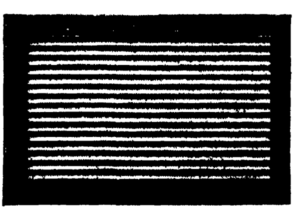
No. 238 Wrought Steel—4-way adjustable direction of air flow. Flexible vertical grille bars, multiple valves.

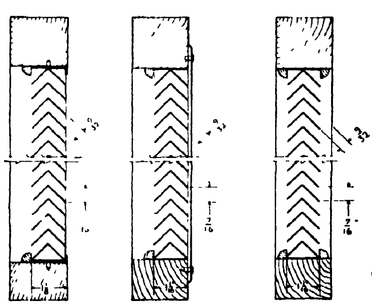


No. 139 Wrought Steel—Flexible horizontal grille bars, bendable for up, down or straight air flow. Single valves.

Independent No-Vision Grilles—No. 1312 for Doors, Walls and Partitions

The grille bars are "V" shaped; it is impossible to see through the grille from any viewpoint.





F R C

No. 1312F—One outer frame edge turned inward. See "F".

No. 1312R—With overlapping rim 5/8 in. wide, on all four sides.

No. 1312C—With grille core only, installed with moulding.

The Pyle-National Company

1334-58 North Kostner Avenue, **Chicago 51, Illinois**

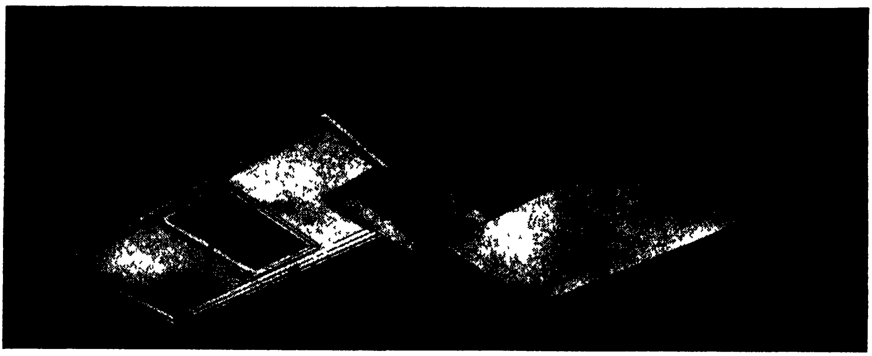
Representatives in Principal Cities

Manufacturers of

MULTI-VENT

Panels

for Heating • Ventilating • Air Conditioning



Control Plate *Valve* *Distribution Plate*

MULTI-VENT Panels are used for the even distribution of warm, cool or fresh air within room spaces The panels are designed to form the fourth side of a standard air distributing duct The control plate is equipped with an adjustable valve to balance the volume of air passing through each panel, and also to maintain a static head within the duct or plenum chamber. The perforated distribution plate forms the decorative exposed face of the panel. Due to its large surface and thousands of perforations, air is distributed over an extremely large area at very low velocity This accomplishes remarkably uniform distribution without excessive rate of air motion, at and below the breathing line. When used with the other correct auxiliaries Multi-Vent results in uniform temperatures and comfort.

Application data will be furnished on request

REGISTER & GRILLE MFG. CO.

Incorporated

70 Berry Street, **Brooklyn, N. Y.**

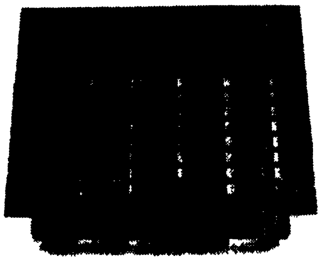
Headquarters for all types of Registers and Grilles

RESIDENTIAL AND COMMERCIAL

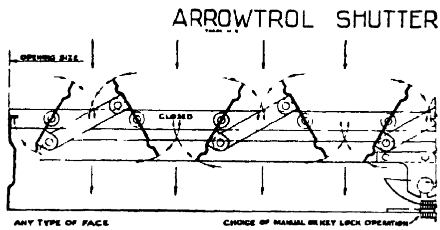
Register shutters of different types can be furnished with all types of Register Faces or Grilles.

All Register Shutters have our exclusive feature of brass collars inserted in the ends of the shutter to minimize rusting.

REGISTERS FOR VENTILATION

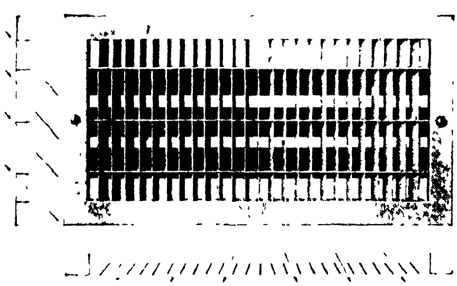


Style 3370 lock type Register allows directional flow of 135 deg either right and left or up and down. Will open 45 deg beyond 90 deg



The Arrowtrol, line cut shown above, gives straight throw in connection with volume control

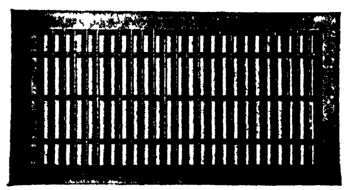
FOUR-WAY DEFLECTION TO AIR FLOWS



Style 3320 Grille and HMV deflecting vane

Front bars vertically adjustable, rear vanes horizontally adjustable; *or* Front bars horizontally adjustable, rear vanes vertically adjustable.

R & G ADJUSTABLE DIRECTED AIR FLOW TWO-WAY DEFLECTION



Use No. 3320 Grille for adjustable right and left deflection. Style 3310 has horizontal adjustable bars for up and down deflection.

THE "THIN MAN" REGISTER FOR RESIDENTIAL USE

Style 1108, shown, allows right and left deflection and up or down control at the back

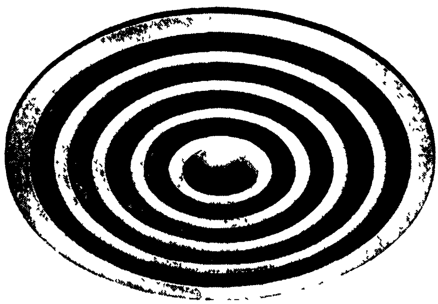
Other designs of faces are available.

Ask for our catalog which shows other types of air controls; also 81 different Stamped Metal designs and over 100 designs in Cast Metals—Iron, Brass or Bronze.

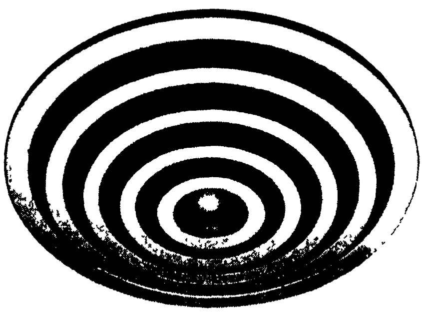
Tuttle & Bailey, Inc.

New Britain, Connecticut

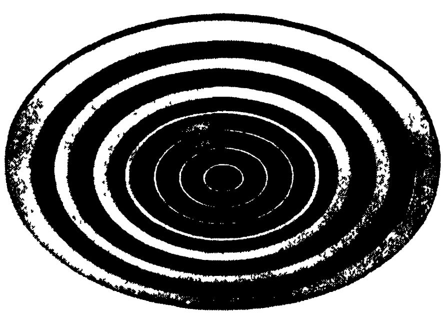
PRODUCTS: Ceiling Diffusers—Grilles—Registers—Intakes—Air Control Devices—Ornamental Grilles, Cast or Wrought Metals—Copper Convector Heaters



Type S1



Type E2



Type R1

AEROFUSE OUTLET

FOR HEATING

VENTILATING AND

AIR CONDITIONING

TYPE S

Flush-type diffuser for installation on ceiling. Perfect combination of real beauty and functional superiority. Provides (1) Maximum Air Mixture (2) Rapid Temperature Equalization (3) Perfect Air Distribution (4) Total Elimination of Drafts.

TYPE E

Type E Outlets are designed for installation on exposed ductwork and provide the same efficient performance as the S Type. The rings of Types E2 and E3 are stepped down, which greatly increases the capacity of a given size of outlet, resulting in an appreciable saving in the cost of the outlet and the ductwork.

TYPE R

Combination supply and return (or exhaust) unit. Designed particularly for use on installations where simplification of the duct layout is of primary importance since the return (exhaust) duct can be run to the same location as the supply duct.

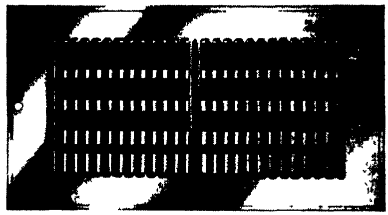
Send for complete engineering data and descriptive catalog.

Tuttle & Bailey, Inc.

New Britain, Connecticut

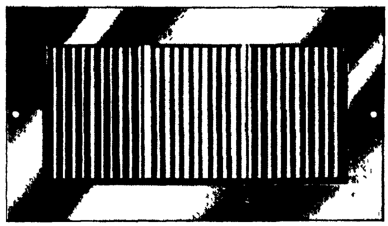
PLIAVANE ADJUSTIBLADE REGISTER

An inexpensive register suitable for war housing installations. The air flow may be directed sideways by the individually adjustable face vanes and up or down by the back blades which can be "set" from the face of the register itself.



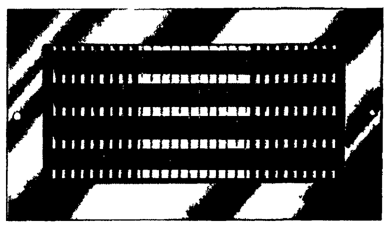
AIR CONDITIONING GRILLES

Furnished in both the fixed deflection (Airline Design) and the sectionally adjustable deflection (Flexair Design) types with bars running either horizontally or vertically. The Tuttle & Bailey Air Conditioning Grille is scientifically and sturdily constructed to perform efficiently under all operating conditions.



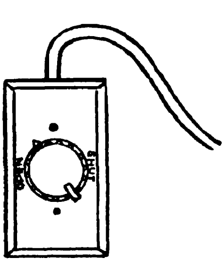
DOUBLE DEFLECTION GRILLES

Furnished with face bars of fixed (Airline) or adjustable (Flexair) deflection types and equipped with individually adjustable back blades. Also available with horizontal face bars and vertical back blades.



McKNIGHT VOLUME CONTROL REGISTER

Provides positive control of air volume at the outlet and eliminates necessity of duct dampers and diffusers. Scientifically designed louvres are easily and accurately adjusted by means of a special key furnished with each register.



REMOTE CONTROL

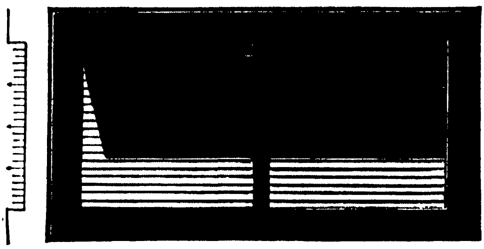
Ideal for hotels, office buildings, large manufacturing plants and public buildings. Provides an economical means of individual control of air volume at each outlet.

United States Register Company

General Offices: **Battle Creek, Mich., U.S.A.**

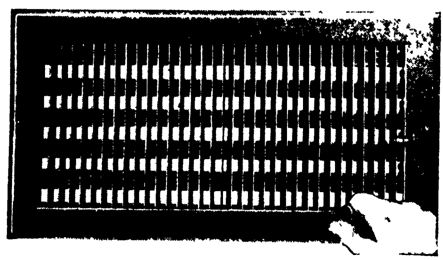
Branches: MINNEAPOLIS, MINN., KANSAS CITY, MO., ALBANY, N. Y., NEW YORK, N. Y., SAN FRANCISCO, CALIF.

Air Conditioning Registers, Vents and Grilles



Style 153LF—Louver-Type Air Conditioning Register-Bars ¼ in. deep—Spaced 4 openings to the inch affords Non-Vision. Can be supplied in Directional Flow in either Horizontal or Vertical Bar Styles. Can be furnished with all styles of Setting Frames.

Style 249LF—Duo-Deflection Air Conditioning Register. Gives complete Air Control. Vertical Front bars—Key-pin adjusted to provide 45 deg Right and Left or Two-way Side Flow. Lever operated Horizontal Back-valves give from Full Closed to any degree of Upflow and to 45 deg Down-flow FULL FACE COVERAGE. Can be supplied with any style of Setting Frame. Fits all Stack Heads of Standard Size Dimensions.

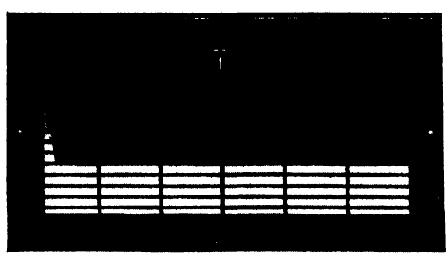


Style 256LF—Flex-bar Air Conditioning Register Vertical Front Bars set 22 deg Right and Left Side Flow Deflection attained by setting of Grille Bars with bending wrench to accommodate room condition. Back-valves give same Up and Down control of air flow as 249LF above. FULL FACE COVERAGE. Can be supplied with any style of Setting Frame. Fits all Stack Heads of Standard Size Dimensions

All of above Styles can be supplied with either Lever or Individually adjusted Multiple Valves or Louvers. *i.e.* 153VVI—Vertical Valves Individually adjusted 145VVL—Lever operated Vertical Valves.

Grilles and Vents in Matching designs are available.

For Complete Information Write for Latest Catalogs with Engineering Data.



Style 103LF—Horizontal Lattice Perforated Register for Forced Air Systems Not directional flow.

Complete Catalogs 41-G, 41-AC, and Catalog 41-F (Fittings) furnished on request.

Young Regulator Company

5209 Euclid Avenue, **Cleveland 3, Ohio**

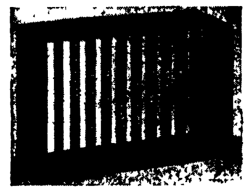
DAMPER REGULATORS; REMOTE CONTROL REGULATORS; DAMPERS

Representatives in Principal Cities

No 700

Young Remote Control Regulator No. 700

Individually controls volume of air to each room. A turn of the knob opens the damper, when turned beyond 180 deg, a spring closes damper Operates dampers at distances up to 250 ft or more Corner pulleys, with connection for tubing used at all bends.



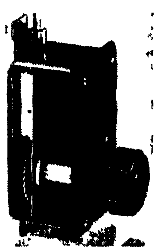
No 805

Young Multiple Blade Damper No. 805

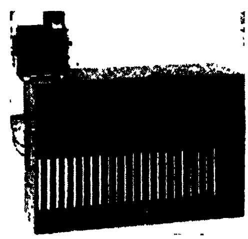
Used with Remote Control.

Young Remote Control No. 703

No 703, cutaway view, has a deep box which extends through the plaster.



No 703



No 815

Young Relief Damper No. 815

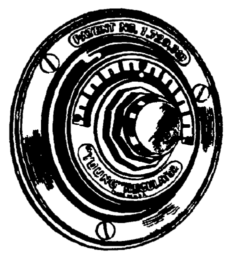
Automatically controlled, with modulating low voltage motor and remote bulb thermostat When the room is cooled to the desired temperature, the upper damper closes and the lower damper opens and diverts some or all of the air into the relief duct

Young Fresh Air Damper No. 820

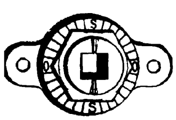
Is for outside air It is similar to No 815 except that it is a single instead of double damper

Young Surface Damper Regulator No. 1

Withstands vibration and shock Locks securely positive in action — tamper-proof Is adjusted and locked with a polygonal wrench, or key



No 1



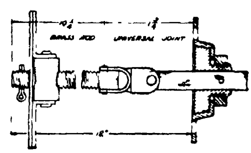
No 401

Young Valcalox Damper Regulator No. 401

Ideal for round duct work Has detachable wrench for adjusting and locking Much cheaper than No 1.

Young Air-Split Damper Regulator No. 900

Can be placed in almost any location accessible for operating splitter dampers Operating crank furnished with unit



No 900

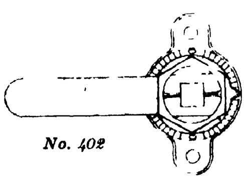
No 301

Young Concealed Regulator No. 301

Flush with plaster when installed. Adjusted and locked with special wrench.

Young Flush Cup Regulator No. 201

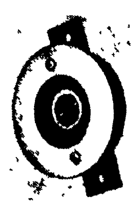
Similar to Concealed Regulator No. 301 except more easily accessible for adjustments



No. 402

Young Valcalox Regulator No. 402

Similar to Model No. 401. Has attached handle Used where it is necessary to continually change setting of damper.



No 201

The American Rolling Mill Company

Executive Offices, Middletown, Ohio

ATLANTA 3, GEORGIA 1437 Citizens & Southern National Bank Bldg.
BERKELEY 2, CALIFORNIA . . Seventh and Parker Streets
BOSTON 10, MASSACHUSETTS ... 201 Devonshire Street
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CHATTANOOGA 2, TENN ... 602 Chattanooga Bank Bldg.
CHICAGO 4, ILLINOIS ... 310 S Michigan Bldg
CINCINNATI 8, OHIO ... 24 Cooper Bldg , Hyde Park
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Choose the Correct ARMCO Grade

There is a grade of ARMCO sheet metal particularly suited to every air conditioning application For detailed information get in touch with the nearest district office or write direct to The American Rolling Mill Company, Middletown, Ohio.

ARMCO Ingot Iron
(Galvanized)

Ducts
Washer Chambers
Plenum Chambers
Steam Line Casings
Furnace Casings
Spray Towers
Drip Pans
Housings
Machine Guards
Unit Conditioners
Roof Ventilators
Eliminator Blades

ARMCO PAINTGRIP
(Galvanized)

A special mill-Bonderized galvanized sheet that can be painted without pretreatment. Preserves life and beauty of paint.

ARMCO Cold Rolled PAINTGRIP: A zinc-flashed sheet with a special mill-Bonderized surface.

Hot Rolled
(Sheets and Strip)

Fan Blades
Blower Casings
Fuel Oil Tanks
Unit Conditioners
Stoker Hoppers

ARMCO ZINCGRIP

A special zinc-coated sheet that can be severely formed without peeling or flaking of the tightly adherent zinc coating. Also available with a PAINTGRIP finish.

Cold Rolled
(Sheets and Strip)

Furnace Casings
Room Unit Casings

Plates
(ARMCO Ingot Iron)

Smoke Stacks
Coal Hoppers
Breeching
Unfired Pressure Vessels
Low-fired Boilers
Tanks

High Strength Steel

A low alloy, high tensile steel possessing great strength. Used with proper design it results in weight reduction of framework, tanks and similar items. Under atmospheric service conditions it has corrosion resistance superior to regular steel

Stainless Steel
(Sheets, Strip and Plate)

Combustion Chambers
Heat Flues and Tubes
Humidifier Pans
Pre-heaters
Furnace Parts and Supports
Fan and Blower Blades

Special grades have excellent resistance to destructive heat-scaling up to 2000° F

ARMCO ALUMINIZED Steel

An aluminum-coated sheet steel with exceptional resistance to heat and corrosion. Resists heat discoloration up to approximately 900° F., and will withstand oxidation at even higher temperatures. Also offers exceptional heat reflectivity where this is desirable

BETHLEHEM STEEL COMPANY

GENERAL OFFICES: BETHLEHEM, PA.

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Akron	Columbus	New Haven	San Francisco
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Cincinnati	Los Angeles	Salt Lake City	Wilkes-Barre
Cleveland	Milwaukee	San Antonio	York

Export Distributors

Bethlehem Steel Export Corporation, New York

BETHLEHEM PRODUCTS FOR HEATING, VENTILATING AND AIR CONDITIONING

Steel Sheets—Bethlehem manufactures a line of sheet steel to handle all types of heating, ventilating and air-conditioning jobs. Sheets are produced in the general classifications of hot-rolled (black); cold-rolled, and galvanized. Hot-rolled sheets are made in thicknesses from No. 18 gage to No. 2 gage, in widths varying with the thicknesses. Cold-rolled sheets are made in widths over 12 in. and in gages No. 11 to 30 inclusive. Galvanized sheets are made in gages 8 to 31 inclusive in widths of 24, 30, 36, 42 and 48 inches.

Especially well-suited for heating and ventilating ductwork are rust-resisting Beth-Cu-Loy copper-bearing sheets. Thorough going tests have proved that the 0.20 to 0 30 per cent of copper in these sheets will double or nearly triple their rust resistance. Thus they rank high in this respect among commercial grades of iron and steel sheets. Galvanized sheets of Beth-Cu-Loy form and seam readily, and the extra cost is only four to five per cent

Steel Pipe—Bethlehem now produces a strong butt-welded pipe, known as Beth-Co-Weld, in sizes from ½-inch diameter to 3-inch, inclusive, and in uniform 21-foot lengths, plus or minus one inch. This pipe is uniform in size, length, and physical characteristics. It will do a consistently good job wherever used.

Ammonoduct, a steel pipe which can be bent cold without annealing or danger of fracture, has long been a Bethlehem specialty. Unusually soft and ductile because it is made of a special quality open-hearth steel, Ammonoduct is easy to fabricate. Made in sizes from ½-inch diameter to 3-inch, inclusive, standard weight and extra strong, and in long lengths. In peacetime it was used extensively for ammonia piping, heater coils, water legs in furnaces and similar uses where pipe must be bent. Ammonoduct is ideal for radiant-heating installations. Will be available in quantity after the war emergency is over.

If you have a contract for heating, ventilating or air-conditioning, use dependable Bethlehem materials.

Steel from this giant mill will eventually become Bethlehem sheets.

Here's how the rolls of our modern mill form Beth-Co-Weld Pipe.

Jones & Laughlin Steel Corporation

Jones & Laughlin Building, **Pittsburgh 30, Pa.**

WELDED and SEAMLESS STEEL TUBULAR PRODUCTS

J & L Welded Pipe

Jones & Laughlin manufactures Standard Weight, Extra Strong, and Double Extra Strong Welded Pipe, Black and Galvanized, for steam, gas, air, water, refrigeration and sprinkler work. Sizes: 1/8 in. to 10 in. inclusive.

Jones & Laughlin Steel Pipe is made of soft, weldable steel rolled from solid ingots made to a special analysis. The steel pipe produced is soft and ductile, free cutting, strong at the welds, and free from excess scale. J & L Pipe is commercially straight and free from blisters, cracks or other injurious defects.

Careful attention is given the threading of the pipe with good clean-cut threads fitted with sound couplings correctly tapped to give a tight joint. Soft, ductile steel of free cutting quality enables the contractor to cut clean, sound threads on the job.

The Jones & Laughlin process of galvanizing assures a thorough coating and insures against pipe being clogged with spelter. The galvanized coating adheres strongly and does not tend to flake off.

J & L Seamless Pipe

J & L Seamless Pipe is made in three weights; standard and extra strong in sizes 1/2 in. nominal to 14 in. O. D. inclusive, and double extra strong in sizes 2 in. to 8 in. inclusive.

J & L Seamless Steel Pipe is pierced from a solid billet—there are no welds. The result is dependable and uniform wall strength. The method of manufacture, and the use of only specially selected steel, assure exceptional ductility, a quality that is essential to successful coiling and bending, and flanging for Van Stone joints.

J & L Seamless Pipe can be used with full satisfaction in either threaded joint or completely welded installations.

Ductility, strength and safety—make this product especially adaptable for air, steam, gas and gasoline lines, boilers, refineries, dry kilns, refrigerating systems and other exacting applications.

J & L Hot Rolled Seamless Steel Boiler Tubes

J & L Seamless Boiler Tubes are manufactured in accordance with the *A.S.M E* Boiler Code and comply with the *A.S.T M* Specifications and the rules and regulations of the Bureau of Marine Inspection and Navigation of the U. S. Department of Commerce. They are supplied in a full range of standard sizes, from 1 1/4 in. O D to 6 in O.D. inclusive.

The process by which Jones & Laughlin manufactures seamless boiler tubes is largely responsible for the unusually high ductility of the product. It is a process in which a forging action is predominant, and produces a desirable combination of strength with a highly ductile nature J & L tubes therefore are installed with ease and safety.

Other J & L Tubular Products

J & L also manufactures Reamed and Drifted Pipe in sizes 1 in. to 12 in. inclusive, Dry Kiln Pipe, Pipe for Refrigeration Service, Water Well and Irrigation Casing, Line Pipe and a complete line of Oil Country Tubular Products in welded and seamless.

Handbook on Standard Pipe

Available without charge or obligation, to heating and piping contractors and heating and ventilating engineers, is the J & L tubular products handbook SP5. In addition to containing catalog information of J & L Standard Pipe this book includes helpful engineering design data for contractors and engineers. Write on your letterhead for a copy of this convenient handbook.

United States Steel Corporation Subsidiaries

Carnegie-Illinois Steel Corporation, Pittsburgh and Chicago

Columbia Steel Company, San Francisco

Tennessee Coal, Iron & Railroad Company, Birmingham

United States Steel Export Company, New York

District Offices in all Principal Cities

U·S·S COPPER STEEL

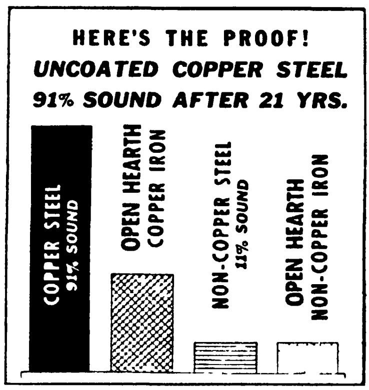
For Superior Rust Resistance at Low Cost

Corrosion resistance and cost are two determining factors of the type of metal to be used for various air conditioning jobs.

Copper Steel has 2 to 3 times the atmospheric corrosion resistance of plain steel or pure iron as shown in the results of unbiased tests made at Pittsburgh, Ft. Sheridan and Annapolis by the American Society for Testing Metals.

The cost of U·S·S Copper Steel is less than that of pure iron or copper-bearing pure iron and only slightly more than plain steel. Thus there often is a dividend of 200 per cent to 300 per cent longer life and a saving in the first cost as well.

When galvanized, U·S·S Copper Steel produces a sheet that is rust resistant all the way through—not just on the surface. It should be used for all ducts carrying humidified air or placed in damp locations such as basements, shower rooms, etc.



Corrosion test of A S T M on 22 gage black sheets exposed at Annapolis, Md., October, 1916. The copper steel sheets outlasted all others in the test

U·S·S PAINTBOND

U·S·S **Paintbond** should be used whenever galvanized steel is to be painted. This special Bonderized sheet can be painted immediately, offers a much better surface for painting, lessens danger of the paint flaking and retards corrosion. It is used for ductwork, furnace housings and outdoor metal work.

Send for our Paintbond booklet.

U·S·S DUL-KOTE

U·S·S **Dul-Kote** is a specially treated non-spangled galvanized sheet which also can be painted immediately without ageing or otherwise preparing the surface. It is available in the South and in the West.

Send for our Dul-Kote booklet.

OTHER U·S·S PRODUCTS INCLUDE:

Black Sheets—All grades, hot rolled, cold rolled in a number of different finishes.

Stainless—Heat resisting steel for various uses where temperatures are high and corrosion severe.

Cor-Ten—High strength steel—greater strength, greater atmospheric corrosion resistance for smokestacks, hoods, etc.

For more information on U·S·S Galvanized, Copper, Black and Stainless Sheets, send for our Guide for Sheet Metal Workers.

The American Brass Company

General Offices: Waterbury, Conn.

Offices and Agencies in Principal Cities

ANACONDA
from mine to consumer

IN CANADA ANACONDA AMERICAN BRASS LIMITED, New Toronto, Ontario

PRODUCTS—Anaconda Deoxidized Copper Tubes and Fittings; Anaconda "85" Red-Brass Pipe; Everdur Metal for storage heaters, storage tanks, ducts and air conditioning equipment

ANACONDA COPPER TUBES AND FITTINGS

For Heating, Plumbing and Air Conditioning

Anaconda Deoxidized Copper Water Tubes assembled with Anaconda Fittings offer an unusual combination of advantages in hot water heating systems at a cost only slightly higher than black iron and approximately the same as wrought iron pipe. These advantages may briefly be summarized as follows:

Low Friction Loss—Because the inside surfaces of copper tubes are inherently smoother than those of pipe and tubes made of ferrous materials and also because they do not become roughened by the formation of rust, these tubes offer a lower resistance to flow. In addition, the long radius turns of Anaconda Elbows and the smooth inside surface of Anaconda Wrought Copper Fittings further reduce friction losses.

These factors naturally increase the efficiency of the system, particularly when it includes a forced pressure circulator.

Ease of Installation—In many places the flexibility of copper tubes simplifies connections that ordinarily would be awkward and expensive to make with rigid pipe and threaded fittings. Anaconda Solder Fittings are compact. They can be installed in restricted space where the use of a wrench would be impossible.

Architects and builders naturally object to large holes and notches cut in the framing members of a building for the passage of piping. Anaconda Copper Tubes can be installed with a minimum of cutting in the structure—although holes should be large enough to permit movement of tubes due to expansion and contraction.

Appearance—Anaconda Deoxidized Copper Water Tubes assembled with Anaconda Solder Fittings present an attractive appearance. It is a frequent practice to clean the tubes after they are installed and apply a coat of clear lacquer or similar substance. This keeps the tubes bright and makes an installation of which both plumber and owner can be proud.

Temper and Thicknesses—Anaconda Copper Tubes are made in both hard and soft temper and in standard wall thicknesses.

They meet the requirements for these types of tubes in Federal Specification WW-T-799a and A.S.T.M. Specification B88-41. Type K, the heaviest, is recommended for heating lines and general piping.

Accuracy of Dimensions—Anaconda Deoxidized Copper Water Tubes are all finished to the close size tolerances required by the A.S.T.M. and Federal Specifications, which have been found essential for efficient assembly with solder fittings.

Permanent Identification—For permanent identification, the name "Anaconda" and the letter designating the type of tube is stamped in the metal at intervals of approximately 18 in., throughout every coil or straight length of tube.

The American Brass Company

Anaconda Copper Tubes, in all standard sizes, up to and including 1¼ in. are furnished soft in 30, 45 and 60-ft coils; also hard and soft in 20-ft straight lengths. Sizes over 1¼ in. are furnished, hard or soft, in straight lengths only.

ANACONDA "85" RED BRASS PIPE

Anaconda "85" Red Brass Pipe, in standard pipe sizes, is considered the highest quality corrosion-resistant pipe commercially obtainable at a moderate price and is recommended for steam return lines.

Anaconda "85" Red Brass Pipe contains 85 per cent copper and conforms to Government specifications for Grade "A" water pipe. The mark "Anaconda 85" is stamped in the metal at one-foot intervals throughout each length.

EVERDUR*

Everdur Metal is the original copper-silicon alloy It is manufactured by The American Brass Company in five standard compositions and in practically all commercial forms.

This high strength engineering metal is resistant to a wide range of corroding agents. Because of a versatile combination of useful properties, Everdur has become standard as a material for equipment in many fields of engineering and industry

In addition to their non-rusting properties and high strength, Everdur alloys possess many qualities not usually found in metals of this character. They are unusually resistant to general atmospheric conditions and other normally corrosive factors. Everdur alloys have excellent machining and working characteristics and can be fabricated into a variety of forms and shapes. Everdur alloys are available for oxy-acetylene or carbon arc welding.

CORROSION RESISTANCE

The corrosion resistance of Everdur is equal to that of pure copper and in some cases, slightly superior.

*"Everdur" is a trademark of The American Brass Company registered at the U. S. Patent Office.

However, like copper and all copper alloys, Everdur is not equally resistant to all corroding agents, nor to the same corroding agents under all conditions. As with copper, the resistance to corrosion may be substantially reduced in some instances by the presence of oxidizing agents. Nevertheless, Everdur does offer excellent resistance to the corrosive action of many solutions and atmospheres.

Everdur Tanks—Everdur copper-silicon alloy is an ideal material for durable, rustless water tanks of every description—from domestic range boilers to large storage heaters for hotels, laundries, hospitals, textile plants, schools or breweries.

Everdur is made in all commercial shapes including annealed tank plates which have physical properties as given in A.S.T.M. Specification B96-42.

Minimum specification requirements for hot rolled-and-annealed tank plates are: Tensile Strength, 50,000 psi.; Yield Strength (at 0.5 per cent elongation under load) 18,000 psi.; Elongation, 40 per cent in 2 inches.

Welds made with annealed Everdur tank plates meet the requirements for Class B and C vessels in the A.S.M.E. Code for Unfired Pressure Vessels.

For additional data and names of fabricators address our nearest office or agency.

EVERDUR FOR AIR CONDITIONING EQUIPMENT

Because of its strength and welding properties, Everdur may be substituted for steel and fabricated by substantially the same methods and with the same equipment as steel.

Everdur metal has been used with marked success for fans and blowers, ducts, humidifiers, cast and wrought parts of other equipment items subject to corrosive influences.

EVERDUR LITERATURE

Descriptive literature containing much pertinent tabular data will be sent upon request.

Mueller Brass Co.

Port Huron, Mich.

Branch Offices and Representatives in Principal Cities

Albany, N. Y.
Atlanta, Ga.
Boston, Mass.
Chicago, Ill.
Cincinnati, Ohio
Cleveland, Ohio
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Dallas, Texas
Detroit, Mich.
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Seattle, Wash.
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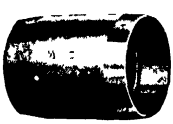
Canadian Sales and Manufacturer

Canada Wire and Cable Co., Ltd., Toronto, Canada

Mexico

George F. Gilfrin, Edificio "La Nacional," Mexico, D. F.

PRODUCTS—STREAMLINE Copper Pipe and Seamless Tubes; STREAMLINE Hard Copper Pipe and Solder Fittings; Valves, Flared and STREAMLINE Solder Fittings for Mechanical Refrigeration; Forgings of Brass, Bronze and Copper; Castings of Brass and Bronze; Rod; Screw Machine Products; Fabricated Parts and Special Nickel and Chromium Plated Parts; Machined Formed Tubes.



Coupling Copper to Copper

Copper to Outside I P S.

45 Deg Elbow

Streamline Copper Pipe and Fittings for heating, plumbing, air conditioning and industrial use are made by the Streamline Pipe and Fittings Division, Mueller Brass Co., Port Huron, Mich.

The **Streamline** Solder Fitting is the original solder type fitting, introduced and manufactured by the Mueller Brass Co. of Port Huron, Mich. It incorporates many advantageous features and has proved to be the revolutionary advance of the age in the development of piping systems for plumbing and heating and for many industrial uses

The **Streamline** Solder Fitting is not connected either by threading or flaring, but by soldering. The outside surface of the copper pipe and the inner surface of the **Streamline** fitting are cleaned with sandcloth, and solder flux is then applied to the cleaned surfaces to eliminate oxidation when the assembled joint is heated The joint is then sufficiently heated with a blow or acetylene torch and the soldering operation is performed by feeding wire or stick solder through the feed hole in the fitting

The **Streamline** Solder Fitting alone has the solder feed hole, groove and taper The solder feed hole, through which the solder is introduced, enters directly into an internal feed channel The feed channel is located equidistantly between the internal shoulder against which the pipe rests and the outer edge of the fitting When solder is introduced it is distributed by capillarity from the feed channel and distributed evenly and thoroughly between the bonding surfaces, traveling inward to the shoulder and outward to the edge of the fitting where it appears as a continuous solder ring around the full circumference of the pipe. This ring, and feed hole completely filled with solder, constitute positive proof to the operator that the joint is permanently leak-proof. An actual pressure test is not necessary.

The tapered ends, since they are the thinner sections of the fitting, hasten the cooling of the solder at these points and facilitate the completion of the joint.

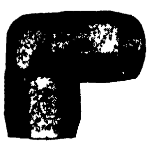
Streamline Pipe and Fittings Division

MUELLER BRASS CO.

Port Huron, Mich.

Patents 1770852; 1776502

Tee



90 Deg Elbow

Tee
Copper to Inside I P S

Crosses

The solder may be fed from any position, whether the feed hole is located at the top, side or bottom Owing to the never failing phenomena of capillarity, the solder will flow up, down or laterally with equal facility.

Streamline Copper Pipe is a seamless cold drawn copper tubing conforming to *A.S.T.M* B 88-33. For most piping purposes, hard drawn pipe is used though annealed material is supplied where bends are to be made Three weights of **Streamline** Copper Pipe, Govt. Types K, L and M, are made in all sizes, and an additional lighter weight is made in sizes 3 in and larger. The latter is used mainly by the paper industry for pressures not over 125 lb

This range of weights permits its use for water or air pressures up to 400 lb.

Streamline Solder Fittings are furnished in sizes from ¼ in. to 12 in inclusive with a full range of reducing sizes.

Fittings above 6 in. are flanged and may be had with either *A.S.A.* or rivetted pipe standard flanges. Mating flanges are soldered to the pipe.

The **Streamline** Solder Fitting permits the use of thin-walled copper pipe and places a non-rusting, non-clogging piping system within the reach of the ordinary investor. Vibration is not localized at the joints, but is harmlessly dissipated throughout the system. (Copper Pipe has the property of transferring the heated element (steam or hot water) from the point of generation (boiler) to the radiators quickly and with slight temperature drop.)

During the last ten years architects and engineers have used **Streamline** Copper Pipe and Fittings successfully in every type of building construction and in thousands of installations throughout the United States and Canada.

In addition to its rust and vibration-proof qualities and long life, **Streamline** has many other advantages such as the reduction in size of pipe lines and radiator connections from those nominally used, a neat, compact installation requiring a minimum of space and important advantages in industrial and drainage applications. There is a **Streamline** product for every piping requirement.

STREAMLINE fittings and copper pipe are now being installed in victory ships, subchasers, submarines, etc. Write Mueller Brass Co., Port Huron, Michigan, for complete information.

Revere Copper and Brass Incorporated

Executive Office: 230 Park Avenue, **New York 17, N. Y.**

MILLS—Baltimore, Md., New Bedford, Mass., Rome, N. Y., Detroit, Mich., Chicago, Ill.

SALES OFFICES—Boston, Mass., Providence, R I., Philadelphia, Pa., Atlanta, Ga., New York, N Y., Pittsburgh, Pa., Cleveland, Ohio, Cincinnati, Ohio, Grand Rapids, Mich., Milwaukee, Wis., St. Louis, Mo , Indianapolis, Ind., Minneapolis, Minn., Dallas, Texas, Seattle, Wash., San Francisco, Calif., Los Angeles, Calif., Hartford, Conn., Dayton, Ohio, Houston, Texas

REVERE COPPER TUBE

Revere Copper Tube is produced by the most modern equipment under close inspection and meets Federal and *A.S.T.M.* specifications. It is cold drawn, seamless, deoxidized, and possesses a gunbarrel finish on the inside as well as the outside. A wide range of standard sizes made in hard and soft tempers is normally stocked by distributors.

Types K, L, and **M** are made in the sizes, tempers, and lengths shown below. **Type K** is a heavy wall tube commonly used for underground water service lines, for plumbing and heating, oil lines, and some air conditioning applications. **Type L** is a medium wall tube furnished in hard or soft temper, and **Type M** is a light wall tube furnished in hard temper only. The latter is not recommended for underground service or for compression fittings. The actual outside diameter of these three types of tube is ⅛ in greater than the nominal size in each case.

Revere Dryseal Copper Tube is a dependable dehydrated utility tube furnished only in soft temper coils having both ends sealed. There are nine sizes available from ⅛ in. to ¾ in. inclusive. In each case the nominal size and the actual outside diameter are the same and the wall thickness is 0 035 in. throughout.

Revere Dryseal Copper Tube is commonly used for air conditioning and heat control lines, refrigeration and oil burner installations, and compressed air lines It can be bent by hand and easily flared for compression fittings

Fittings—Standard solder type fittings in cast or wrought patterns are available for use with **Types K, L,** and **M** copper tube. Various types of compression fittings are made for use with **Types K** and **L** tube and **Revere Dryseal Tube.** Connections made by the use of these fittings are proof against leaks so often caused by stresses or vibration. Joints of this kind are quickly made without any thread cutting and are generally stronger than the tube itself.

Advantages of Revere Copper Tube—The many advantages of **Revere Copper Tube** of particular interest to design engineers can be summarized briefly as follows:

1. A piping system comprising **Revere Copper Tube** and soldered fittings can be installed in a minimum of space as no allowance need be made for swinging wrenches.

2. Soft temper copper tube can be bent easily for offsets and changes in direction.

3. There need be no fear of rust accumulation on the inside of **Revere Copper Tube.**

4. It is particularly suitable for forced circulation hot water heating systems. The very smooth interior of **Revere Copper Tube** insures high velocities with a minimum of frictional resistance.

To Engineers and Manufacturers of Heating and Air Conditioning Equipment—Revere representatives will gladly assist and cooperate with engineers and suppliers of heating and air conditioning equipment at any time regarding matters involving applications of non-ferrous products

Revere Copper Tube

STANDARD DIMENSIONS AND WEIGHTS

Size In In.	Type K			Type L		Type M	
	O D. in In.	Wall Thickness In.	Wt. Lb per Ft	Wall Thickness In.	Wt. Lb per Ft	Wall Thickness In.	Wt. Lb per Ft
¼	.375	.032	.134	.030	.126	.025	.107
⅜	.500	.049	.269	.035	198	.025	.145
½	.625	.049	.344	.040	.285	.028	.204
⅝	.750	.049	.418	.042	.362	.030	.263
¾	.875	.065	.641	.045	.455	.032	.328
1	1.125	.065	.839	.050	.655	.035	.465
1¼	1.375	.065	1 04	.055	.884	.042	.682
1½	1.625	072	1.36	.060	1.14	.049	.940
2	2 125	.083	2 06	070	1 75	0 58	1.46
2½	2 625	.095	2 93	.080	2 48	0.65	2 03
3	3.125	.109	4.00	.090	3 33	.072	2.68
3½	3.625	.120	5.12	100	4.29	083	3 58
4	4 125	.134	6 51	110	5 38	.095	4 66
5	5.125	.160	9 67	.125	7 61	.109	6 66
6	6.125	.192	13 9	.140	10.2	.122	8.92
8	8 125	271	25.9	200	19.3	.170	16.5
10	10 125	338	40 3	250	30 1	.212	25.6
12	12 125	405	57.8	.280	40 4	.254	36.7

Recommended Operating Pressures

Type K—Hard Temper	up to 400 lb
Type K—Soft Temper	up to 250 lb
Type L—Hard Temper	up to 250 lb
Type L—Soft Temper	up to 150 lb
Type M—Hard Temper	up to 250 lb

Tempers and Lengths

Type K } Hard and Soft Temper in straight 20 ft lengths
Type L } Soft Temper in 30 ft, 45 ft, 60 ft and 100 ft coils
Type M—Hard Temper in straight 20 ft lengths.

Wolverine Tube Division

Calumet & Hecla Consolidated Copper Company

1411 Central Avenue, **Detroit 9, Michigan**

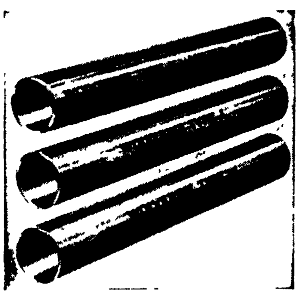
SEAMLESS TUBE

COPPER - BRASS - ALUMINUM

Sales Offices:

ATLANTA, GA	P O Box 38—Northside Station	NEWARK, N J	727 Mosswood Ave.
BOSTON, MASS	12 Ashburton Place	NEW YORK, N Y	60 East 42nd St.
BUFFALO, N Y	416 Jackson Bldg	PHILADELPHIA, PA	Commercial Trust Bldg—Room 1208 15th and Market Sts.
CHICAGO, ILL	3348 S Pulaski St.	PITTSBURGH, PA	1214 Liverpool St.
CLEVELAND, OHIO	1740 East 12th St.	PORTLAND, ORE	416 Northwest 14th St.
DAYTON, OHIO	P O Box 1041	ST LOUIS, MO	4565 McRae Ave.
HOUSTON, TEXAS	P O Box 259—Petroleum Bldg	WASHINGTON, D C	808 Investment Bldg—15th and K St.
LOS ANGELES, CALIF	1015 East 16th St		
MILWAUKEE, WIS	647 W Virginia St		
MINNEAPOLIS, MINN	529 S Seventh St		

COPPER WATER TUBE



TYPE K—Recommended for Air Conditioning, Refrigeration, Oil Burner, and Plumbing and Heating installations.

TYPE L—For Oil Burner, Air Conditioning, Refrigeration and general plumbing uses.

TYPE M—Suitable for Air Conditioning and Refrigeration installations and for interior plumbing and heating purposes.

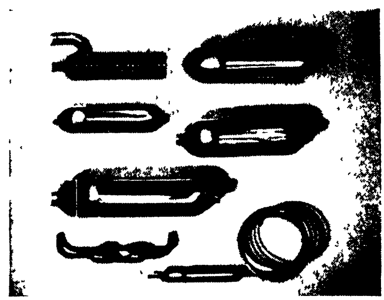
Types K and L furnished in hard or soft temper; Type M, hard only.

Wolverine Water Tube is made according to U. S. Government and A.S.T.M. specifications. For a complete list of these data, write Detroit for Form 575.

REFRIGERATION TUBE

Wolverine refrigeration tube has long been the standard of the industry. Dehydrated, sealed, paper-wrapped; uniform soft temper, and moisture content well below minimum specified by A.S.R.E. Available from stock in standard coils.

ACCUMULATOR SHELLS

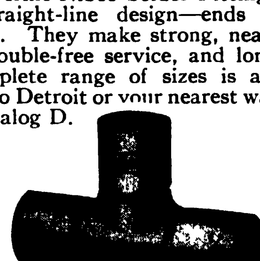


A new accumulator shell developed by Wolverine and produced to customers' specifications in a variety of shapes and sizes up to 3½ in. diameter.

It combines many advantages including one-piece construction and is especially adaptable to refrigeration problems. Send for Catalog E-1 describing other spun-end tubular parts.

WROUGHT FITTINGS

Wolverine-Nibco Solder Fittings are of the straight-line design—ends not expanded. They make strong, neat joints; give trouble-free service, and longer life. A complete range of sizes is available. Write to Detroit or your nearest warehouse for Catalog D.



The experience of 27 years of seamless tube manufacture, the use of the latest equipment, and adherence to Government and customer specifications, are responsible for the uniform, high quality of Wolverine products. And now, backed by the 76-year experience and large resources of Calumet & Hecla, Wolverine quality is controlled from ore to finished product.

E. B. Badger & Sons Co.

DESIGNERS • ENGINEERS • CONSTRUCTORS • MANUFACTURERS

NEW YORK 18, N. Y.	BOSTON, MASS.	PHILADELPHIA, PA.
500 Fifth Avenue	75 Pitts Street	Northeast Corner Broad and Arch Streets

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PRODUCTS AND SERVICES

BADGER "PACKLESS" CORRUGATED EXPANSION JOINTS

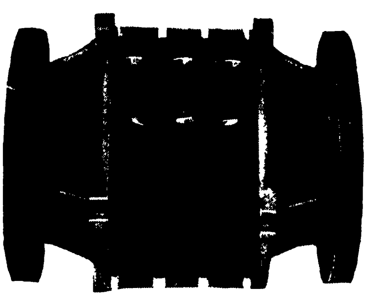
Engineers and Manufacturers of Chemical, Petro-Chemical and Petroleum Refining Equipment; Process Engineers, Designers and Constructors of Complete Plants.

BADGER "PACKLESS" CORRUGATED EXPANSION JOINTS

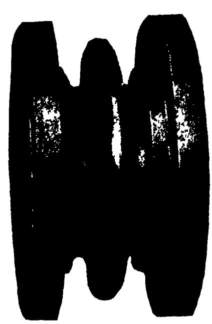
Protect equipment; eliminate costly shut-downs; prevent loss of materials flowing through pipe lines. The Badger "Packless" Corrugated type of expansion joint has been used for many years to relieve stresses, absorb vibrations between connected equipment, and compensate for changes in pipe lines due to temperature differentials

BACKED BY FIFTY YEARS OF EXPERIENCE

Badger engineers pioneered the corrugated "packless" joint and brought it to its present high state of dependability and usefulness. To meet the increasingly severe conditions in industry, the more recent developments are included in the following brief descriptions



Standard Flanged-End Badger Directed Flexing, Self-Equalizing "Packless" Corrugated Expansion Joint. *Also made with standard welding end. (Double units available) Bulletin No 100.*



Badger Non-Equalizing "Packless" Corrugated Expansion Joint, Single Corrugation. *Also made in multiple corrugation and special shapes. Bulletin No. 200.*

FLANGED AND WELDING ENDS. Badger "Packless" Corrugated Expansion Joints can be furnished with either flanged or welding ends.

TELESCOPING SLEEVES. Badger "Packless" Corrugated Expansion Joints can be furnished with telescoping sleeves if required.

INSTALLATION, OPERATING, MAINTENANCE ECONOMIES

• ***"Packless"***—Since this type of expansion joint is made from a *single tube,* no packing is required—hence no servicing. This is particularly advantageous in underground installations, because the expense of manholes and tunnels can be eliminated.

• ***Flexible***—This type of joint is readily flexed, thus reducing to a minimum the thrust on adjacent equipment or fittings.

• ***Wide Range of Traverses***—By varying the number of corrugations, the Badger "Packless" Corrugated Expansion Joint can be made to take care of traverses ranging from small fractions of an inch up to any practical limit.

• ***Wide Range of Pressures***—Standard joints are constructed for normal operating pressures; special joints for higher pressures.

• ***Compactness***—Outside diameter of Badger "Packless" Corrugated Expansion Joint is about that of a flanged fitting.

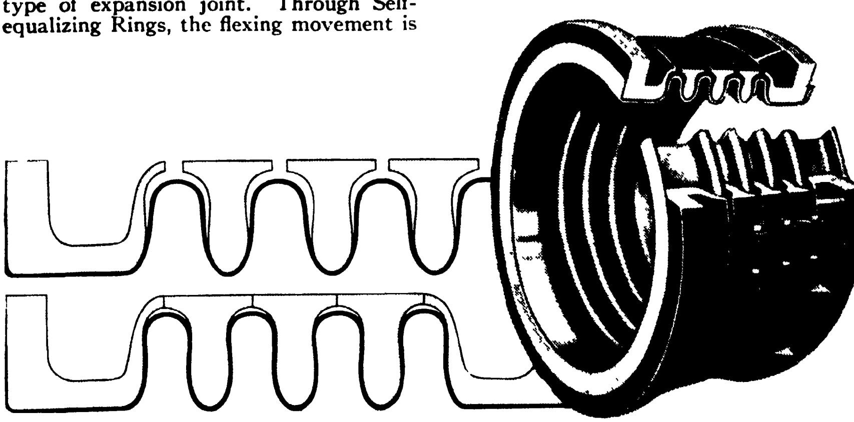
• ***Ease of Installation***—The Badger "Packless" Corrugated Expansion Joint is as easily installed as any fitting.

COPPER AND STAINLESS STEEL. Badger "Packless" Corrugated Expansion Joints are made in self-equalizing or non-equalizing types; and are made of copper, stainless steel, or other metals to meet varying conditions with respect to temperature, pressure and corrosion.

HEAT TREATMENT. Every Badger "Packless" Corrugated Expansion Joint is scientifically heat-treated during the process of manufacture to remove forming stresses. This Badger process increases resistance to flexing stresses, lengthening the life of joint.

DIRECTED FLEXING—SELF-EQUALIZING

This feature, *exclusive* with Badger "Packless" Corrugated Expansion Joints, greatly lengthens the life of the corrugated type of expansion joint. Through Self-equalizing Rings, the flexing movement is not only limited but is progressively controlled throughout the movement. Flexing stresses are kept distributed.



The Dow Chemical Company

Midland **Michigan**

NEW YORK • BOSTON • PHILADELPHIA • WASHINGTON • CLEVELAND • DETROIT • CHICAGO
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Plastics Division

SARAN PIPE, TUBING AND FITTINGS

FOR INSTALLATIONS REQUIRING HIGH CHEMICAL RESISTANCE

These thermoplastic products made from Dow raw materials are recommended for a wide variety of applications—especially those jobs requiring high chemical resistance. (See table below.) Additional engineering data and literature are available from the following manufacturing and sales licensees—or any Dow sales office

Tubing and fittings are available from: Allied Plastics Co., 5225 Wilsbie Blvd , Los Angeles, California; Extruded Plastics, Inc., Norwalk, Conn ; The Parker Appliance Co , 17325 Euclid Ave , Cleveland, Ohio; Skuttle Mfg Co , 507 E Larned St., Detroit, Michigan.

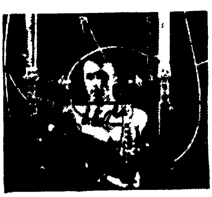
Pipe and fittings are available from: American Hard Rubber Co., 11 Mercer St., New York, New York.

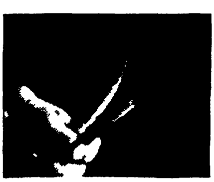
Tubing, pipe, and fittings for both are available from: Commercial Plastics Co., 201 N. Wells St , Chicago 32, Illinois; Haveg Corporation, Newark, Delaware; Hodgman Rubber Co , Framingham, Mass ; Elmer E Mills Corp., 153 W. Huron St., Chicago, Illinois, St Louis Plastic Molding Co , 4605 Olive St., St. Louis, Missouri; Western Felt Works, Inc., 4115 Ogden Ave., Chicago, Illinois; Yardley Plastics Co , 138 Parsons Ave , Columbus, Ohio; Dominion Rubber Company of Canada, Montreal, Quebec

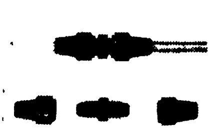
CHEMICAL RESISTANCE OF SARAN AT ROOM TEMPERATURE

Reagent	Stability Rating	Reagent	Stability Rating	Reagent	Stability Rating
98% (Conc) H_2SO_4	Fair	50% HaOH	Good	Linseed Oil	Excellent
60% H_2SO_4	Excellent	10% HaOH	Good	Bromine Water	Not Recommended
30% H_2SO_4	Excellent	28% NH_4OH	Not Recommended	Chlorine Water	Not Recommended
10% H_2SO_4	Excellent	10% NH_4OH	Not Recommended	"Chlorax" Bleaching	
35% (Conc) HCl	Excellent	Ethyl Alcohol	Excellent	Solution	Excellent
10% HCl	Excellent	Carbon Tetrachloride	Good	10% Duponol	Excellent
65% (Conc) HNO_3	Good	Di Ethyl Ether	Good	10% Zinc Hydro-	
10% HNO_3	Excellent	*Benzene	Not Recommended	sulfite	Excellent
Glacial Acetic	Excellent	Ethyl Gasoline	Excellent	15% Ca Cl_2	Excellent
10% Acetic	Excellent	Turpentine	Excellent	15% $FESO_4$	Excellent
5% H_2SO_3	Excellent	Triethanolamine	Excellent	Water	Excellent
Conc Oleic	Excellent	Lubricating Oil	Excellent	Air	Excellent

It should be noted that chemical resistance decreases with rise in temperature
***Shrinks slightly upon immersion in these solvents.**







TUBING

Saran tubing accommodates both high and low pressures, withstands freezing, and is heat resistant up to 170° F. Its unusual chemical resistance gives it wide use in installations where acids and corrosive chemicals must be handled.

Saran tubing is readily flared with a special flaring tool, assuring a tight joint in special Saran fittings. Tees, elbows, and straight joints (illustrated) are available in sizes to match tubing (up to ¾ in. O. D.)

PIPE

Chemical resistance makes Saran pipe invaluable for many industrial installations. It is only ¼ the weight of comparable iron pipe. Available in sizes up to 4 in. O D.

A complete weld is formed in one minute by placing on heated welding plate until molten and pressing ends together.

Easily threaded with modified standard equipment. Flanges, couplings, ells and plugs are available Further fittings being developed.

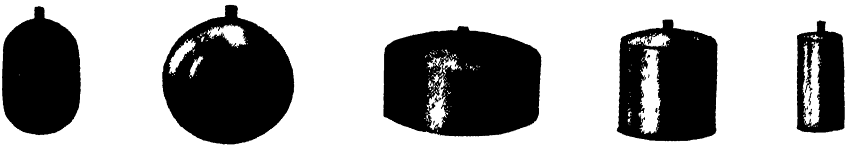
Arthur Harris & Co.

210-218 N. Aberdeen Street — **Chicago 7, Ill.**

ENGINEERS — FABRICATORS OF NON-FERROUS METALS AND STAINLESS STEEL

Metals Fabricated—Aluminum, Block Tin, Brass, Bronze, Copper, Everdur, Monel, Nickel, Inconel, Stainless Steel and KA2 SMO. Bulletin on request.

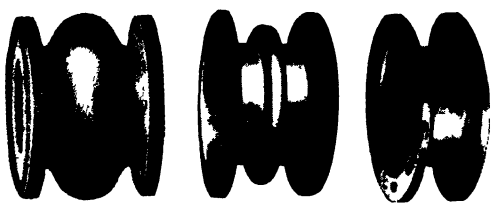
Metal Floats



Column — *Ball* — *Flat Cylindrical* — *Cylindrical* — *Cylindrical*

Made of copper, plain steel, stainless steel, KA2 SMO, aluminum, brass, Monel, pure nickel, Admiralty and Everdur, for open tank and all pressures.

Seamless copper ball floats carried in stock in diameters of 3 in , 4 in , 5 in., 6 in., 7 in., 8 in , 10 in , 12 in. for open tank and pressures of 25, 50, 100 and 150 lb. Floats in special sizes and pressures—made to order Stainless steel ball floats 2½ in to 12 in. for high pressure and corrosion carried in stock—special stainless steel floats made to order—stainless steel ball floats larger than 12 in. diameter can be made up specially. Float catalog sent on request



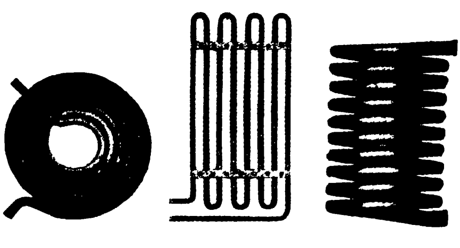
B-280 Convex — *B-290 Convex* — *B-281 Concave*

Copper Expansion Joints

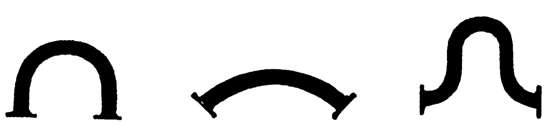
For low pressure and vacuum. Made in two styles—convex and concave. Sizes 4 in. to 60 in. diameter. Cast iron or steel flanges. Flanges drilled to American standard unless otherwise ordered: B-290 available only in sizes 4 in. to 15 in. inclusive.

Coils

For heating, cooling and condensing. All shapes made from any size pipe or tube—standard or special connections, of copper, brass, aluminum, stainless steel, KA2 SMO, monel, inconel, nickel, block tin, and Everdur



Bends



We make bends in every shape from all sizes of copper water tube, pipe and tubing in copper, brass, aluminum, stainless steel, monel, tin and nickel. Standard or special connections. U-bends for storage water heaters.

Also special pipe work for industrial installations, plumbing, heating and brewing. Perforated pipe, double pipe coolers, etc.

Non-Ferrous Castings—"Dairywhite" nickel silver for Process Industries Equipment. Suitable for milk and food products machinery Castings also of 88-10-2, 80-10-10, 85-5-5-5, silicon bronze and manganese bronze, and special mixtures Many patterns available without charge.

GRINNELL COMPANY INC

Heating, Industrial and Power Plant Piping, Fittings, Hangers, Valves, Pipe Bending, Welding, Piping Supplies, Etc.

Executive Offices: **Providence 1, R. I.**

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PRODUCTS AND SERVICES—

Complete Service on materials to Specification on Power Plant Piping, Industrial Piping, and Industrial Heating Systems; Prefabricated Piping including Pipe Cutting and Threading, Pipe Bends, Welded Headers, Welded and Welding Fittings, Lap Joints and the Grinnell line of products for Super Power.

Grinnell Equiflo Valves for forced hot water heating systems; Grinnell Adjustable Pipe Hangers and Supports; Grinnell Cast Iron and Malleable Iron Pipe Fittings; Grinnell Malleable Iron Unions; Grinnell Welding Fittings; Grinnell Thermoliers (Unit Heaters); Thermoflex Traps and Heating Specialties.

Also Humidifying Systems; Piping for acids and other special materials.

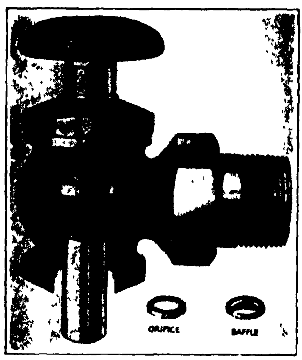
Malleable Iron, Brass, Bronze and other Castings; Brass, Cast Iron, Wrought Iron and Steel Pipe; Seamless Steel Tubing in Iron Pipe Sizes.

Valves: Check, Diaphragm, Globe, Pressure Reducing and Regulating, Quick Opening, Safety and Y.

Automatic Sprinkler Systems; Stand Pipes; Underground Supply Mains; Hydrants; Fire Pumps; Pressure and Gravity Tanks.

Grinnell Equiflo Valves

For Forced Hot Water Heating



Equiflo Valve

The designing of forced circulation hot water heating systems is so simplified by the Grinnell Equiflo Valve that they can be laid out and installed as easily as vapor or steam systems. This valve consists of a regular type packless radiator valve with a cartridge or tube made up of a series of orifices and baffles capable of setting up any required frictional resistance This method of establishing any desired resistance does away with elaborate calculation of pipe sizes. Grinnell guarantees perfectly balanced circulation to each and every radiator where these valves are installed throughout the system.

Equiflo Data Book sent to interested parties.

For Data On Thermoflex Traps and Heating Specialties, see page 1104

GRINNELL ADJUSTABLE PIPE HANGERS AND SUPPORTS

One of the chief advantages of Grinnell Adjustable Hangers is that they permit adjustment of pipe lines after installation, thus obviating the necessity of turnbuckles or the removal of hangers. Their time and trouble-saving qualities during installation are equally exceptional. Below are shown a few Grinnell Hangers and Supports of particular interest to heating engineers. Send for Hanger Catalogue showing complete line.

Adjustable Swivel Rings (Patented)

Fig No 101
Solid Ring

These Malleable Iron Adjustable Swivel Rings can be used with Coach Screw Rod or Machine Threaded Rod in connection with practically any type of Ceiling Flange, Expansion Case, Insert, etc.

Adjustment of at least 1½ in. is secured by turning Swivel Shank. Swivel Shank automatically locks, preventing loosening due to vibration in the pipe line.

The Split Ring permits adjustment either before or after Ring is closed. A wedge type pin is loosely but inseparably cast into the hinged section for fastening this section after pipe is in place.

Fig No 104
Split Ring

Adjustable Swivel Pipe Rolls (Patented)

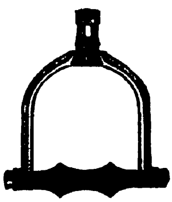


Fig No 174
Swivel Pipe Roll

An adjustable type of pipe roll using a single hanger rod. Swivel Shank allows vertical adjustment and automatically locks, preventing loosening from vibration.

CB-Universal Concrete Inserts
(Patented)

Made of malleable iron, in one body size, to take a special removable nut, tapped for ⅜ in., ½ in, ⅝ in., ¾ in. or ⅞ in. rod as required. Nuts automatically lock by means of V-type teeth on both insert and nuts.

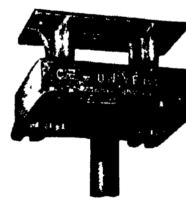
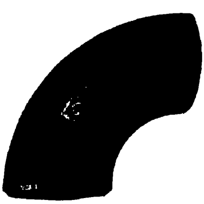


Fig No 282
CB-Universal Insert

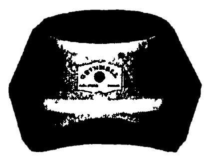
GRINNELL WELDING FITTINGS



90° Elbow, Long Turn

Grinnell Welding Fittings are made from Seamless Steel Pipe or tubing and possess the same physical characteristics as standard, extra strong and o.d. steel pipe or seamless steel pipe of comparable size. They can be used under the same conditions, pressures and temperatures as the pipe itself.

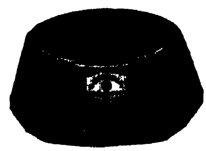
All Grinnell Welding Fittings have welding faces for all plain circumferential butt welds scarfed or beveled as follows: For wall thicknesses 3/16 to ¾ inch inclusive, 37½ deg. ± 2½ deg, straight bevel. Angles of bevel other than 37½ deg. can be furnished on special order.



Welding Outlet

Welding Tee

Lap-Joint Stub End with Lap Flange attached



Threaded Outlet

Tube Turns
INCORPORATED

General Offices and Factory: **Louisville 1, Ky.**

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New York, 150 Broadway—Rector 2-7844
Philadelphia, Broad St., Station Bldg —Rittenhouse 0722
Pittsburgh, 3001 Grant Bldg.—Atlantic 8848
San Francisco, 2611 Russ Bldg.—Garfield 2594
Seattle, 422 Smith Tower Bldg.—Main 2778
Washington, D. C., 1026 17th St., N. W.—National 0860

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FIRST SEAMLESS WELDING FITTINGS 1ST AND STILL THE BEST

TUBE-TURN *Welding Fittings and Flanges*

Tube Turns Offers a Line of Seamless, Uniform-Wall, Forged Fittings and Flanges for Pipe Welding Including Elbows, Returns, Tees, Reducers, Caps, Nipples and Laterals in All Sizes and Weights.

TUBE TURNS is the **original** manufacturer of seamless welding fittings and the pioneers also in their development and use. A few of the many advancements of Tube-Turn fittings are listed here Complete details and engineering data are included in the **Tube-Turn Catalog No. 111,** sent on request.

Outstanding Advantages of Tube-Turns

Many sound features which **Tube Turns originated** make them universally preferred today. (1) Seamless welding elbows and returns with **uniform wall thickness and true circularity,** making possible exact alignment for better, faster, lower-cost welding. (2) Permanently leakproof, trouble-free construction assuring minimum maintenance cost. (3) Stronger, lighter, tighter piping systems made possible. (4) Full effective radius of elbows permits more efficient flow and reduces pressure loss. (5) Smooth inner walls minimize corrosion and lengthen life. (6) Proper grain structure with no internal stress provides greater strength.

Tube-Turn Materials and Construction

Regular stock **Tube-Turn** elbows and returns, in both long and short radius types, are made in Standard, Extra Heavy Schedule 160 and Double Extra Heavy pipe thicknesses, conforming to A. S T. M. specifications A-234. **Tube-Turn** welding fittings are likewise available conforming to U. S. Navy specifications 45-F-11. Also obtainable in special thicknesses, carbon-molybdenum steels, copper, brass, toncan iron, nickel, monel and aluminum.

Tube-Turn elbows, returns and tees are forged by exclusive processes, giving uniform high quality In addition the designs of these products provide many special superiorities.

Highest Grade Forged Steel Flanges

Highest Grade Forged Steel Flanges—of outstanding strength and dimensional accuracy—are manufactured by **Tube Turns.** A complete line—welding neck, slip-on, lap joint, threaded, blind, and slip-on orifice.

Distributor Stocks of Tube-Turn welding fittings are available from distributors located in every important city and area in the United States and Canada.

Catalog No. 111, 240 pages, is an engineering data book and encyclopedia of welding fitting, flange and piping information. Conveniently indexed. Write for free copy.

Alco Valve Company

ENGINEERED REFRIGERANT CONTROLS

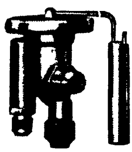
851 Kingsland Ave., **St. Louis 5, Mo.**

NEW YORK OFFICE: 381 Fourth Ave.

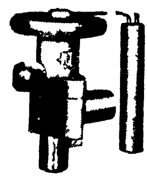
A Complete Line of Engineered Refrigerant Controls

THERMO EXPANSION VALVES

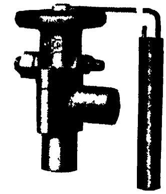
For automatic control of liquid refrigerant on all types of air conditioning and refrigeration systems.



Type TK



Type TJL



Type THL

CAPACITIES—From fractional tonnage to 100 tons Methyl Chloride, 50 tons Freon-12.

SOLENOID VALVES

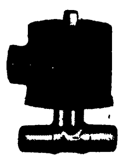
For all types of service

Magnetic Liquid Stop Valves

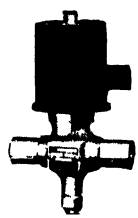
Freon—up to 75 tons, Methyl Chloride—up to 150 tons

Magnetic Suction Stop Valves

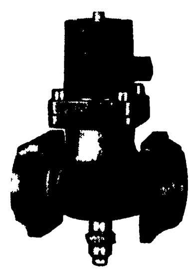
Freon- up to $1\frac{1}{2}''$ or 8.8 tons Methyl Chloride up to $1\frac{1}{2}''$ or 17 tons



Type S1



Type M8



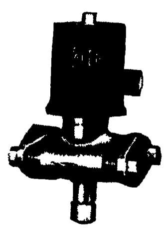
Type R2

AMMONIA CONTROLS

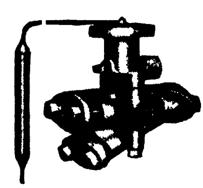
Magnetic Liquid Stop Valves—up to 172 tons.

Magnetic Suction Stop Valves—up to $1\frac{1}{2}''$ or 28 tons.

Thermo Expansion Valves—from fractional tonnage to 60 tons.



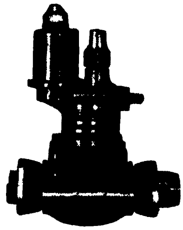
Type M6



Type TGS

EVAPORATOR PRESSURE REGULATORS

For Freon, Methyl Chloride and Ammonia, with port sizes up to 2 in., and a wide variety of connection sizes.



Also complete line of Float Valves, Float Switches, Constant Pressure Expansion Valves and Strainers.

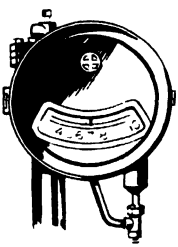
American Meter Company

INCORPORATED

General Offices: 60 East 42nd St., **New York 17, N. Y.**

ALBANY BALTIMORE BIRMINGHAM BOSTON CHICAGO ERIE KANSAS CITY
LOS ANGELES PHILADELPHIA PITTSBURGH SAN FRANCISCO
DALLAS, HOUSTON, TULSA—Westcott & Greis, Inc. ALHAMBRA, CALIF—Reliance Regulator Corp
HAMILTON, ONT—Canadian Meter Co., Ltd.

Measurement and Control of Gas, Oil, Steam, Air and Liquids



FLOWMETERS

Indicating . Recording . . . Integrating

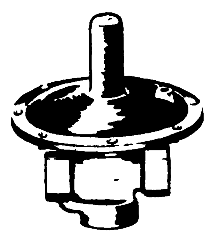
The **American Indicating Flowmeter** is an orifice type meter which *indicates* the rate of flow of air, gas, oil, steam or water through a line in which an orifice is installed. The **Integrating Flowmeter** *charts* a totalized reading of the quantity of gas, air, or steam at constant pressure—or for liquids, regardless of the rate.

American Flowmeters known for their simplicity of design—assure trouble-free operation and ease of adjustment

Write for Bulletin E-3

RELIANCE REGULATOR

Type A

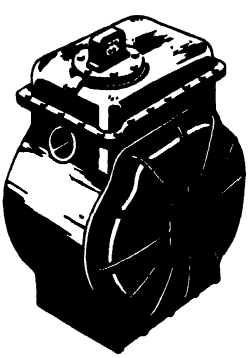


Reliance *Type A* regulators are designed to give absolute control of low pressures and are used extensively for controlling pressure to air and gas appliances requiring positive control of operating pressures

A complete line of Reliance regulators are notable for exacting duty and minimum pressure loss at required capacity, as well as ease of servicing and low cost of upkeep

Automatic shut-off valves are available for jobs that require a safety feature, so that when inlet pressures fall too low, the flow of gas through the regulators is stopped

Write for Bulletin 43.



The Compact

IRONCASE GAS METER

Space requirements of these meters are cut down to the least possible dimensions without sacrifice of **American Ironcase** advantages

Ironcase positive displacement meters set up a standard of unsurpassed accuracy in the measurement of gas by making use of the approved slide valve and full bellows principle . . . then by adding to this principle an extraordinary ruggedness of construction.

The body is cast in one piece. No joints and gaskets between internal parts . . . and no possibility of internal leaks. Special safeguards against friction loss are found throughout each meter.

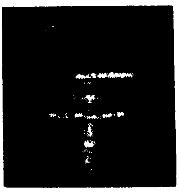
Low pressure capacities up to 10,000 cu ft at 2 in. differential.

Write for Bulletin EG-40.

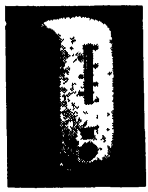
AUTOMATIC PRODUCTS COMPANY

2450 NORTH THIRTY — SECOND STREET
MILWAUKEE WISCONSIN

A-P DEPENDABLE CONTROLS
For Heating, Refrigeration and Air Conditioning

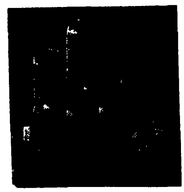


• A-P Thermostatic Expansion Valves. Several models and sizes, for capacities up to 16 tons Freon or 32 tons Methyl Chloride

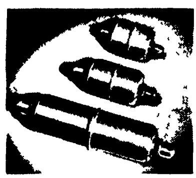


A-P Thermostats.
For Cooling or Heating

A-P Solenoid • Operated Water Valve. Made especially for Deep Well Cooling.



• A-P Solenoid Refrigerant Valves. Capacities up to 50 tons Freon.

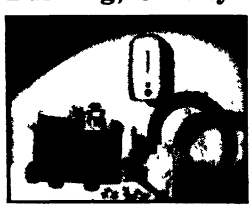


A-P "Trap-Dri."
Combined Filter-Strainer-Drier
Traps dirt, scale, moisture in refrigeration systems

A-P Water • Regulating Valves. Capacities up to 1440 Gallons per hour.

A-P Controls for Oil Burning, Gravity-Feed Heating Plants.

A-P Constant Level Oil Control Valve— With Fuel Compensator. Used on Gravity Oil Burning Appliances.



A-P Complete Furnace Control Set— Made in variety of types for Gravity-Feed Oil Burning Furnaces.

A-P Fuel Oil "Trap-It"— Traps dirt and water in fuel systems. Improves operation of all oil burning devices.

A-P Valve DEPENDABILITY

is widely recognized in Refrigeration, Air Conditioning and Heating. This reputation is born of close adherence to a rigid standard of perfection—in materials used, careful testing and inspection, simplicity of construction, and many unusual features.

Detroit Lubricator Company

Division of American Radiator & Standard Sanitary Corporation

Detroit 8, Michigan, U. S. A.

New York 18, N. Y., 40 West 40th Street

Chicago 5, Ill., 816 S. Michigan Avenue Los Angeles 13, Calif., 320 Crocker Street

Canadian Representative:

Railway and Engineering Specialties Limited, Montreal, Toronto, Winnipeg

Air and Vent Valves

No. 861

The "DL" line of air and vent valves is one of the broadest on the market. The Ideal Fast Venting System, using the No. 300 Multiport and the No. 861 Hurivent (illustrated) is particularly advantageous on one pipe steam jobs fired automatically and makes possible complete venting of the system in the shortest time, together with even heat distribution to all radiators. The No. 300 valve is limited to systems operating at less than 3 pounds pressure. The No. 5000 Airid Variport is an adjustable valve for systems which must operate at more than 3 pounds pressure. The No. 841 Ideal Quick Vent is an inexpensive fast venting main vent. For hand-fired coal jobs operating on vacuum, there is the No. 510 Vac-Airid Air Valve; the No. 862 Vac-Hurivent, and the No. 842 Vac-Vent for mains. Write for Bulletin No. 197.

Radiator Valves and Balancing Fittings

The broad line of "DL" radiator valves, both packed and packless, covers many types and sizes. All valves are sturdily built and have clean-cut threads and accurate dimensions. Hot water valves feature the patented "swinging plate" design, which eliminates restriction to flow and always turns freely.

A line of specially designed valves and balancing fittings for forced circulation hot water systems is also available. With "DL" balancing elbows or straightway fittings, the heat output from each radiator may be controlled by a screw driver adjustment without the necessity of draining the system. "DL" special circulator type valves are designed to close off tight enough to prevent heating of radiators when so desired, but allow a small leakage to avoid freezing. Write for Bulletins 56 and 85.

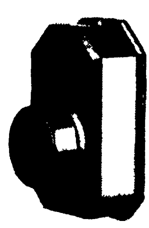
No. 133

Automatic Controls

No. 411

Detroit Lubricator Company manufactures a very complete line of electrical controls, designed to open or close an electrical circuit with changes of temperature or pressure. The No. 411 Thermostat (illustrated) is a low voltage model and is made in plain and Day and Night types. All No. 411 Thermostats are available with heat compensation to provide smooth, accurate temperature control. The No. 855 Mercoid Room Thermostat (illustrated) is a line voltage type—available in heating, air conditioning and refrigeration ranges.

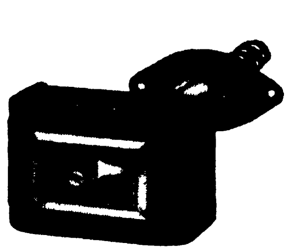
For industrial use the No. 250 and No. 450 line of pressure and temperature controls is available, in pressure ranges from 20 in vac. to 350 lbs, and in temperature ranges from −30°F. to 495°F.



No. 855

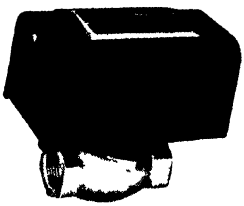
Also available is a full line of blower controls, combination blower and limit controls, such as the CA-815 illustrated, and a special line of water-tight equipment for use in wet locations.

There is a "DL" control available for practically every application where a dependable device is required to open or close an electrical circuit with changes of pressure or temperature. Write for complete information. Our Engineering Department is always ready to make recommendations on any specific problem.



No. CA815

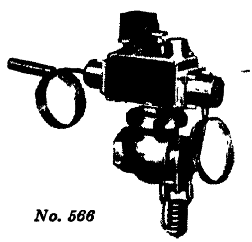
Gas Valves



No V-570

The No. V-570 Electric Gas Valve is an electrically operated valve for control of gas lines from ½ to 1¼ in. It provides partial opening upon initial operation, permitting quiet ignition of gas. Inexpensive, compact, and attractive in appearance, it employs an easily serviceable bimetal strip motor for actuating force. Write for Bulletin 201.

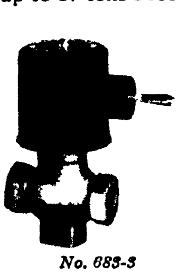
The No. 566 Valve combines in one compact unit all of the functions of a control system for gas heating. All safety functions are mechanical and operate independent of electric current. Adjustable for full snap or any degree of throttling action. Limit control closes valve with a snap action. Valve may be manually operated in case of current failure. Applicable to all gases, natural, manufactured, or mixed. Write for Bulletin No. 80



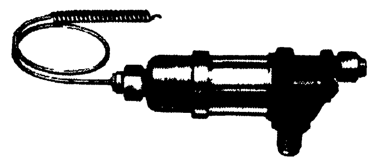
No. 566

Solenoid Valves

"DL" Solenoid Valves for control of water, air, oil, gas, or refrigerant, embody many desirable features. They are free from A C. hum and will open against high pressures. Available in all standard voltages and cycles A.C. or D.C. No. 683-3 (illustrated) is a small size valve with ⅜ in. connections. No. 681 is a pilot operated, intermediate size valve, and the No. 686 is a large pilot operated valve with capacity up to 17 tons Freon. No. 686 valve available with flanged connections. No. 681 and 686 are furnished with manual opening feature to permit opening in case of current failure. All models may be taken apart and cleaned in the field without removing from pipe lines. Write for Bulletin No. 199.



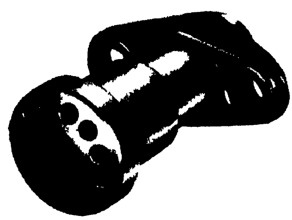
No. 683-3



No. 673

Expansion Valves

"DL" Thermostatic Expansion Valves are designed to keep the evaporator in a refrigerating system completely refrigerated. All power elements are "gas charged" to a definite pressure, preventing motor overload and providing quicker response and more sensitive control. Capacities from ⅛ ton to 30 tons Freon. The No. 673 valve employs a double bellows as the actuating means, while the Durafram line is constructed with a single diaphragm power element. All needles and seats are made of Delubaloy, a very hard, corrosion resistant alloy to insure long, trouble-free service. Write for Catalog No. 200 on refrigeration equipment.



Refrigerant Distributors

Detroit No. 790 Refrigerant Distributors insure a uniform supply of refrigerant to all sections of multiple circuit evaporators and are attached directly to the expansion valve by means of a flanged connection. These corrosion-resistant distributors are available for Dura-fram Expansion Valves No. 786, No 787 and No. 788, and are interchangeable on all types.

They are made in two types, as follows:

Type A—2 to 8 passes.
6 passes standard.

Type B—9 to 18 passes.
12 and 18 passes standard.

All distributors are designed for use with ¼ in. O.D. copper outlet tubes.

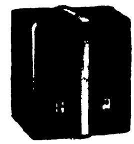
For complete capacity tables write for Bulletin No 207.

Barber-Colman Company

Rockford, Illinois

Automatic Control Systems for Heating, Ventilating, Air Conditioning

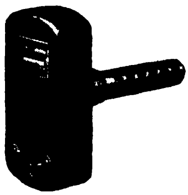

Room Thermostat


Room Type Microtherm

Barber-Colman Controls are all electric; precision built to insure long, continuous and dependable service; easy to install in either new or existing buildings; and ready for instant service, even after long shut-down periods.

Thermostats. All types—room, duct, immersion, air stream and remote bulb. For 2-position, floating, and proportioning control.

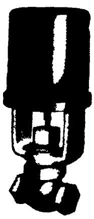
Hygrostats. Room and duct types.

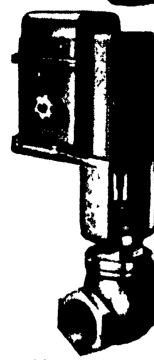

Duct Thermostat


Hygrostat

Motor-Operated Valves. Packless, packed single-seat, pilot piston, V-ported, balanced, 3-way, and butterfly. For 2-position, floating, or proportioning control. Also Solenoid Valves for air, oil, water and gas. *Motor-operated valves are powered with Barcol motors which have only one moving part and require no attention except oiling; oil submerged operators require no attention.*

Control Motors. Uni-directional, or reversible fixed or adjustable speed. For 2-position, floating or proportioning operation of dampers in heating, ventilating or air conditioning applications. *Oil submerged models have the motor and gear train entirely submerged in oil.*


Motor-operated Shut-off Valve


Motor-operated Proportioning Valve

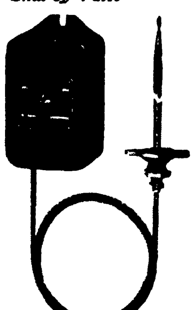
Program Switches. Automatic contact making mechanism for multi-compressor control or similar applications.

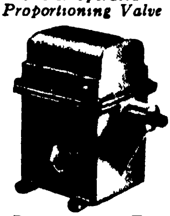
Econostat (not illustrated), a complete self-contained thermostatic unit for automatic regulation of the heat supplied to a building in accordance with outdoor temperatures.

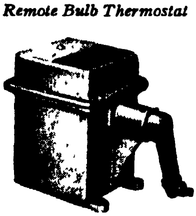
Write for descriptive literature.

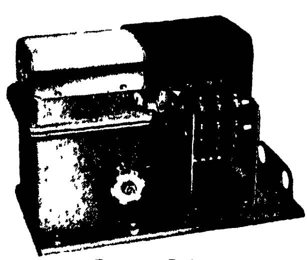
LISTED AS STANDARD BY UNDERWRITERS LABORATORIES

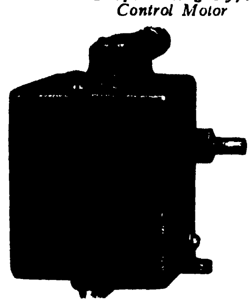
(*See also Page 972*)


Remote Bulb Thermostat


Proportioning Type Control Motor


Stall Type Control Motor


Program Switch

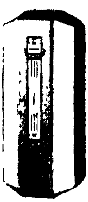

Heavy Duty Industrial Type Control Motor

Friez Instrument Division

BENDIX AVIATION CORPORATION

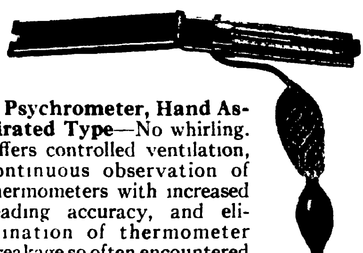
Towson, Baltimore 4, Maryland

Manufacturers of Automatic Controls for Heating, Air Conditioning and Refrigeration and of Precision Recording and Measuring Instruments.

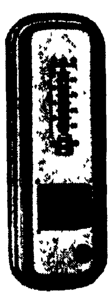


Humidistat—Employs the ever reliable dual hair element to operate electrical contacts. The ideal control for residential, commercial or industrial humidifying or dehumidifying equipment Insertion type, not shown, also available in several models.

Thermostat—A complete range of highly sensitive and thoroughly dependable thermostats

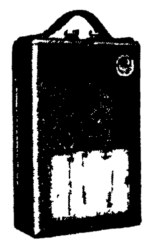


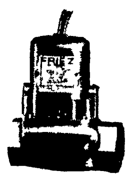
Psychrometer, Hand Aspirated Type—No whirling. Offers controlled ventilation, continuous observation of thermometers with increased reading accuracy, and elimination of thermometer breakage so often encountered in the sling type. Complete instrument with psychrometric slide rule, folds into protected pocket item.



Humidity Indicator—Range: 10-95 per cent Relative Humidity. Employs the Friez dual hair element to provide high sensitivity and best accuracy. Includes thermometer as illustrated.

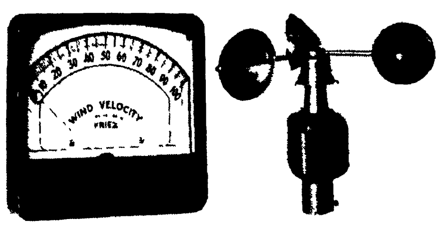
Recorder, Portable—For surveys, studies, and tests of humidity and temperature. A third pen (optional) provides an on-off record of associated electrical equipment under test. Provides inked record on 3 x 5 cards, suitable for filing.





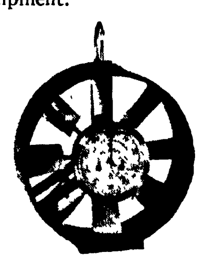
Magnetic Gas Valve—Low or high voltage electromagnet in external chamber controls free floating internal disc.

Wind Speed System—A 3 cup wind driven rotor actuates a d.c. generator connected to a remote indicator. No external



source of power necessary. Friez manufactures a complete line of meteorological instruments and equipment.

Anemometer, Biram Type—Accurate and reliable measurement of air speed. Ideally suited for measurements of air flow in ducts, or flues, and from fans or grilles. Start and stop lever and instantaneous zero reset.



Write for Descriptive Literature Stating Your Requirements

The Fulton Sylphon Company

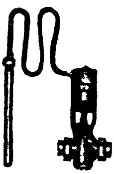
Manufacturers of Sylphon Automatic Temperature Controlling Instruments and Packless Expansion Joints

Sales Representatives in Principal Cities

Knoxville, Tenn.

HOT WATER SUPPLY

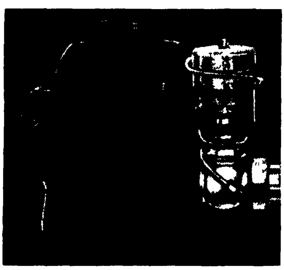
No. 923 Temperature Regulator—For controlling water temperature in heaters, open or closed tanks and other equipment. Operation unaffected by temperature fluctuations at the valve, either above or below bulb temperature. All parts, except steel adjustment spring, made of non-ferrous metals. May be installed in any position. Ranges from 40 −80 F to 290 −330 F. Bulletin HVG-20.



No. 923 Temperature Regulator

Sylphon Thermostatic Water Mixers

Utilize hot water from any storage tank or instantaneous heater, and effectively regulate the amount of cold water required to temper it to the desired degree, actually mixing the hot and cold water together before delivery. Temperature remains constant in spite of fluctuations in supply water temperatures or pressures.



No 902 Sylphon Thermostatic Water Mixer—14 to 131 gpm depending on water pressure

Four sizes with capacities ranging from 5 to 131 gpm. Bulletin HVG-40.

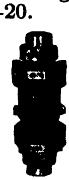
REFRIGERATION CONTROLS

Adaptable wherever brine is used as the refrigerant. Latest development is a "freeze-proof" valve (illustrated at left on the popular Sylphon No. 945-Z Regulator). Bulletin HVG-20.

No. 945-Z Regulator

PACKLESS EXPANSION JOINTS

The Sylphon Packless Expansion Joint eliminates useless building height, expensive construction and non-revenue producing space. No costly leaks and repairs, no repacking, always tight, allows heating system to operate at full efficiency. Write for Bulletin HVG-140.



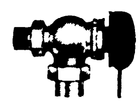
No. 110 Sylphon Expansion Joint

SPACE HEATING CONTROL

No. 885 Automatic Radiator Valve—For exposed radiation. Small, neat, finely finished, adjustable to room temperature desired. Simply replace ordinary radiator valves with these Sylphon Automatic Regulators—no wiring, piping or auxiliary equipment are required. These valves answer the demand for an inexpensive means of providing accurate, dependable space temperature control in rooms, sections or throughout large buildings, new or old. Similar type valves for concealed radiation — get Bulletin HVG-80.

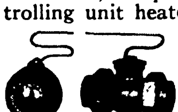
Sylphon No 885 Automatic Radiator Valve

No. 890 Electric Radiator Control Valve—For either exposed or concealed radiation. Similar in appearance and action to Sylphon Automatic Valves, but operated by an electric wall thermostat. The closing of the thermostat circuit energizes a low voltage electric heater coil surrounding a bulb containing a volatile liquid. This liquid expansion causes pressure on a bellows in the valve head operating the valve. This provides radiator valve control from a remote location, permits regulation of several radiators from a single thermostat, enables a time switch to be installed, if desired, offers effective zone control of large areas at a fraction of the cost of conventional motor-operated valve systems Bulletin HVG-70.



Sylphon No 890 Electric Control Valve

No. 7 Temperature Control—A self-contained, self-powered regulator for controlling unit heaters, wall or ceiling type radiators, heating coils in duct-type heating systems, etc. Quickly installed, holds temperatures within close limits. Valve placed in steam line to one or a battery of heaters, thermostat mounted on wall or column. For use on regular heating pressures up to 15 lb. Similar regulators, Nos 7-2 and 7-3 for 50 and 75 lb pressure and temperatures up to 170 F. Bulletin HVG-50



Sylphon No. 7 Temperature Control (Self-operating)

HEATING AND AIR CONDITIONING CONTROL

Almost any type of heating, ventilating or air conditioning system can be advantageously controlled wholly or in part by Sylphon Regulators. Basic advantages of Sylphon Controls are:

Modulating—Maintains ideal conditions—not continually correcting too hot, too cold, too humid or too dry conditions.

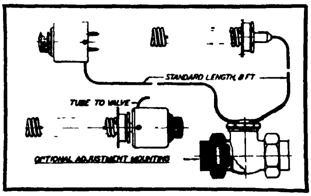
Compensating—Many Sylphon Regulators offer compensating control, automatically raising their low limit setting at a predetermined rate as outside temperatures fall.

Sensitive—Close operating temperature differentials. Quick response.

Simple—in design.

Rugged Construction—To give years of satisfactory service.

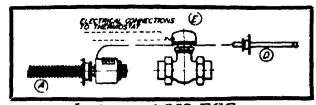
Adaptable—Any one of many combinations of Sylphon Instruments can be arranged to control any air conditioning system and to provide exactly the conditions desired. *Write for Bulletin SA C—820.*



Regulator 928-C

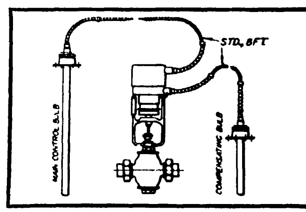
The No. 928-C Regulator—Simple, compact yet highly sensitive. Suitable for modulating control of air temperatures in ducts. Bulb is constructed of numerous coils of copper tubing giving sensitivity to the slightest temperature variation. Packless valve eliminates service problem and makes this regulator ideal for installation in inaccessible locations. Suitable for steam pressures up to 15 lb; other types available for pressures up to 75 lb.

The No. 928-ECC Sylphon Instrument—Room control and low-limit control in a single valve regulator for modulating control of ventilating systems. Main control from an electric room thermostat operating through the electric head "E" on the valve. Low-limit control by Bulb "A" located in discharge duct from the heater. Bulb "D", located in inlet side of the duct to the heater, compensates Bulb "A". Compensating thermostat can be furnished to raise low-limit setting at predetermined rate with falling outside temperature. Suitable for steam pressures up to 15 lb.

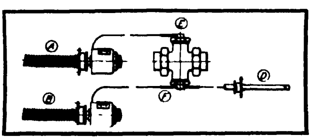


Instrument 928-ECC

Sylphon No. 971 Differential Regulator—For controlling room temperature on the cooling cycle, where chilled water or brine is used as cooling medium and where it is desired to have a gradual increase in room temperature as outside temperature increases. This regulator is modulating in action, thereby affords better control over humidity than is procured when usual on-and-off type control is employed.



Regulator 971

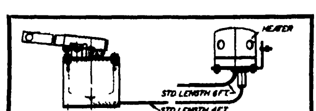


Control 889-C

The Sylphon No. 889-C Control—A modulating, dual-function regulator for control of duct heating and ventilating systems—two independent valves in a single body.

Adjustable Thermostat "A" governing Valve "E" functions to maintain room temperature from temperature of recirculated air. Adjustable Thermostat "B" acts as a low-limit ductstat controlling Valve "F" to maintain minimum discharge air temperature. Bulb "D" compensates Bulb "B" to maintain even discharge air temperature irrespective of demand. Compensated Thermostat "B" can also be furnished to raise its setting at a predetermined rate with falling fresh air temperatures, if desired. Suitable for steam pressures up to 15 lb.

Sylphon No. 371 Positive Type Damper Motor—On-and-off control of dampers. Operation controlled by room thermostat, by hand-operated switch, by motor starting switch, etc. Advantages include: (a) motor returns to closed or safety position in event of current failure; (b) heat-motor bulb and motor separate enhances convenience of installation; (c) damper motor lever adjustable; (d) positive, powerful operation.



Damper Motor 371

GENERAL CONTROLS

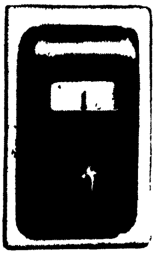
Atlanta ..... 319 Spring St., N.W.
Boston .... 687 Boylston St.
Chicago .... 450 E. Ohio St.
Cleveland ...... 3224 Euclid Ave.
Dallas .... 1100 Cadiz St.
Detroit............ 6523 Grand River Ave.
Kansas City ...... 421 S.W. Boulevard
New York .... 101 Park Ave.
Philadelphia ...... 4515 N. Broad St.
San Francisco ...... 471 Ninth St.
Denver...... 2011 S. Gilpin St.

Home Offices and Factory—801 Allen Ave., Glendale 1, California

DISTRIBUTORS IN PRINCIPAL CITIES

AUTOMATIC TEMPERATURE—PRESSURE—FLOW CONTROLS

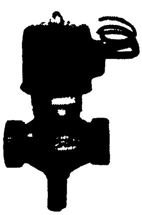
THERMOSTATS



The Trimtherm

Trimtherm provides accurate, remote control of desired temperature. Streamlined, compact—extends only 7/8 in. from wall, yet sensitive to slightest temperature change (Differential 1/2 F.) Chrome cover, natural ivory base.

MAGNETIC GAS VALVES



Type K-8B

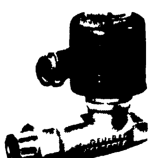
Versatile, two-wire, straight magnetic current-failure valve. Packless. Insures tight shut-off indefinitely. Humless Size range, 7/8 in to 6 in I.P.S Operating pressures up to 5 lb Voltages and frequencies, A.C or D.C. Quiet, positive, trouble-free. Available in explosion-proof housing.

SLOW OPENING GAS VALVES

Type B-66

For industrial and commercial burners and furnaces. Adjustable opening time, 5 to 50 secs. Wide pressure range up to 5 lb. Size 3/4 in. to 6 in. I.P.S. Ample power for louvre control. Damper arm easily rotated. Packless.

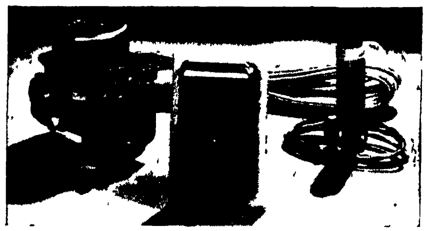
MAGNETIC VALVES



Type K-10

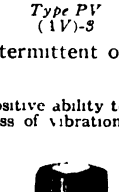
Provides six times more power than ordinary solenoid valves. Controls air, gas, water, light, heavy oils, steam. Positive opening, complete shut-off, packless, hum-free. Available for any voltage, A.C. or D.C., in sizes up to 1 1/4 in. I.P.S., port sizes up to 5/8 in. K-20 Series is designed for applications where single needle port sizes provide sufficient flow capacity. W-3 Series are magnetic 3-way valves for controlling fluid to piston and diaphragm operators on valves, doors, gates, etc.

B-60 GAS ACTUATED PACKAGE SETS



Everything needed in a convenient package for remote gas control—B-60 Valve, Trimtherm (with or without thermometer, or Timer Thermostat), Pilot Burner Generator, and 30-ft wire. No outside current required. Safe, quiet, dependable. Applicable to gas furnaces, floor furnaces, boilers, circulators, gas radiators.

*hi-g MAGNETIC VALVES

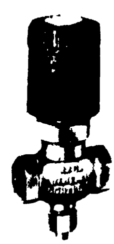


Type PV (1V)-3

Designed for positive operation on aircraft, trucks, tractors, tanks, graders, ships, and other moving equipment. Handle all fluids, vapors and gases on anything that rolls, floats or flies at pressures up to 3000 lb or more. Packless, two-wire, current-failure type, available normally open, normally closed for intermittent or continuous duty.

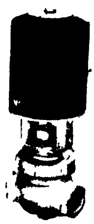
*Trade Mark—"hi-g" indicates positive ability to function in any position, regardless of vibration, change of motion or acceleration.

REFRIGERANT CONTROLS



Type K-15

Magnetic piloted, two-wire, current failure, high pressure, packless. Handle large capacities with minimum pressure drop and loss. Tight shut-off. Operates on wide variety of fluids and gases.

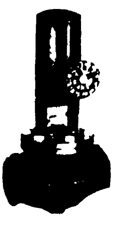


Type G-1-7

HYDRAMOTOR VALVES

Simplify valve control installations. Two-wire, current failure, electric-hydraulic operation. Ample motor-driven power, slow opening and closing movement. Operator drives against a spring in one direction; power failure or opening of circuit causes spring to operate in other direction.

MANUAL RESET VALVES



Type MR-1-2

Equipped with manually-reset electromagnetically-held valve operator. Current flowing to operator permits manual opening by turning valve wheel at side. Current failure releases operator allowing valve to close. Trip-free mechanism cannot be opened under unsafe conditions. Once closed, valve is re-opened manually because applying current has no effect

STRAINERS

Type S-5-1

Actual tests prove importance of strainers in prolonging operating life and reducing leakage of flow controls S-5 Series STRAINERS come in 8 types; meshes $\frac{3}{16}$ in diam to 120-per-inch

THERMOVALVES

Type MR-2

Manually-reset, electromagnetically-held-open valve with current generated by single couple subject to heat of pilot flame. Available $\frac{3}{8}$ in. to $1\frac{1}{2}$ in. I.P.S.

THERMOPILOTS

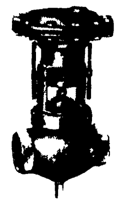
Type A-100

Proven principles of operation insure dependability. Flame applied to thermocouple maintains electrical contact, allowing electrical gas valve to open. When flame fails, Thermopilot opens circuit to main gas valve. Flexible, armored, asbestos-covered cable detachable from relay. 2 or 3-wire control. Electrical rating, 2 amp., 24 volt; 1 amp, 115 volt; 0.5 amp., 230 volt.

THERMAL EXPANSION VALVES

Type V-200 provides unmatched sensitivity and dependability. Handles freon, methyl chloride, sulphur dioxide. Quickly removed orifice cartridges eliminate need for stocking several sizes for low tonnage installation.

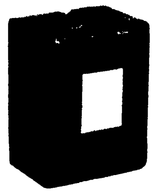
GAS FUEL GOVERNORS



Type V-25-1

Throttle gas lines according to boiler pressure applied to diaphragm. Ball bearing thrust adjustment, ground and polished non-corrosive stems, low friction packing gland seal, multiple calibrated springs, high lift for maximum capacity. Suitable for butane, natural or manufactured gas. Available $\frac{3}{8}$ in. to 3 in., I.P.S.

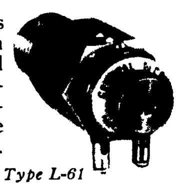
RELAYS AND TRANSFORMERS



Type RS-100

Type RS-100 handles single phase motor loads up to 1 hp or heating loads up to 1.1 kw. Combines double-break relay and integral transformer. Normally open; large double-break contacts. Two-wire control circuit; maximum holding current 0.4 amps. Furnished with $\frac{1}{2}$ in. conduit connections and low voltage outlet. A C. only.

TANK THERMOSTATS



Type L-61

Averages 6 to 10 degrees effective differential on water heater Saves fuel by eliminating overheating due to large differentials. Temperature range, 60 to 170 F. Also to 230 F.

LOW PRESSURE GAS REGULATORS

Type V-300

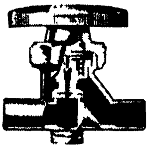
New V-300 Series are reliable, trouble-free valves with high capacity, close regulation, yet small and compact. Regulator sizes range from $\frac{3}{8}$ to 2 in. I.P.S. Internal parts, corrosion-resistant. Cast iron regulator bodies, long life calibrated springs, properly fitted all-metal valves.

GENERAL CONTROLS manufactures a complete line of Automatic Temperature, Flow and Pressure Controls. For complete specifications write for *Catalog 52B*.

Henry Valve Company

1001-19 North Spaulding Ave., **Chicago, Ill.**

PACKLESS AND PACKED VALVES • DRYERS FOR REFRIGERATION AND AIR CONDITIONING • STRAINERS • AMMONIA VALVES • FORGED STEEL VALVES AND FITTINGS FOR OIL, STEAM AND OTHER FLUIDS.



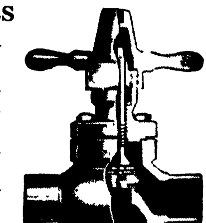
Balanced-Action Diaphragm Packless Valves

VALUE OF BALANCED-ACTION

Regardless of operating conditions or the differential in the pressure above and below the valve seat, balanced-action assures positive and instantaneous opening. The balancing action is really the equalizing of the pressures on the two sides of the valve seat at the instant of opening. This is accomplished by a channel in the axis of the valve stem. When the valve is closed, the upper port of this channel is sealed by the diaphragm assembly itself. At the instant of opening, the pressure above the seat forces the diaphragm assembly upward, exposing the upper port of the balancing channel. The pressure is released through the channel to the region below the seat, equalizing the pressures, thus assuring positive opening. A spring-tensioned ball check seals the channel for diaphragm inspection

Other Important Features are: Oval handwheel, ports-in-line, non-rotating bearing plate to protect diaphragm from rotating friction of stem, and use of multiple puncture and fracture-proof diaphragms designed to resist wear and corrosion Available in a complete range of sizes with flare and solder connections.

WING CAP VALVES



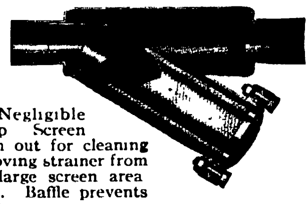
Designed especially for Freon and Methyl Chloride. Have patented rotating self-aligning stem disc. Special resilient packing. May be repacked under pressure. Wing cap can be inverted and socket used for operating valve.

Made of non-ferrous alloy to meet government specifications Solder connections machined directly in valve body

HENRY STRAINERS

There is a size and type of Henry Strainer for every installation requirement.

Type 895 "Y" Strainer



With solder fittings for use with copper pipe Welded steel construction Negligible pressure drop Screen can be taken out for cleaning without removing strainer from line Very large screen area Light weight. Baffle prevents heavy particles injuring screen

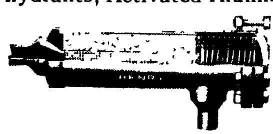
ABSO-DRY PRESSURE SEALED DRYERS

For Refrigeration and Air Conditioning

The exclusive Henry vacuum process first removes every trace of moisture, then the dryer is charged with dehydrated air. Loosening a seal cap prior to installation produces hissing sound, a guarantee of original factory dryness and freedom from leaks.

Other Features of Henry Dryers—*Perforated Dispersion tube* is connected to inlet port and exposes entire volume of dehydrant to penetration by refrigerant *Minimum* pressure drop. No channelling. *Compression Spring* maintains uniform tension on dehydrant at all times and compensates for changes in volume. *Soldered or Flanged Shells*—models are available with either soldered cap or flanged end shells. Flange is distortion-proof. Shells not exceeding 5½ in in length are drawn in dies, so that they have only one joint.

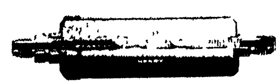
TWO DEHYDRANTS—Choice of two dehydrants; Activated Alumina or Silica gel.



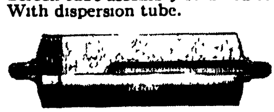
Type 757 Cartridge Dehydrator

Flanged shell dehydrator with replaceable cartridge.

Type 743 Refillable Dehydrator



Screen tube assembly soldered to outlet connection. With dispersion tube.



Type 712 Dehydrator

Soldered brass shell with dispersion tube

APPROVED FOR ARMY AND NAVY USE

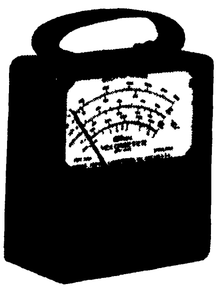
Illinois Testing Laboratories, Inc.

422 North LaSalle Street, **Chicago, Illinois**

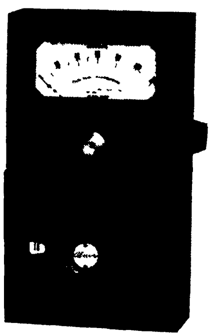
Alnor

Manufacturers of

ALNOR Thermometers, Pyrometers, Temperature Controllers, and Air Velocity Meters

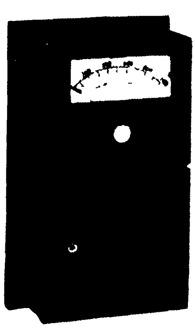


The Alnor Velometer is an instantaneous, direct-reading air velocity meter designed for convenient, rapid determination of air velocities in air conditioning, heating and ventilating, and exhaust systems. It gives instantaneous direct readings in feet per minute, without timing, calculations, or reference to tables or charts. Low range air velocity readings are taken by holding the **Velometer** in the air stream. High range readings are obtained by the use of tube connected special jets, available in 17 types providing for convenient use in any location. Accurate information on performance of equipment, duct systems, etc., can be obtained with a few moments inspection with the **Velometer.** It can be effectively used to locate drafts and leaks around windows and doors, or in duct systems.



The Alnor Velometer is built in several standard ranges from 20 fpm to 6000 fpm, and up to 3 in. static or total pressure. Special ranges available as low as 10 fpm and up to 25,000 fpm velocity and 20 in. pressure.

Alnor Electrical Resistance Thermometers, available in single and multi-point types, are ideally suited to air conditioning, heating and refrigeration installations. Thermobulbs may be located at any point up to 1000 ft from the instrument. Indicating thermometers can be furnished with capacities up to 14 points. Standard types include round, rectangular, horizontal edgewise, and portable thermometers. Instruments may be installed in office or engine room and thermobulbs located at any desired point in the building or on the roof. Accurate temperature measurements at the remote locations are obtained merely by moving the selector switch.



Alnor Pyrometers include a wide range of portable and wall mounting types, both single and multi-point instruments, for a variety of industrial temperature measurements. Temperature readings of molten metals, industrial furnaces, and similar equipment, and temperatures of surfaces, either metallic or non-metallic, moving or stationary, are readily obtained with standard **Alnor** Pyrometers.

Bulletins describing **Alnor** instruments and accessory equipment will be sent upon request.

Johnson Service Company

AUTOMATIC TEMPERATURE AND AIR CONDITIONING CONTROL

General Offices and Factory

Milwaukee, Wis.

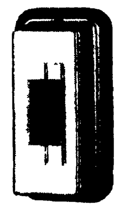
Branch Offices in all Large Cities

JOHNSON TEMPERATURE REGULATING CO. OF CANADA, LTD., 113 SIMCOE ST., TORONTO, ONT.
HALIFAX, N. S. MONTREAL, QUE. WINNIPEG, MAN. CALGARY, ALTA. VANCOUVER, B. C.

PRODUCTS AND SERVICES

Manufacturers, Engineers, and Contractors—For automatic temperature and humidity control systems applied to all types of heating, cooling, ventilating, and air conditioning installations.

Space Control—Automatic control of room temperatures and humidities, applied to radiators, unit ventilators, unit heaters, and heat delivery ducts. Johnson "Duo-Stats" to maintain the proper relationship between outdoor and heating system temperatures for groups of radiators, or "heating zones." A complete line of devices for automatic control of air conditioning systems, heating, cooling, humidifying, dehumidifying.



Single Room Thermostat

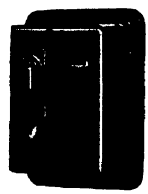
Process Control—Automatic temperature and humidity control devices for manufacturing and industrial processing, applied to tanks, dryers, vats, kettles, curing rooms, coolers, kilns, etc.

Nation-wide Service—Johnson sales engineers, technicians, and trained installation men are available at all branch offices. None of the men in the nation-wide Johnson organization are agents, jobbers, or part-time representatives. All are salaried employees, devoting their entire efforts to the interests of the JOHNSON SERVICE COMPANY and its customers.

Send for Bulletins describing the detailed characteristics of any of the Johnson devices.

JOHNSON THERMOSTATS

Room Thermostats—Intermediate (gradual) or positive (snap) action, maintaining temperatures accurately within one degree above or below point of setting. "Dual" (two-temperature) and "Summer-Winter" types, as well as standard instruments. Various types of covers allow wide selection of adjusting features, guards, and method of mounting. Red-reading thermometers with magnifying tube attached to each cover.

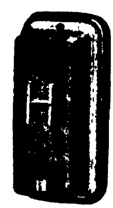


Dual Room Thermostat

Insertion and Immersion Thermostats—Control temperatures in ducts, tanks, and similar locations. High grade insertion or immersion thermometers for mounting adjacent to the thermostats, including the distinctive Johnson insertion thermometer, with red-reading mercury column in heavy lens glass tube and 9-in. scale with patented adjustable tilting feature. Also, separable socket liquid thermometers.

Extended Tube Thermostats—Liquid-filled or vapor tension type, to measure temperatures at a point remote from the location of the operating mechanism. Various types of bulbs. Connecting tubing up to 50 ft in length for vapor tension.

Special Thermostats—For applications encountered in industrial control, including the "Record-O-Stat," combination extended-tube temperature controller and recorder. Full 10-in. chart. Single or duplex type, the latter controlling and recording both wet and dry bulb temperatures.

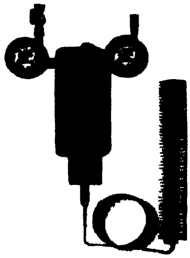


Room Humidostat

Remotely Adjusted Thermostats—A distinctive Johnson feature, applied to various types of instruments where readjustment must be accomplished from a remote point, such as another thermostat or a manual switch.

Johnson Sensitivity Adjustment—An important development in automatic temperature and humidity control for air conditioning. A unique and convenient means of adjusting the sensitivity of Johnson thermostats and humidostats, on the job, balancing "time-lag" with respect to the capacity of conditioning apparatus. "Hunting" and temperature fluctuation prevented. Available on all Johnson gradual action insertion and immersion thermostats, insertion humidostats, and certain room type thermostats and humidostats.

"Sylphon" Radiator Valve



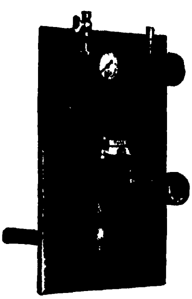
Extended Tube Thermostat

JOHNSON HUMIDITY CONTROL

Johnson Humidostats—Automatically control the supply of moisture delivered by a humidifier or by other means, maintaining a constant percentage of relative humidity. Available in both room and insertion patterns and with various types of elements as determined by requirements, the most sensitive controlling within 1 per cent at relative humidities as high as 95 per cent at 100 F. Humidostatic elements are wood-strip, human hair, animal membrane, or other suitable substances as selected.

Johnson Humidifiers—"Steam grid" type (perforated pipe supplied with low pressure steam) or pan type with copper evaporating pan, brass heating coils, and float control.

JOHNSON VALVES



Remotely Adjusted Duct Thermostat

Johnson Diaphragm Valves—Simple and rugged. Seamless metal bellows and heavy spring operate the valve stem. Available, if desired, with diaphragms of special molded rubber, resistant to aging, heat deterioration and oxidation. No complicated moving parts. Made in all standard sizes and patterns. Direct acting (normally open) or reverse acting (normally closed). Also, three-way mixing and by-pass valves. For steam, water, brine, and freon.

Johnson "Streamline" Diaphragm Valves—Modulating discs and special internal construction, insure superior gradual control . . . Where maximum power is required for repositioning at the slightest demand of controlling instruments, Johnson molded rubber diaphragm valves are fitted with Johnson's dependable pilot feature, for smooth gradual operation, independent of friction and pressure variations.

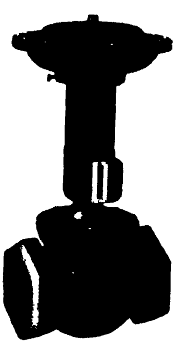
JOHNSON DAMPERS AND SWITCHES

Standard Johnson Dampers—Steel blades in flat steel frames with adequate bracing to form a rigid assembly Finished in two coats of black lacquer. Special corrosion-resisting finishes on order. Angle iron frames optional **Special Dampers**—Galvanized iron, monel metal, aluminum, copper, rust-resisting steel, etc. Brass pins in steel bearings or ball bearings.

Modulating Attachment for Expansion Valves

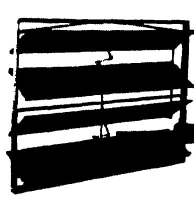
Johnson Damper Motors—In principle, similar to valves. Seamless metal or specially molded rubber diaphragm operates damper through suitable linkage. Various types of brackets. Distinctive Johnson "Piston-type" damper motors afford long travel at full power, a feature not found in other such devices. With or without pilot mechanism, as described above under "Valves."

Johnson Pneumatic Switches—Various patterns for operation of dampers and for placing thermostats and other devices in and out of service, as required, from remote points Standard switchboards are oiled slate. Ebony asbestos, polished oak, and genuine or imitation marble on order.

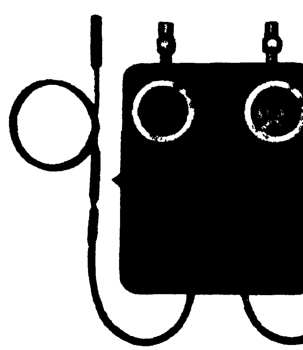


Rubber Diaphragm Coil Valve

Piston Type Damper Motor



Louvered Damper



Johnson "Duo-Stat"

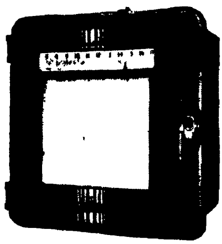
Leeds & Northrup Company

4941 Stenton Avenue, Philadelphia 44, Pa.

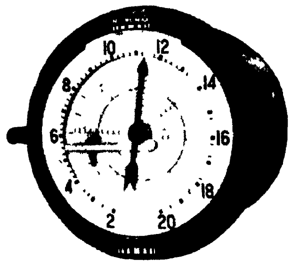
Branch Offices:

Boston Buffalo Chicago Cincinnati	Cleveland Detroit Hartford	Houston Los Angeles New York	Pittsburgh St Louis San Francisco Tulsa

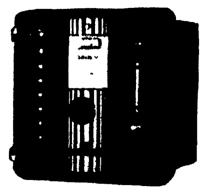
RUGGED, ELECTRICAL-BALANCE INSTRUMENTS



Model S Micromax Recorder
Records from 1 to 16 points on a single strip-chart Extremely open record. Can also signal or control (About 1/15th size)



Model R Micromax Recorder
Records 1 or 2 points on a round-chart Has extremely readable dial Can also signal or control (About 1/15th size)



Panel Indicator
Hand-operated Can be connected through selector switches to any number of points (About 1/12th size)

Electrical Thermometers for Air Conditioning

No method for measuring temperatures fits the specific needs of air conditioning as does the three-lead, null-type resistance thermometer method. It is independent of distance and disregards all temperatures except those right at points of measurement. Thermohms (electrical resistance thermometers) can be placed anywhere—in rooms, air ducts or water lines. They are connected by simple electrical wiring to instruments at a central location. Instruments may be: Micromax Recorders, Model S for up to sixteen Thermohms, Model R, for related pairs such as wet and dry bulb; indicators with switches for any number of Thermohms; or indicating and recording combinations.

Sound in principle, reliable in operation, instruments and Thermohms are highly responsive, yet ruggedly built. A complete system is economical to install, regardless of distances. It is easy to operate; requires little maintenance. Thermohms and instruments are interchangeable; can be replaced without disturbing wiring or returning anything to the factory.

L&N Resistance Thermometers make it possible to operate efficiently; to maintain comfort or correct process atmosphere constantly . . . so that maximum return is realized on the conditioning investment. Ask for Catalog N-33C.

Jrl Ad N-225(2)

Electrical Instruments for the Steam Plant

Facts needed to run a modern heating plant so as to save fuel, protect equipment, and operate efficiently at varying loads are provided reliably by rugged L&N instruments. Readings can be indicated, recorded, or both. Recorders can be equipped to operate signals or alarms; in some cases to control automatically.

Micromax Model S provides a permanent record on one wide-scale chart of conditions at from 1 to 16 points. Micromax Model R concentrates on conditions at one point, provides a permanent record, has a giant indicating dial. A Panel Indicator provides intermittent checks on conditions at one or several points.

L&N equipments for the steam plant are described in the following publications:

Metermax Combustion Control (for large plants; central stations) Cat N-01M-163.

L&N Combustion Control, Type P (for industrial and smaller stations) Cat. N-01P-163.

Micromax Temperature Instruments (for the Steam Plant) Cat. N-33-163

Micromax Temperature Instruments (for Superheated Steam) Cat N-33-163(1).

Centrimax Flowmeters, Cat N-28-160.

Micromax CO_2 Recorders Cat. N-91-163.

Micromax Smoke Density Recorders, Cat. N-93-163.

Micromax Condensate Purity Instruments (for the Steam Plant) Cat. N-95-163.

ASHCROFT GAUGE DIVISION
AMERICAN SCHAEFFER & BUDENBERG INSTRUMENT DIVISION

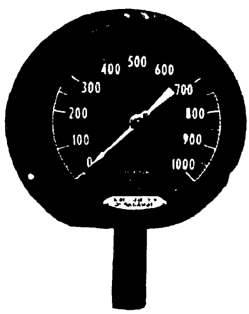
Manning, Maxwell & Moore, Inc.

Bridgeport, Conn.—BRANCHES IN PRINCIPAL CITIES

Makers of AMERICAN INDUSTRIAL INSTRUMENTS—Since 1851

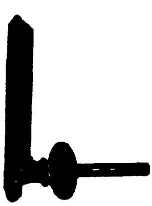
Manufacturers of Indicating and Recording Gauges; Gauge Testers; "U" Gauges; Draft Gauges; Indicating and Recording Thermometers; Tachometers; Dial Thermometers; Pressure and Temperature Controllers; Electric Temperature Controllers; Pop Safety and Water Relief Valves; Steam Traps; Absolute Pressure Gauges.
Also manufacturers of Bronze, Cast Steel and Forged Steel Valves, Engine Room Clocks; Barometers; Mercury Column Gauges; Gauge Boards.

Ashcroft Turret Case Duraguage—Is recognized for durability and accuracy. The entire gauge system is easily removed from the case as a unit The socket, tube, movement, dial, and pointer are an integral unit, independent of the case The Phenol Turret Case is light, strong, rigid, non-corrosive, and will not warp or crack affected by continuous heat under 250 to 300 F. Seventeen types for a variety of services



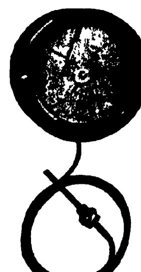
Recording Duragauges—Recording Duragauges are made for all pressures from 15 in. of water to 10,000 lb and for vacuum. They are made in one size only to accommodate a 10 in. chart, having an effective scale width of 3⅝ in. The case is die cast with a dull black hard-rubber finish and with either bottom or back connection. The pen-arm is made of non-corrosive monel metal and is of the inverted type. Operating instructions are lithographed on the chart plate so that they cannot be lost. Write for Catalog E-59.

American Air Duct Thermometer-Designed especially for both warm and cold air ducts. Dull black finish, glass front. Furnished with 9-in. scale graduated 0-160 F. Write for Catalog F-59.



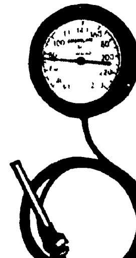
American Recording Thermometers—Made for recording temperatures from minus 40 to plus 1000 F or equivalent C. Very flexible connecting tubing up to 200 ft. One size to accommodate 10 in. chart, with an effective scale width of 3⅝ in.

Same case as for the American Recording Gauge, so that all instruments are uniform in appearance when mounted on Gauge Boards. Write for Catalog H-59.

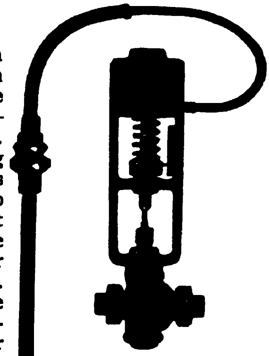


American Dial Thermometers—American Dial Thermometer (mercury-filled) has the accuracy of the standard glass tube thermometer and the reading convenience of a dial face. Entire working mechanism is made of steel, meaning long life.

Three sizes, ranging from 4½ in. to 8½ in. diameter dials. Furnished with rigid connection or flexible capillary tubing up to 200 ft. For temperature ranges from minus 40 to plus 1000 F. Write for Catalog G-59.



American Precision Temperature Controllers—Self-operated. For regulating temperatures from 20 to 325 F. For hot water service tanks, water heaters, etc. Size of valve must be specified. Write for R-59 Bulletin.



The Mercoid Corporation

Main Office and Factory, 4201 BELMONT AVENUE, Chicago 41, Illinois

NEW YORK OFFICE, 205 EAST 42ND ST.   PHILADELPHIA OFFICE, 3137 N. BROAD ST.

AUTOMATIC CONTROLS FOR HEATING, AIR CONDITIONING, REFRIGERATION AND VARIOUS INDUSTRIAL APPLICATIONS

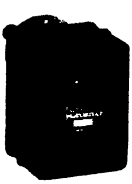
FOR PRESSURE & TEMPERATURE

These controls have a wide range of applications. They are noted for their accuracy and dependable performance. The outside double adjustment feature and visible dial eliminate all guesswork when setting the operating range

VISAFLAME

The Mechanical Eye Actuated by Light. A positive safety control system for domestic and industrial oil burners. Operates direct from the light of the flame instead of from heat in the stack Tried and proven to be the most practical method for dependable oil burner performance. (See next page)

OIL BURNER SAFETY CONTROLS



Type JMI provides positive protection against flame or ignition failure on intermittent ignition oil burners. This control insures having ignition circuit closed before every starting operation of burner. Type JM is used for constant ignition burners.

THE ONLY 100% MERCURY SWITCH EQUIPPED CONTROLS

The distinguishing feature of Mercoid Controls is the exclusive use of Mercoid hermetically sealed mercury switches. These switches are not subject to dust, dirt or corrosion, thereby assuring better performance and longer control life. The items shown below are but a few miscellaneous items. See Catalog No 600 for complete line.

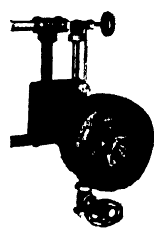
MERCOID THERMOSTATS

Mercoid low voltage room thermostats are known by their trade name SENSATHERM. They operate on a total differential of 1 degree F. Type H is the regular popular room thermostat. Type DNH is a hand wound day and night Sensatherm. Type HBH is the two-stage type for control of high-low oil or gas burners. Recommended also for stokers.

Type 855 is designed for direct line voltage applications recommended for unit heaters, air conditioning, etc

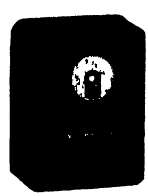
LOW WATER CONTROLS

Available also as a combined low water and pressure control to prevent firing into a dry boiler or building up excess pressure There are a number of different types for various requirements The illustration shows type provided with quick-hook-up fittings designed in accordance with the A.S.M.E. code



STOKER TIMER CONTROLS

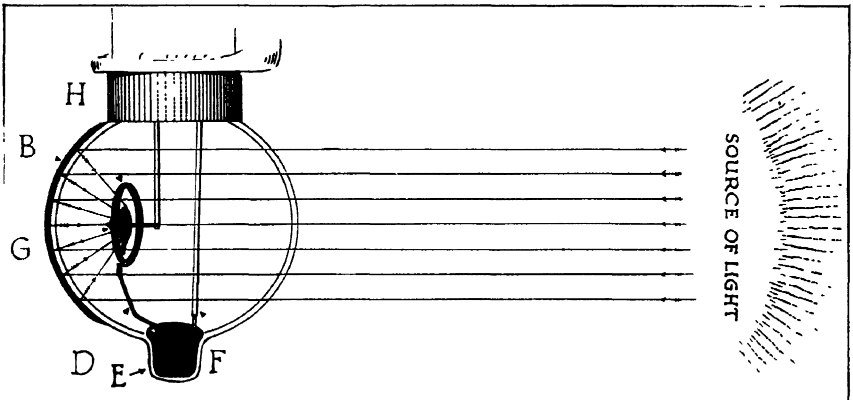
Type TV2 Stok-A-Timer combines a Mercoid Transformer-Relay and a synchronous motor timer mechanism for maintaining the stoker fire during periods when thermostat is not calling for heat. Interval adjustment can be set for ½ hour or 1 hour merely by moving a lever.

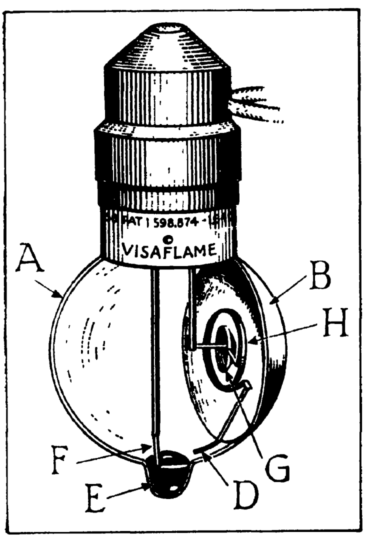


MERCOID VISAFLAME CONTROL SYSTEM

(Actuated by Light)

A POSTWAR RECOMMENDATION THAT HAS BEEN TESTED IN THE LABORATORY OF EXPERIENCE





The Visaflame is a departure from the usual oil burner control operation. Instead of having a safety control on the stack, actuated by the heat from combustion, the Visaflame is located within or adjacent to the oil burner where it uses the light of the flame as a means of actuating the control system. This permits a more direct and positive method for protection against flame failure, power failure, or low line voltage

The Visaflame System lends itself to streamline burner design as it can be built into the burner unit by the manufacturer, thereby making it possible to test the complete burner with safety device at the factory which also simplifies and reduces the cost of installation in the field.

MANNER OF OPERATION

The Visaflame, when properly focused, closes its circuit in the presence of light and opens its circuit when the light is absent. The light of the flame passes through bulb "A" and is picked up by the concave reflector "B" which concentrates the light waves on the small bimetal coil "G " This bimetal coil is opaque and therefore transmutes the light waves into heat, which causes it to move the attached electrode "D" into the pool of mercury "E." The electric circuit is then closed between the fixed electrode "F" and the movable electrode "D." The outer bimetal coil "H" to which the movable electrode is attached, compensates for varying ambient temperatures.

The Visaflame is available for Intermittent Ignition or for Constant Ignition oil burners. Also for manually operated industrial burners. Adaptable to high-low flame operation.

Minneapolis-Honeywell Regulator Company

2711 Fourth Ave., So., **Minneapolis 8, Minn.** Cable Address: MINNREG, MINNEAPOLIS

Electric or Pneumatic Control Systems for Heating, Ventilating, Air Conditioning

BROWN INSTRUMENTS for Indicating, Recording, Controlling

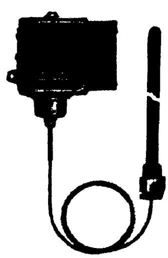
Factories: MINNEAPOLIS, MINN., PHILADELPHIA, PA., WABASH, IND., CHICAGO, ILL.

Branch Offices or Distributors are located in all principal cities.

Albuquerque	Cleveland	Indianapolis	New York	St Louis
Allentown	Columbus	Jackson, Mich.	Oklahoma City	Salt Lake City
Atlanta	Dallas	Kansas City	Omaha	San Francisco
Baltimore	Davenport, Ia	Los Angeles	Philadelphia	Seattle
Birmingham	Denver	Louisville, Ky	Pittsburgh	Springfield, Mass.
Boston	Detroit	Lowell, Mass	Portland, Me	Syracuse
Buffalo	El Paso, Texas	Mason City, Iowa	Portland, Ore.	Toledo
Butte	Fairhaven, Mass	Milwaukee	Providence	Tulsa
Charlotte, N C.	Hartford	Minneapolis	Richmond, Va.	Washington, D C.
Chicago	Houston	New Orleans	Rochester, N Y.	Wichita
Cincinnati				Worcester, Mass

In Canada: MONTREAL, TORONTO, CALGARY, LONDON

In Europe: LONDON, ENGLAND; STOCKHOLM, SWEDEN



Temperature Controller

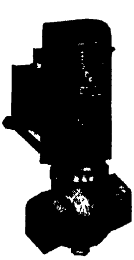
AUTOMATIC CONTROLS FOR EVERY APPLICATION

Minneapolis-Honeywell manufactures a complete line of electric, pneumatic, and self-contained controls and regulators for every type of heating, ventilating, and air conditioning installation. In addition, the Brown Instrument Division of Minneapolis-Honeywell manufactures a complete line of indicators, recorders, and controllers for Industrial Process applications.

Each of the branch offices of Minneapolis-Honeywell maintains a staff of experienced engineers who are qualified to give unbiased advice on any type of control application and to install and service control equipment of any type. They are prepared to assist in the writing of specifications and to furnish control layouts and cost estimates without charge.

Line Voltage Thermostat

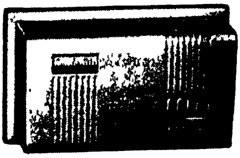
Minneapolis-Honeywell, with 55 years of experience in the control field, is the only company which is prepared to furnish every type of control, whether electric, pneumatic, or self-contained for any type of installation. This eliminates the necessity of purchasing controls from more than one company, which often results in split responsiblity and unsatisfactory results.



"Modutrol Valve"

THE MODUTROL SYSTEM OF ELECTRIC CONTROL

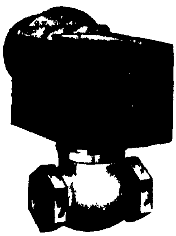
The Modutrol System designation is applied to any combination of Minneapolis-Honeywell Automatic Electric Controls or Self-contained Automatic Valves used to govern the operation of air conditioning or heating systems other than the small domestic installations. A wide variety of both modulating and two position motors, controllers and valves are available thus making the Modutrol System extremely flexible as to the selection of control equipment to produce the desired results.



"Gradustat"
Pneumatic Thermostat

THE GRADUTROL SYSTEM OF PNEUMATIC CONTROL

The Gradutrol System designation is applied to any combination of Minneapolis-Honeywell Automatic Pneumatic Controls used to govern the operation of air conditioning or heating systems. Such features as infinite positioning with the Gradutrol Relay and accurate graduation of valve and damper motors makes the Gradutrol System a truly remarkable advance in pneumatic control of commercial air conditioning and space heating installations.



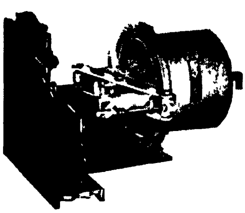
"Grad-U-Valve"
Pneumatic Control Valve

COMBINATION ELECTRIC AND PNEUMATIC SYSTEMS

The outstanding advantages of both the electric Modutrol System and pneumatic Gradutrol System of control may be combined in a single installation. Thus maximum flexibility and low installation cost are obtained. Minneapolis-Honeywell can offer either an electric or pneumatic system, or a combination of the two. This is your guarantee of an unprejudiced recommendation.

BROWN INSTRUMENTS

The extent to which air conditioning equipment is being used in office buildings, theatres, stores, industrial buildings, etc., has opened up a wide demand for indicating and recording resistance thermometers because the temperatures throughout these air conditioning systems should be checked periodically in order to obtain the best results at minimum operating cost. To obtain uniform conditions from modern equipment, it is necessary that the engineer in charge of operation have a visual picture of actual conditions

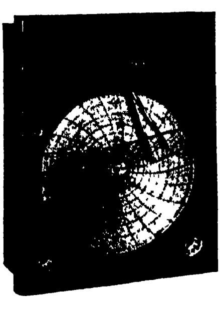


"Grad-U-Motor"
Pneumatic Damper Motor

Brown Resistance thermometers are available for indicating, recording, and controlling service and are applicable to all types of air conditioning and space heating installations.

In addition to Resistance Thermometers, The M-H Brown Instrument Division manufactures:

Thermometers	Flow Meters
Hygrometers	CO_2 Meters
Pressure Gauges	Tachometers
Vacuum Gauges	Liquid Level Gauges
Potentiometer Pyrometers	Protectoglo System



Brown Air Operated,
Controller Recorder

RESPONSIBILITY FOR ENTIRE CONTROL SYSTEM

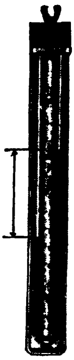
Minneapolis-Honeywell Regulator Co. is equipped to assume the entire responsibility for any control installation, thereby eliminating the difficulties and misunderstandings which division of responsibility may create.

The Meriam Instrument Company

10984 Madison Avenue, **Cleveland 2, Ohio**

Representatives in Principal Cities

MANOMETERS, METERS AND GAUGES FOR ACCURATELY MEASURING PRESSURES, VACUUMS AND FLOWS OF LIQUIDS AND GASES



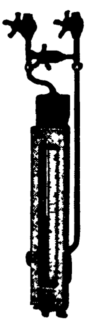
Clean Out U-Type Manometer

U-TYPE MANOMETERS

For measuring pressure, vacuum or differential from only a few ounces to many pounds.

In the Clean Out design (patented) the Pyrex U-tube and housing can be disconnected from the head and cleaned without disconnecting the piping. Available in 8 sizes from 6 in. to 50 in. for pressures up to 100 lb per sq in.

Gland Packed type available for 250 lb per sq in. *Write for Bulletin 1.*



"H" Type Flow Meter

FLOW METERS

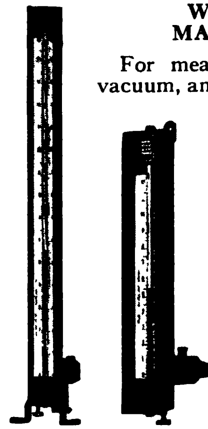
The H-Type Flow Meter—Illustrated at the left is a direct reading well type manometer for measuring the flows of liquids, gases and steam across calibrated plates inserted in the line. The single straight Pyrex glass tube is gland packed top and bottom. Overflow well prevents indicating fluid from being blown into the line if the differential pressure should exceed the meter's range. Plain or flush type mounting for line pressures up to 250 lb per sq in. *Send for Bulletin 18*

6 In Mercury Pressure Gauge

MERCURY PRESSURE GAUGES

For measuring low range air and gas pressures. Gauges are direct reading and so designed that the mercury can not be lost from the gauge either through spilling or overpressuring.

Available in 3 and 6 in. sizes to measure 22 oz and 48 oz respectively. Unit scale graduations are in ounces. Each gauge is furnished complete with pet cock and necessary mercury. *Write for Bulletin No. 3.*



A-203 Table Mounting A-275 Wall Mounting Well Type Manometers

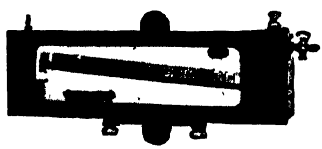
WELL TYPE MANOMETERS

For measuring pressure and vacuum, and differential directly from the scale, using water, red oil or mercury, without adding distances above and below reference zero.

Model A-203—Furnished in wall, table and flush front mounting styles. Ranges 30 in. to 100 in. for pressures up to 250 lb per sq in.

Model A-275—Furnished in wall mounting and flush front styles only Ranges from 6 in. to 24 in. for pressures up to 150 lb per sq in. *Send for Bulletin B.*

DRAFT GAUGES



Single Tube Draft Gauge

Model GP Draft Gauges take accurate readings of low pressures, vacuums and differentials in fractional inches of water. Furnished in single tube, also in 2-tube and 3-tube models for taking multiple readings on the same gauge. *Send for Bulletin No. 4.*

Gauge Pulsation Absorber

GAUGE PULSATION ABSORBERS

The Gauge Pulsation Absorber provides a simple, effective method of preventing "hammer" and pulsing flow in a line from affecting the accuracy of gauge readings, and protects the gauge from damage. *Send for Bulletin No. 2.*

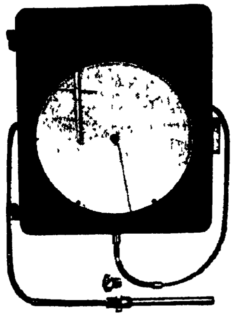
The Palmer Company

Main Plant: 2506 Norwood Ave., **Cincinnati (Norwood), Ohio**

Canadian Factory: King and George Sts., Toronto

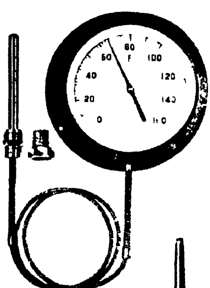
Manufacturers and Originators—"Red-Reading-Mercury" Thermometers

RECORDING THERMOMETERS

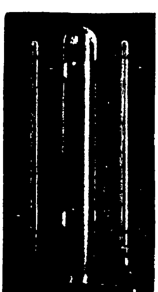


Mercury Actuated. 12 in. die-cast aluminum case. Wrinkle or Satin finish. All parts are rust-proof. Flexible armoured tubing and bulb of stainless-steel. Fittings: Plain, Union, Separable-socket and adjustable or union flange. All ranges up to 1000 F or 550 C. Guaranteed extremely accurate and sensitive. Every part strengthened for long and satisfactory service. *Write for Bulletin No. 1300.*

DIAL THERMOMETER



Mercury Actuated. 8 in. case. Black rubberized finish. Flexible armoured tubing and bulb of stainless-steel. All parts are rust-proof. Fittings: Plain, Union, Separable-socket and Adjustable or Union flange. All ranges up to 1000 F or 550 C. Guaranteed sensitive and accurate and to give long and satisfactory service. *Write for Bulletin No. 1500.*



WALL HYGROMETER and SLING PSYCHROMETER

Wet and Dry bulb; Mercury tube, with RED column. Chart furnished. Guaranteed sensitive and accurate. *Send for Bulletin No. 500.*

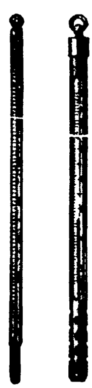
"RED-READING-MERCURY"

Industrial Thermometers—These mercury tubes will show a bright RED color, visible at a great distance. The color is reflected and cannot fade. (Patented by Palmer). Thoroughly annealed and guaranteed permanently accurate. Costs no more. STRAIGHT, ANGLE, SIDE - ANGLE, RECLINING AND INCLINING Case, OBLIQUE STEM, etc. 7, 9 and 12 in. case, with or without glass front. Standard 3½ in. stem and longer lengths. Fittings: Fixed Thread, Union, Separable-socket and Adjustable or Union Flange. All ranges up to 750 F or 400 C. For ranges up to 1000 F or 550 C, with plain mercury tube, borosilicate glass. *Write for Catalog No. 200-F.*

REPAIRS—To all makes of Industrial Mercury Thermometers, furnishing "Red-Reading-Mercury" tube, at no extra cost and replacing all worn or broken parts, making the thermometer as good as new. Guaranteed accurate. A trial order will convince you.

LABORATORY THERMOMETERS

Glass engraved mercury tube; shows bright RED column . . . so easy to see. With or without metal armour; Round or Lens glass; ranges to 750 F. or 400 C Plain mercury tube borosilicate glass on ranges 1000 F. or 550 C. Correctly annealed and guaranteed accurate.



POCKET THERMOMETERS . . . for quick tests. Reliable and accurate. With RED column.

− 20 + 120 F.
0 + 220 F.

Write for Catalog No. 300-D.

Penn Electric Switch Co.

Goshen, Indiana

Offices

ATLANTA; BOSTON, NEWTON, MASS.; CHICAGO; DALLAS; DAYTON; DETROIT; MOLINE, ILL ; NEW YORK; PHILADELPHIA; ST. LOUIS

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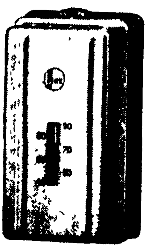
Representatives—GARLAND-AFFOLTER ENGRG CORP., San Francisco, Los Angeles, Seattle, Portland; SPECIALTY SALES CO., Salt Lake City; FORSLUND PUMP AND MACHINERY CO., Kansas City, VINCENT BRASS AND COPPER CO., INC , Minneapolis; H. M. OLMSTEAD, Denver

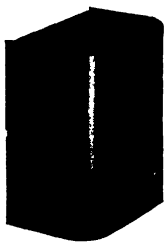
In Canada—POWERLITE DEVICES, LTD., PENN ELECTRIC SWITCH DIV., Toronto, Ont.

Distributors and Jobbers in All Principal Cities

Automatic Controls for Heating, Refrigeration, Air Conditioning, Engines, Pumps and Air Compressors



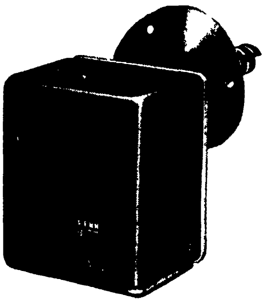
Thermostats



Stoker Timers

HEATING CONTROLS

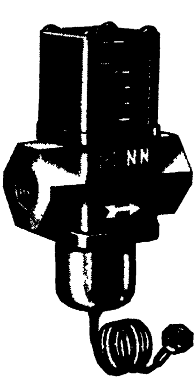
The Penn line of controls includes models for every type of heating service on steam, vapor, hot water or warm air systems whether gas, oil or coal fired. In this complete line there are: Low and line voltage Room Thermostats—Stoker Timers—Oil Burner Stack Switches—Hot Water and Warm Air Temperature Controls—Vapor Controls—Steam Pressure Controls—Relays—Humidistats—Day-Nite Tem Clocks—Damper Motor Controls—Boiler Water Feeders—Boiler Water Level Controls—and Solenoid Valves Condensed description and specifications are given in Bulletin 1508-H. *Write for your free copy*



Oil Burner Stack Switches

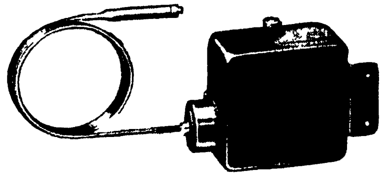
REFRIGERATION CONTROLS

Whatever your need may be in refrigeration controls—choose the exact type to fit the job from Penn's complete line of automatic controls. Pressure and temperature controls are available with or without calibrated adjustments for single phase or polyphase service. Also available are: Cooling Room Thermostats—Humidistats—Relays—Solenoid Valves—and Water Regulating Valves. All of these controls are described and illustrated in condensed Bulletin 1487-K. *Write for your free copy.*

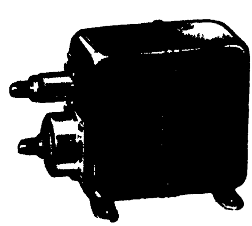


Water Valves

Write for catalog on Penn Controls to cover your particular applications.



Temperature Controls



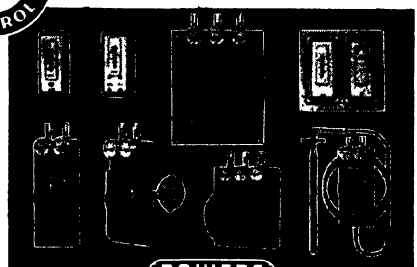
Dual Pressure Controls

THE POWERS REGULATOR CO.

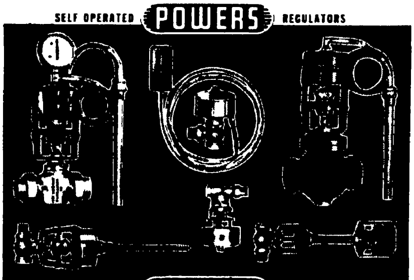
Over 50 Years of Temperature and Humidity Control—Offices in 47 Cities

NEW YORK 17, N Y , 231 East 46th St.—**Chicago 14**, 2719 Greenview Ave —LOS ANGELES 5, 1808 W. Eighth St —BOSTON 15, 125 St. Botolph St —PHILADELPHIA 32, 2240 N Broad St.—GREENSBORO, Jefferson-Standard Bldg —ATLANTA 3, Bona Allen Bldg DETROIT 2, Boulevard Bldg —CINCINNATI 2, American Bldg —NEW ORLEANS 12, Balter Bldg —ST LOUIS 3, 2726 Locust St CLEVELAND 13, 2012 West 25th St.—KANSAS CITY 6, 409 East 13th St —SEATTLE 4, 500 First Ave., S—DALLAS 1, 2438 Brandon St

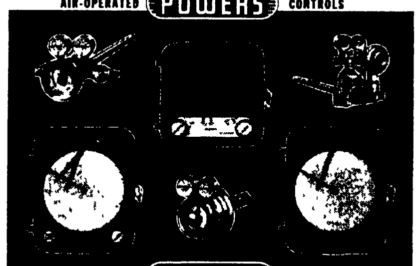
A very complete line of compressed air operated and self-operating temperature, humidity and air flow controls for automatically regulating heating, cooling, ventilating and air conditioning systems and industrial processes.



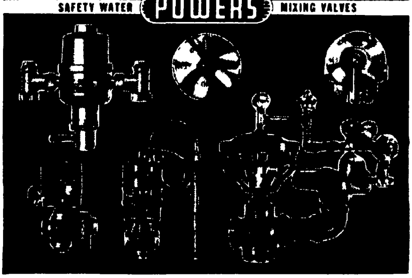
SELF OPERATED POWERS REGULATORS



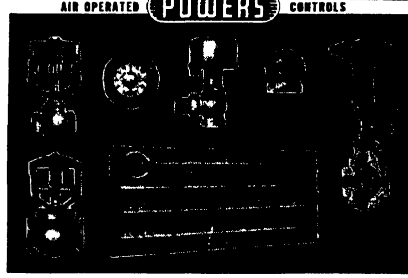
AIR-OPERATED POWERS CONTROLS



SAFETY WATER POWERS MIXING VALVES



AIR OPERATED POWERS CONTROLS



Over 50 years of experience in furnishing and installing temperature and humidity control for every conceivable purpose in all types of buildings have given us a wealth of experience from which you can draw in selecting the proper type of control for any purpose.

Catalogs and Bulletins describing any or all of our products furnished upon request. Phone or write our nearest office. See your phone directory.

Spence Engineering Company, Inc.

28 Grant Street, **Walden, N. Y.**

SPENCE METAL DIAPHRAGM "DEAD END" REGULATORS

Advantages of Spence Regulators

Dead-end Shutoff—Spence Regulators are guaranteed to hold a dead-end

Single Seat—Spence design makes possible a balanced single seat even in large sizes.

Metal Diaphragms—Under normal conditions never require replacement.

Accurate Regulation—Regardless of fluctuations in either load or initial pressure.

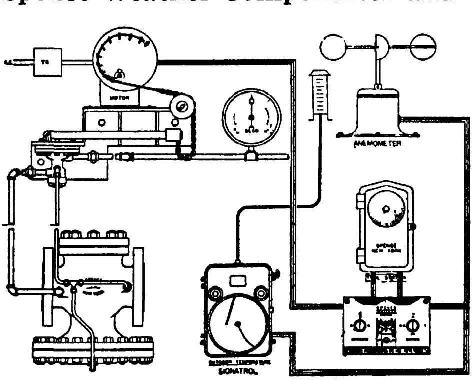
SECO Metal—Guaranteed to resist the wiredrawing action of steam.

Interchangeable Pilots—Any type of pilot will fit any size main valve.

Accessibility—Pilot is connected to main valve with unions.

No Stuffing Boxes—All main valves and most pilots are packless.

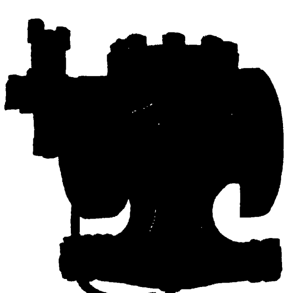
Spence Weather Compensator and Orifice Zone Control System



This simple, dependable Control, when installed on a properly designed orificed heating system, will show a substantial degree-day steam saving, at a low maintenance cost.

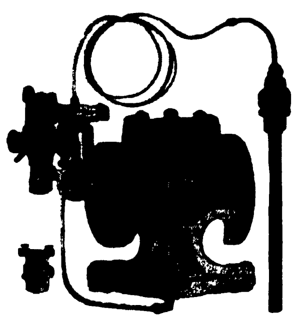
The delivery pressure of the Regulator is automatically adjusted in direct proportion to the building heat losses. In other words, as the losses become greater, steam pressure on the system is automatically increased.

Any number of zones can be controlled by one automatic Signatrol, automatic Wind Loss Compensator (Anemometer) Time Switch and Master Control Panel equipped with Manual and Automatic Dials for each zone. In this way each zone can be set individually and at the same time be under the Master Control.



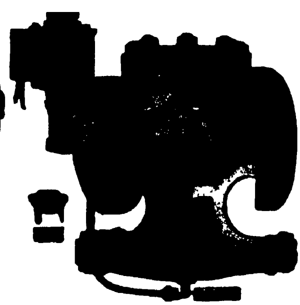
Pressure Regulator—Type ED

Designed to regulate a steady or varying initial pressure so as to maintain a constant, adjustable, delivery pressure. Applicable to heating systems, power plant operations, or manufacturing processes.



Combined Temperature and Pressure Regulator —Type ETD

Self-contained, pilot operated, dead-end. Designed to control flow of fluid to a heating or cooling element, so as to maintain a constant, adjustable temperature, and protect the element against excessive pressure.



Electrically Operated Valve—Type EM

Can be opened or closed independently by an electrical switch

Type ET—Same as ETD except pressure control is omitted

Order a SPENCE Regulator for 40 days' free trial.

Fall-O-Matic Universal Pipe Intersection Cutter.

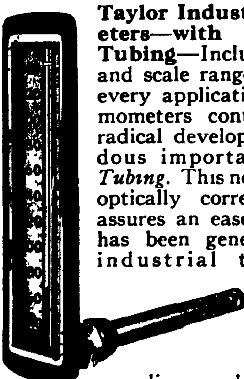
Taylor Instrument Companies

Rochester, N. Y., U. S. A.

IN CANADA—TAYLOR INSTRUMENT COMPANIES OF CANADA, LTD , TORONTO

NEW YORK LOS ANGELES ST. LOUIS DETROIT
CHICAGO PITTSBURGH BALTIMORE CINCINNATI ATLANTA
BOSTON CLEVELAND SAN FRANCISCO TULSA MINNEAPOLIS
PHILADELPHIA *Manufacturing Distributors in Great Britain, Short & Mason, Ltd. London* WILMINGTON

Taylor Instruments for Indicating, Recording and Controlling Temperature, Pressure, Humidity, Flow and Liquid Level

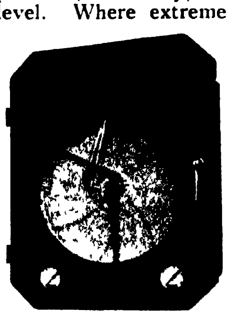


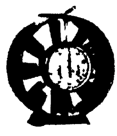
Taylor Industrial Thermometers—with new "BINOC" Tubing—Includes many styles and scale ranges with bulbs for every application. These thermometers contain a new and radical development of tremendous importance—"*BINOC*" *Tubing*. This newly designed and optically correct glass tubing assures an ease of reading that has been generally lacking in industrial thermometers. "*BINOC*" *Tubing* more than doubles the angle of vision within which readings can be made. Because of the patented Triple-lens construction its broad mercury column can be read easily and accurately with both eyes. Bore reflection is absent.

Taylor "BINOC" Pocket Test Thermometer—Ideal for frequent testing of important temperatures. Taylor patented "*BINOC*" *Tubing* eliminates juggling and guesswork. High accuracy—Easier to Read.

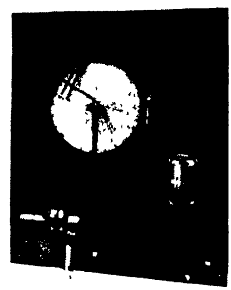
The New Taylor "Fulscope" Recording Controller—An air-operated controller that gives practically any character of process control regardless of *time lag* in apparatus.

Available for controlling temperature, pressure, humidity, rate of flow, liquid level. Where extreme load changes or badly balanced operating conditions exist, precision control can be maintained by the automatic reset feature. For applications where a record is not needed, Taylor supplies an Indicating "Fulscope" Controller.



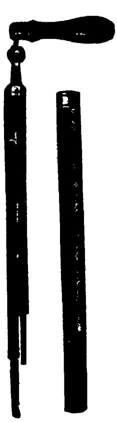


Taylor Biram's Anemometer—Ideal for measuring air velocities with the fan revolutions indicated on the dial. Various models for a wide range of air speeds and registration limits.



Taylor Recording Hygrometer—Records both wet- and dry-bulb temperatures on the same chart in different colored inks, making comparison very easy.

Type shown has motor-driven fan for conditioned rooms or passages where circulation is poor. Furnished without fan for installations where circulation across bulb is good.



Taylor Sling Psychrometer—The advantage of this form of Wet- and Dry-Bulb Hygrometer over the stationary form is the facility with which tests can be made and the accuracy of the readings obtainable, as the whirling bulbs are subjected to perfect circulation. Two accurate etched stem thermometers are mounted on a die-cast frame, with the bulb of one covered with a wick to be moistened.

These thermometers have scales of 20 to 120 degree F, graduated in ½-deg divisions. A copper case protects the tubes when not in use

Taylor also offers a complete line of the famous Taylor Recording and Dial Thermometers, Self-Acting and Type "P" Controllers, the 10-BG Hygrometer and many types of Humidiguides.

White-Rodgers Electric Company

1293 Cass Avenue, **St.Louis, Mo.**

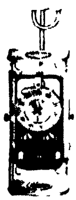
Distributors in Principal Cities

HYDRAULIC-ACTION CONTROLS

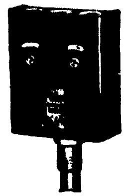
Hydraulic-Action temperature controls combine the powerful uniform expansion and contraction of a solid liquid charged stainless steel diaphragm with mechanical simplicity to provide accuracy and dependability at all times.

Take advantage of **Hydraulic-Action** on your next installation. Specify White-Rodgers Controls. The latest condensed control catalog is awaiting your request. Write for it today!

Line voltage Thermostat for Unit Heater and Air-Conditioning Installations



Single speed fan control-cover removed showing visible dial



Dual Immersion Control—Limit-Circulator or Summer-Winter service

Low Voltage Room Thermostat—anticipating type

8 EXCLUSIVE FEATURES OF WHITE-RODGERS HYDRAULIC-ACTION TEMPERATURE CONTROLS

1. May be mounted at any angle or position, above, below or on level with control point.

2. Hydraulic-Action Principle incorporating solid-liquid filled bulb and capillary provides expansion force comparable to that of a metal bar.

3. Diaphragm motion uniform per degree of temperature change.

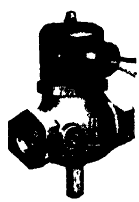
4. Power of solid-liquid charge permits unusually sturdy switch construction resulting in positive contact closure

5. Heavier, longer-wearing parts are possible because of unlimited power

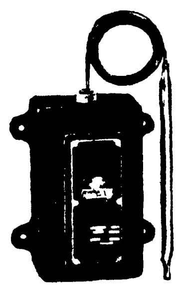
6. Dials are evenly and accurately calibrated over their entire range because of straight-line expansion.

7. Controls with remote bulb and capillary are not sensitive to change in room temperature Accuracy of control is not affected by temperature changes in surrounding area.

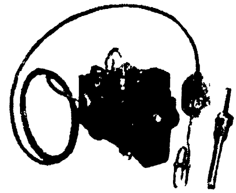
8. Not affected by atmospheric pressure. Works accurately at sea level or in the stratosphere without compensation or adjustment.



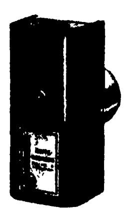
Solenoid Gas Valve—High plunger torque and silent operation



Explosion-Proof Thermostat for hazardous locations—remote type



Diaphragm Gas Valve with puff bleed and built-in mechanical limit control.



Steam Pressure Control—for safety limit service

The Webster Engineering Co.

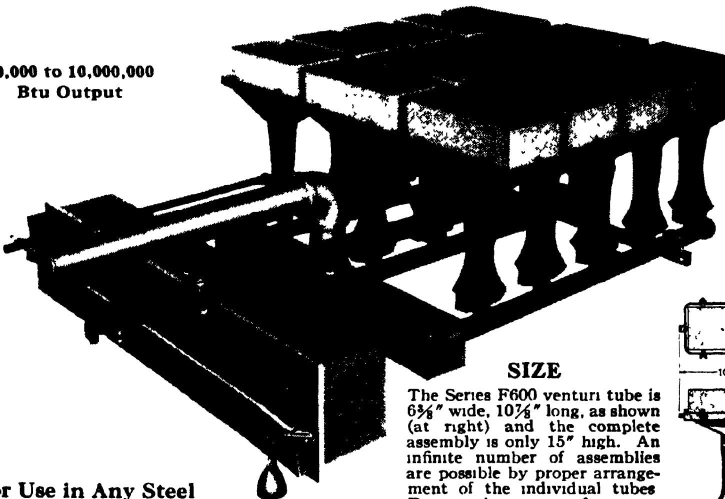
419 West 2nd St., **Tulsa, Oklahoma**

Division of

SURFACE COMBUSTION, TOLEDO, OHIO

WECO-N.G.E. SERIES F600 GAS BURNERS

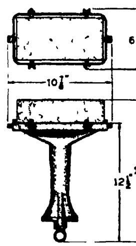
50,000 to 10,000,000 Btu Output



For Use in Any Steel Firebox or Sectional Boiler

SIZE

The Series F600 venturi tube is 6⅝" wide, 10⅞" long, as shown (at right) and the complete assembly is only 15" high. An infinite number of assemblies are possible by proper arrangement of the individual tubes For complete sizing information see Bulletin F600H



Improved venturi and greater port area insure much higher capacities at lower pressures.

Unique baffles at the outlet of the mixing tube make possible perfectly even distribution of flame completely around the baffle brick. As a result the maximum flame length is greatly reduced.

Interchangeable grills with multiple ports can be varied to suit the combustion characteristics of various gases. The proper sizing of these grills prevents any possibility of flash back.

In addition to the above major improvements the F600 possesses the same desirable features that made the 600 so popular.

1. Simple installation requiring no expensive insulated combustion chamber and having no furnace radiation loss.
2. Extreme quietness due to low rate of combustion over a large area
3. Flexibility from infinite number of possible combinations varying both size and shape to meet load and firebox conditions at various gas pressures.
4. High radiant transmission rate due to radiant temperature of the standard firebrick baffles on the top of the burner tubes.
5. Low draft loss because of ample secondary air openings.
6. Plain gas pilots of heat resistant material and of a design that will not allow flame to pull off.
7. Safety pilot applied in a cool zone in a manner that insures perfect direct ignition of the burner yet allowing the the thermal element to cool quickly upon flame failure.
8. Guaranteed vibrationless under all conditions.

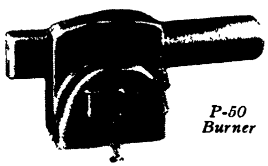
CAPACITY OF SINGLE F600 VENTURI TUBE—No. 17 MTD ORIFICE

Manifold Gas Pressure	0 5" W.C	1 0" W C.	2 0" W C	3 0" W.C.	4.0" W.C.	5 0" W C	6.0" W C.	4 oz. W C.	6 oz. W.C.	8 oz. W.C.
Input—Cu Ft, 1 hr	24 5	38 5	58 0	72 0	84 0	94.5	104.0	112 0	138.0	159.5
Output—Sq Ft, St Rad.	75	117	178	222	258	289	318	343	423	488
Output—Boiler H P.	54	.84	1 28	1 59	1.85	2 07	2 28	2.46	3 04	3.4

Herco Oil Burner Corporation

Lancaster, Pa.

The Herco Line—established since 1925 includes burners ranging from small residence units to burners suitable for large industrial applications. Herco Oil Burners are listed as approved by Underwriters' Laboratories, Inc., for Commercial Standard CS75-39 *Write for descriptive literature.*



P-50 Burner

Herco P-50 and P-35—The P-50 efficiently handles the requirements of industrial, commercial and large residential installations requiring oil burning capacities of 9.00 to 20.00 gals. per hour.

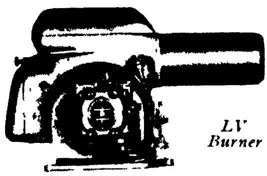
The P-50 is equipped with long-hour duty 1/8 H.P. motor with built-in micro switch for thermal overload protection; 9¼ in. x 4 in. multiple blade Sirocco fan; twin nozzles.

The P-35 provides excellent performance for installations with requirements of 3.00 to 7.00 gals. per hour. The P-35 is equipped with long-hour duty 1/6 hp. motor with built-in micro switch for thermal overload protection; 6 in. x 4 in. multiple blade Sirocco fan.

Standard equipment on each of these Herco P models includes: latest type positive pressure pump; mid-point grounded transformer; electrode porcelains guaranteed for 60,000 volts.

Mercoid Magnetic Valve (K-20-2)—Optional on Herco P-50, this valve cuts off oil flow to nozzle and permits operation of fan for sufficient time to eliminate gas from combustion chamber.

Herco LV 1-2-3—Capacities range from .75 to 7.00 gals. per hour. All three LV models are equipped with long-hour duty motors with built-in micro switch for thermal overload protection

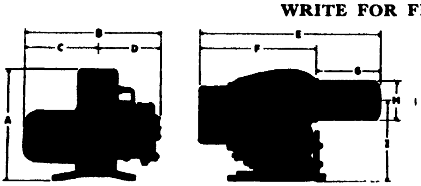


LV Burner

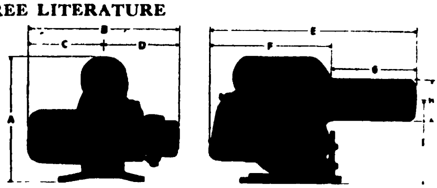
Mercoid "Visaflame"—Standard on all LV models, this light-actuated control shuts off burner within 45 seconds in case of ignition failure. (Standard controls are available on LV models if desired).

Herco "Thrifti-Fier"—Standard on all LV models, the "Thrifti-Fier" permits hot gases to travel slowly through the boiler so that maximum heat is delivered to all heating surfaces.

WRITE FOR FREE LITERATURE



P-50 Measurements



LV Measurements

TABLE OF DIMENSIONS (In Inches)

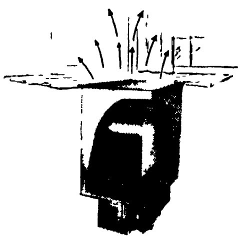
	P-35	P-50	LV-1	LV-2	LV-3
A—Bottom of flange to top of burner (Minimum)	13	19½	13	13¾	15½
A—Bottom of flange to top of burner (Maximum)	15	21½	15	15¾	17½
B—Overall width	18	25	17	17	18
C—Center of housing to end of motor	9¾	13½	8½	8½	9½
D—Center of housing to end of pump	8¼	11½	8½	8½	8½
E—Overall length (Minimum)	25¼	36	19¾*		
E—Overall length (9" tube)			21¾	21¾	24
F—Depth of housing	16¼	21½	12¾	12¾	15
G—Length of tube	9	14½	7 to 9	9	9
H—Diameter of tube	5¾	6⅝	4⅜	4⅝	5¾
I—Bottom of flange to center of tube (Minimum)	9	14	8½	8¾	10
I—Bottom of flange to center of tube (Maximum)	11	16	10½	10¾	12

*(7" tube)

H. C. Little Burner Co.

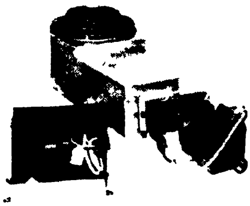
Head Office: **San Rafael, Calif.**

Branches in Principal Cities



Fully Automatic Oil Burning Floor Furnace

These low cost units are factory assembled, including all controls, ready to hang in floor. Easy installation, no basement needed. Natural draft vaporizing burner, automatic operation, with electric ignition and thermostatic control. Inexpensive to operate. Listed by Underwriters' Laboratories for No 3 oil (27° + Diesel oil). Two sizes: No **70-42**—47,250 Btu output (22 in. wide x 28 in long x 44 in high; No. **100-42**—75,000 Btu output (22 in. wide x 40 in. long x 45½ in. high).

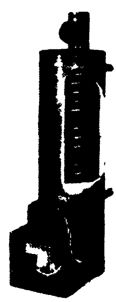


Natural Draft, Automatic, Vaporizing Burner

Natural draft vaporizing burner specifically designed for No 3 oil (27°+ Diesel oil) and listed by the Underwriters' Laboratories Model shown (with controls attached) is fully automatic, with electric ignition (no pilot light) and thermostatic control. Manually operated models, which do not require electricity, are also listed for No. 3 oil Seven sizes. 1 45 qts to 2¾ gal. per hour.

Aquatherm Water Heaters

Economical operation and ample capacity for all domestic requirements. Aquatherm "30" has 28 gal. storage, 40 g p.h. recovery rate, is 19 in. wide x 21 in. deep x 56 in Aquatherm "60" has 60 gals storage, 160 g.p.h. recovery, is 24½ in. wide x 36 in. deep x 68 in. Both units listed by Underwriters' Laboratories for No. 3 oil.



Coil Water Heater

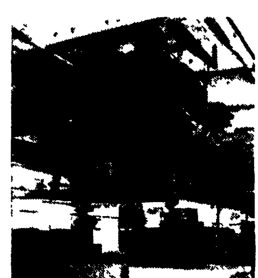
For camps, auto courts, apartments or small hotels. Uses separate hot water storage tank. Also makes excellent boiler for hot water installations in home heating. Two sizes: No. 150—106,960 Btu output, 150 g.p.h. (80° rise); No. 250—160,440 Btu output, 250 g p.h. (80° rise).

Winter Air Conditioner

Size A, 77,000 Btu output. Size B, 125,000 Btu output. Multivane blower, filters, vaporizing burner.

Ceiling Furnace

Industrial overhead heating saves floor space. Units are factory assembled, including vaporizing oil burner (no refractory brick saves much weight), thermostatic control, high capacity heat distributing fan with ¼ hp motor. Conforms to requirement of the Underwriters' Laboratories for use in garages.



A complete line of oil-fired small home heating units —plus special industrial units of moderate capacity.

S. T. Johnson Co.

940 Arlington Ave., **Oakland 8, Calif.**

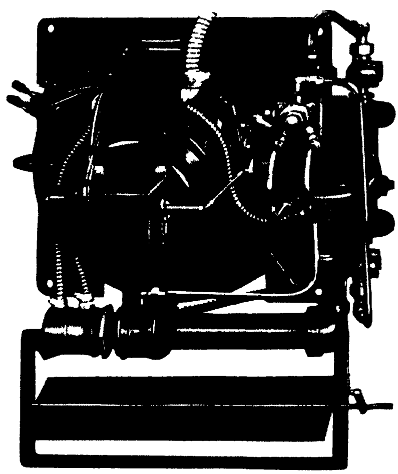
401 No. Broad St., **Philadelphia 8, Pa.**

Builders of Heavy-Duty Industrial Oil Burners

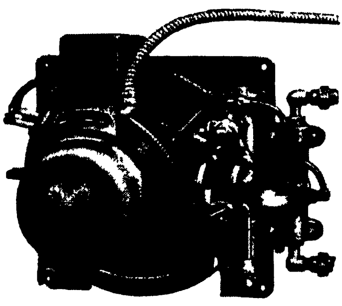
Johnson Industrial Burners are designed to operate on Heavy Oils which produce the greatest heat at the lowest cost. They increase the capacity of equipment formerly fired with coal and produce desired steam pressures more quickly. Automatic regulation permits the boiler to operate with maximum efficiency at any specified steam pressure, without watching, care or attention, thus reducing labor costs

They have been installed with marked success in hotels, hospitals, factories, office buildings and other large structures all over America because they provide heating engineers with a wide range of capacities and with every desired feature of economy, performance and automatic control.

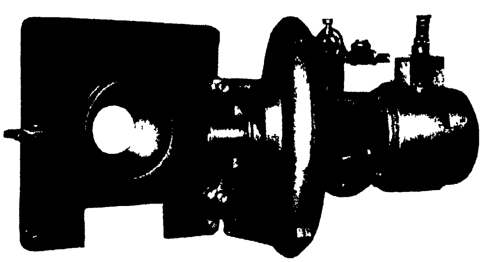
In their design and in their construction, Johnson Burners represent the most advanced engineering and building techniques backed by 41 years of practical experience.



TYPE 30 AVH
Fully automatic Pre-heater type Burns No 6 Oil. Six sizes, 20 to 300 g p h.



TYPE 30 AV
Fully automatic Burns No. 5 Oil Six sizes, 2 to 100 g p h.



TYPE 28
Manual and semi-automatic. With or without built-in pumps. Burns No 5 and No 6 Oils. Seven sizes, 2 to 135 g p h. Illustration shows burner swung away from fire-hole-plate for easy inspection.

S. T. Johnson Co.

Builders of Domestic and Commercial Oil Burners

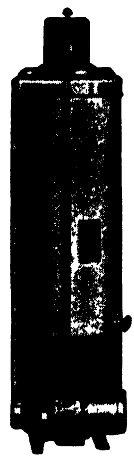
940 Arlington Ave., **Oakland 8, Calif.**

401 No Broad St., **Philadelphia 8, Pa.**

Self-storage water heaters, separate burner units, burner-boiler units, conditioned air units, range burners and various specialized items comprise the line-up of Johnson light-oil Burners.

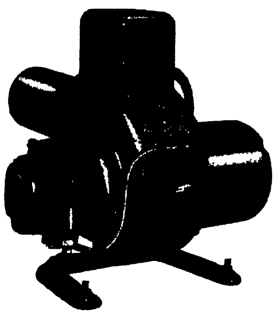
There is a wide range of sizes and capacities in each classification with which heating engineers and contractors can successfully meet every type of problem.

Every Johnson Burner is backed by an unbroken 41-year record of fine engineering and excellent craftsmanship.



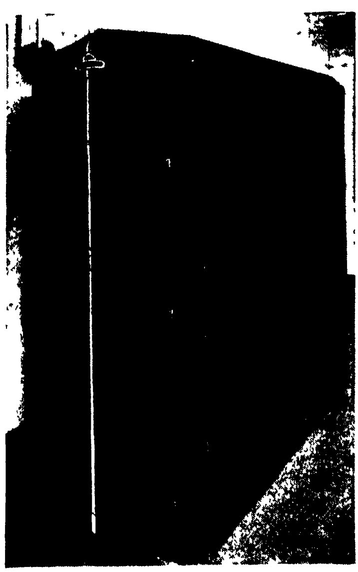
AQULUX WATER HEATERS

Fully automatic, self-storage Cap: 100 to 540 g p h.



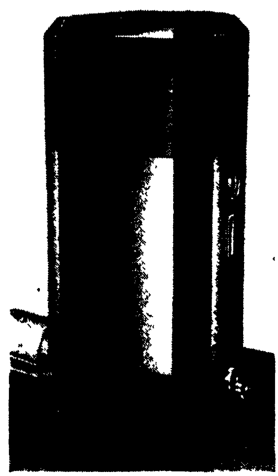
BANKHEAT BURNERS

Fully automatic, pressure atomizing type. Sizes up to 20 gal per hr



SELECTAIR HEATERS

Hot air, hot water or steam heat 3 sizes Bankheat Burners



ECONOLUX HEATERS

Heat and hot water. Fully automatic Bankheat Burners. 9 sizes

TODD COMBUSTION EQUIPMENT, INC.

(Division of Todd Shipyards Corporation)

601 West 26th Street, New York 1, N. Y.

NEW YORK MOBILE NEW ORLEANS GALVESTON SEATTLE BUENOS AIRES LONDON

THE TODD HEX-PRESS REGISTER in combination with the **TODD "VEE-CEE" VARIABLE CAPACITY BURNER . . .** makes possible increased combustion efficiency under almost any type of boiler of 100 hp. capacity or larger, operating at 50 pounds steam pressure or higher.

It provides equal efficiency under either forced or natural draft conditions. The Hex-Press Register assures the most intimate mixture of oil and air as well as quicker, more complete combustion . . . with minimum draft loss at high capacity . . . effecting great economy in maintenance and materially reducing fuel costs.

Through the exclusive "variable range" feature of the "Vee-Cee" Burner, practically unlimited firing range is assured . . . without change of burner tips, oil delivery pressure or angle of spray.

Constant steam pressure can be maintained regardless of demand . . . changing load requirements are met instantly under manual or fully automatic control.

COMBINATION GAS AND OIL BURNERS

For Natural or Refinery Gas and/or Fuel Oil. Available in wide range of capacities. Quickly adjustable for the combustion of either fuel alone, or both in combination. Of special value where fluctuating comparative costs of these fuels call for equipment suited to changeover without time-consuming structural changes.

Maintenance and operation are reduced to a minimum by compactness and simplicity of design . . . accessibility of all parts . . . rugged construction and positive overall efficiency.

Design features eliminate possibility of escaping gas due to structural distortion . . . prevent stratified combustion resulting from improper air distribution and high gas pressure.

Providing sufficient flexibility to care for varying loads, these units assure high furnace temperature and radiant heat transfer with low stack temperature . . . thorough mixture and optimum air-fuel ratio with utmost ease of adjustment.

ROTARY FUEL OIL BURNERS

For firing high or low pressure steam or hot water boilers of all types . . in smaller factories and industrial plants, laundries, dryers and cleaners, office buildings, hotels, apartment houses. Also applicable to industrial ovens, kilns, etc., where furnace and general physical conditions permit.

Available with manual, semi-automatic or fully automatic control . . . in varying sizes and types . . . for burning light or heavy oil.

Horizontal atomizing cup is rotated by direct-connected electric motor, assuring constant firing as long as motor is in operation. Motors are of extra large frame size, air-cooled and built to withstand long, hard service Positive air-oil interlocking device automatically shuts off oil supply following any burner stoppage.

Of rugged construction . . . with all parts easily accessible for cleaning or renewing . . . these burners provide a flexible capacity range, with complete and efficient combustion under widely fluctuating loads.

TODD MANUFACTURES: Mechanical Pressure Atomizing Oil Burners—VEE-CEE Variable Capacity Burners—Horizontal Rotary Oil Burners—Oil Burning Air Registers for Natural, Assisted, Induced or Forced Draft—Inside Mixing Steam Atomizing Oil Burners—Combination Gas and Oil Burners—Furnace Doors and Interior Castings for Converting Howden Type Furnace Fronts to oil firing—Oil Burning Galley Ranges—Oil Heating, Pumping and Straining Equipment.

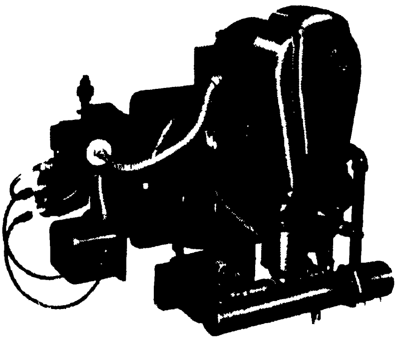
All installations of Todd Equipment are always *individually engineered* to fulfill specific requirements. *Send for descriptive literature.*

Todd engineers are always available for consultation and analysis of combustion problems—without obligation.

York-Shipley, Inc.

York, Pa.

YORK HEAT OIL-BURNING EQUIPMENT



Model AHPBDM Pump-Type Burner for Bunker C oil, belt-driven, with fully-automatic and full-modulating firing control

YORK HEAT INDUSTRIAL HORIZONTAL ROTARY TYPE OIL BURNERS

A range of sizes is available from 18 Boiler hp to 400 Boiler hp. These burners are built in both direct-drive and belt-driven motor applications. Models for using No. 5 and No. 6 oils are available for manual, semi-automatic, and full modulating control.

The basic frame sizes are listed as follows:

A Frame	18 hp to 60 hp
B Frame	60 hp to 125 hp
C Frame	125 hp to 225 hp
D Frame	225 hp to 400 hp

York Heat Horizontal Rotary Burners are equipped with many exclusive and patented advancements, designed to insure high efficiency and superior operation. Outstanding among these exclusive York Heat features are:

(*a*) The Iris Shutter, which gives absolute control of the burner air. It causes the metered volume of air to remain constant, insuring perfect combustion.

(*b*) The York Heat Flame Former, which makes possible accurate flame-adjustment, to suit the combustion chamber.

(*c*) The torch-type pressure igniter, for smoother and positive automatic starting.

YORK HEAT SELF-CONTAINED BOILER-BURNER UNITS FOR FUEL OIL

A complete line of fully-automatic self-contained boiler-burner units is available, in capacities of 4 hp to 100 hp. These are constructed for both 15 lb and 100 lb steam operation. Larger sizes are equipped with an oil-burner for No. 5 or No. 6 fuel oil.

Model N-15 York Heat Domestic Oil Burner

YORK HEAT DOMESTIC OIL-BURNERS

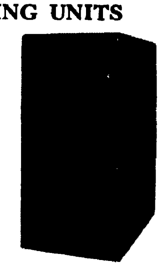
This complete line of pressure, atomizing-type burners carries the Underwriters' and Bureau of Standards approval for liquid fuels up to and including No. 3 fuel oil.

Models and capacities are listed as follows:

C-5	0.7 to 2.25 gph of oil
N-15	1½ to 4½ gph of oil
N-25	3 to 5½ gph of oil
N-35	5 to 10 gph of oil
T-45	8 to 18 gph of oil
T-55	16 to 30 gph of oil

YORK HEAT DOMESTIC HEATING UNITS

York Heat offers a complete line of domestic heating units, in a full range of capacities. These can be furnished for steam, hot water, and forced warm air applications.



York Heat Domestic Boiler-Burner Heating Unit

The Brownell Company

ESTABLISHED 1855

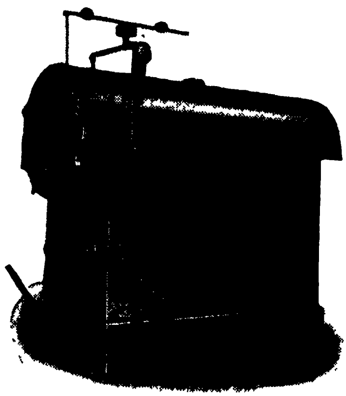
Dayton, Ohio

Manufacturers of

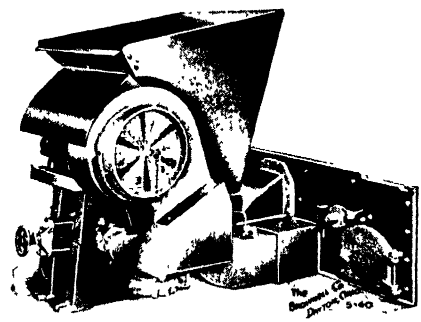
BROWNELL BOILERS AND STOKERS

Representatives in All Principal Cities

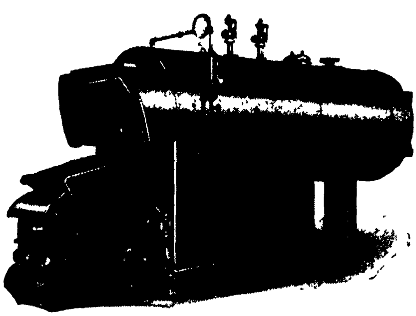
FIRE TUBE BOILERS of various types. HEATING BOILERS, riveted and welded. UNDERFEED STOKERS from 5 Horse Power up. STEEL STACKS, TANKS AND SPECIAL STEEL PLATE WORK.



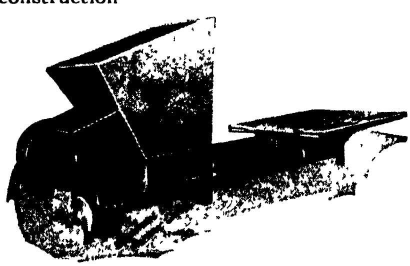
Welded Triple Pass Heating Boilers built in either high leg or low water line types. Hand fired ratings 500 to 35,500 sq. ft. steam, 800 to 56,800 sq. ft. water radiation. Stoker, Oil or Gas fired up to 43,100 sq. ft. steam or 69,000 sq. ft. water radiation. A.S.M.E. Code construction.



Type L R (Low Set) Underfeed Ram Type Stoker with automatic air volume control. Can be furnished with Brownell exclusive, fully automatic coal feed control. Sizes up to 300 horse power. An ideal stoker for firebox boiler or other installations where height of setting is limited.



High or Low Pressure Double Pass Boiler with Type L R Stoker. Designed and manufactured as a matched unit steam generating plant. Furnished in working pressures from 15 to 150 pounds and sizes up to 300 horse power. For power, heating and process steam. Steam ratings 3,600 to 42,500 sq. ft. Water rating 5,800 to 68,000 sq. ft. when used with stoker, oil or gas. A.S.M.E Code construction



Type C Screw Feed Stoker, proved by years of service to be sturdy, reliable and efficient. Illustration shows dead plates can also be furnished with dump plates in the larger sizes. 30-300 hp.

The illustrations above show only a part of the complete Brownell line. We shall gladly send literature describing BROWNELL BOILERS and STOKERS. Our nation wide field organization is ready to assist in problems of steam generation.

Combustion Engineering Company, Inc.

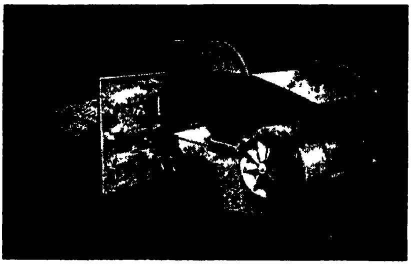
All Types of Fire Tube and Water Tube Boilers
Mechanical Stokers

Complete Steam Generating Units
Pulverized Fuel Systems

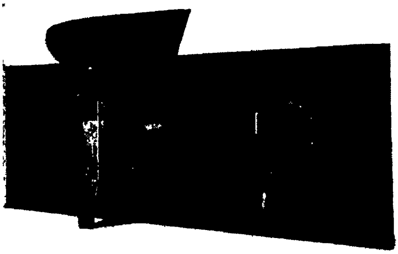
200 Madison Avenue, **New York 16, N. Y.**

Offices in all principal cities of the United States and Canada

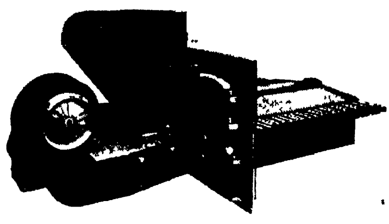
More than 18,000 C-E Stokers purchased to date



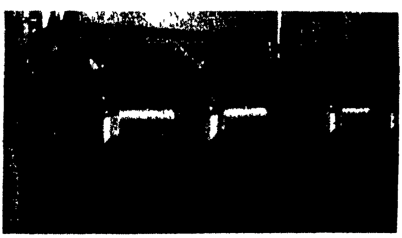
C-E Skelly Stoker Unit



Type E Stoker



C-E Low Ram Stoker



C-E Spreader Stoker

C-E Skelly Stoker Unit—A compact, self-contained unit with integral forced-draft fan, adapted to burn either anthracite or bituminous coal. Alternate fixed and moving grate bars assure lateral distribution of fuel. Automatic control is standard equipment. Approximate application range—20 to 200 rated boiler hp.

Type E Stoker—A single-retort, underfeed stoker with an established reputation of many years' standing for dependable service. Designed to burn a variety of bituminous coals under boilers up to about 600 rated hp. Available with steam, electric or hydraulic drive.

C-E Low Ram Stoker—A single-retort, stationary-grate underfeed stoker for burning bituminous coals under boilers in the upper size range of the C-E Skelly Stoker.

C-E Spreader Stoker—A simple, rugged overfeed stoker designed to burn a wide variety of coals. Fines are burned in suspension and the coarser coal on a grate which may be of either stationary or dumping type. Rate of coal feed and air supply may be regulated over a wide range and are readily adaptable to automatic control. Applicable to boilers from about 100 boiler hp up.

C-E Multiple Retort Stoker—For burning bituminous and semi-bituminous coals under boilers up to the largest sizes.

C-E Traveling Grate and Chain Grate Stokers—Including both Coxe and Green types. Available with grate surfaces suitable for anthracite, coke breeze, lignite or bituminous coal, as required. Traveling grates are all forced-draft types; chain grates are either forced or natural draft types.

C-E Boilers—All fire tube and water tube types in sizes ranging from 25 hp up to the largest. Standard and special designs to suit all conditions of fuel, load and space. Included are all types formerly known by the trade names "Heine," "Walsh & Weidner," "Casey-Hedges," "Ladd" and "Nuway".

Separate Catalogs describing each of these stokers are available. A-531-C

Detroit Stoker Company

Sales and Engineering Offices
General Motors Bldg., Detroit, Mich.

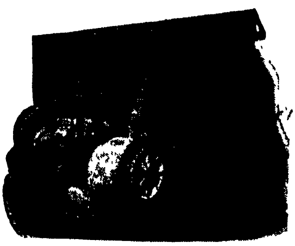
Main Offices and Works at Monroe, Mich.

Since 1898

District Offices in
Principal Cities

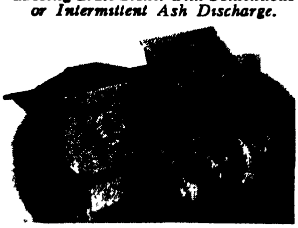
Built in Canada at London, Ont.

Detroit Stokers are unsurpassed for economy and dependability. They include Underfeed and Overfeed Stokers of many sizes and capacities for all types of boilers, All grades of Bituminous Coal successfully burned. Operating costs are low. Substantial, heavy designs represent over forty years' experience in Stoker manufacture. Catalogs of various types, furnished on request. Write us at Detroit or telephone nearest district office.



Detroit UniStoker with Detroit Adjustable Feed provides a wide range of coal feed control.

Detroit C-D Stoker is a Single Retort. Moving Grate Stoker with Continuous or Intermittent Ash Discharge.



Detroit Double Retort Stoker, a multiple retort side cleaning stoker for medium size boilers.

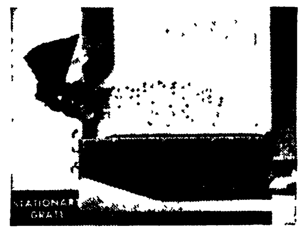
Detroit UniStoker with Detroit Adjustable Feed (Coal Feed Control) insures accurate fuel in proportion to the air supply for best economy. Single Retort, side cleaning for boilers approximately 150 to 300 horsepower with furnace widths from five feet six inches to eight feet.

C-D Stoker—Single Retort, moving grate stoker. Either continuous or intermittent ash discharge. Motor or steam ram driven. For boilers 300 to 500 horsepower having furnace widths seven and one-half to twelve feet.

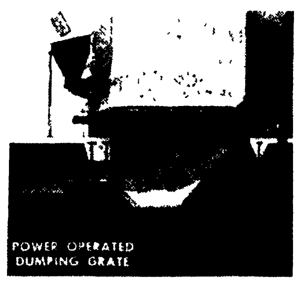
Double Retort Stoker—a multiple retort stoker having two retorts with the side cleaning feature. For medium sized boilers with wide furnaces

Detroit RotoStokers are Overfeed Spreader Stokers, having an Overthrow Rotor action, which insures uniform fuel distribution over the entire area. Offers advantages over other firing methods for burning inferior fuels and efficiently handling extremely fluctuating loads.

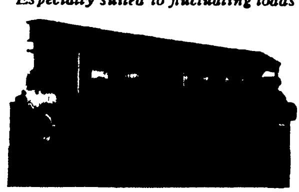
Detroit Multiple Retort Stoker for large boilers and high capacities. An inclined fuel bed Stoker, possessing all outstanding modern features.



Detroit RotoStoker Ash removed through doors at grate level successfully burns a wide range of fuels



Detroit RotoStoker (Either Power or Hand Operated) for large boilers. Especially suited to fluctuating loads



Detroit RotoGrate Stoker — Burns wide range of fuels. Fine particles burned in suspension, coarse coal on continuous cleaning forward moving grate. Combustible content of ash usually less than two per cent Fuels of high ash content burned with higher burning rates for long periods. Dependable operation for large boilers—high capacities

DETROIT LOSTOKER

Detroit LoStoker is a complete mechanical firing unit in many grate area sizes and capacities for application to all types of boilers from approximately 30 to 150 hp. Burns various grades of Bituminous Coal with high efficiency. Fuel is fed only when needed—none wasted. Single Retort, Side Cleaning, Adjustable Plunger Feed Type, mechanically driven from electric motor, requires little power for operation. Automatically controlled from steam pressure, water temperature or room thermostat. Compact, easily installed, responsive and automatic. A great coal saver.

ADVANTAGES:

Continuous Adjustable Plunger Feed with control of the quantity of coal fed and its distribution.

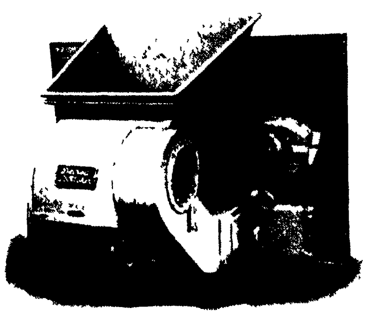
Heavy Mechanical Drive of simple design, requires little power.

Side Cleaning with dumping grates, ashes removed through doors provided in the Stoker front. No hand cleaning.

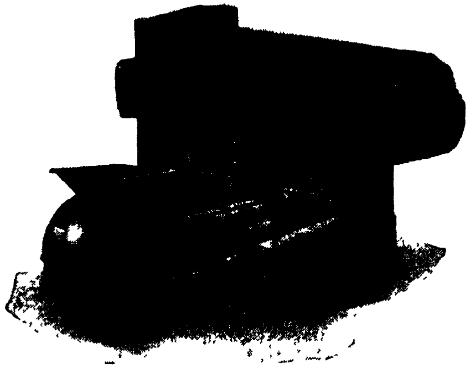
Agitator in coal hopper for continuous coal feed, cannot stick or jam with wet coal.

Automatically Controlled. Motor or steam turbine driven, controlled from steam pressure, water temperature or thermostat.

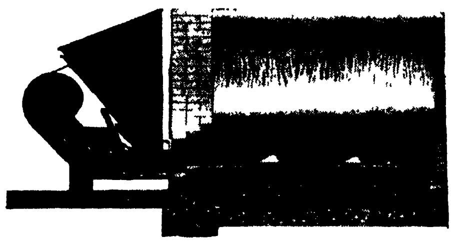
Many grate area sizes and capacities to fit the furnace and provide the proper grate area to readily handle heavy loads and also to operate efficiently under light load conditions.



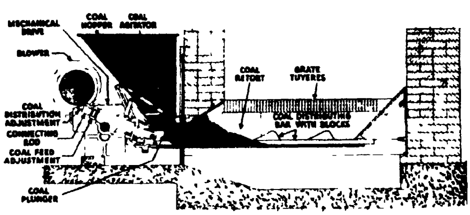
Detroit LoStoker (brickset type) for application to horizontal return tubular or water tube boilers



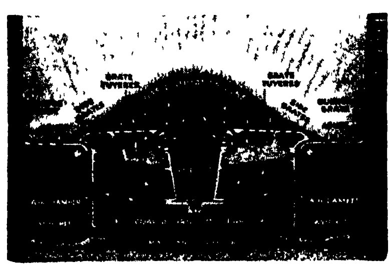
Detroit LoStoker readily applied to Firebox Boilers—built to fit the Furnace or Firebox. Coal Hopper with Agitator designed to clear Boiler Doors. Plunger Feed-side cleaning feature eliminates arduous hand cleaning of fires.



Detroit LoStoker (Side elevation in brick setting) for horizontal return tubular or water tube boilers.



Detroit LoStoker showing adjustable plunger feed.



Front Elevation of Detroit LoStoker (brickset type) built to fit the furnace. For use with horizontal tubular, firebox boilers on brick foundations or water tube boilers. Arrows indicate flow of air to fuel bed.

Iron Fireman Manufacturing Company

Automatic Coal Stokers

Portland, Oregon **Cleveland, Ohio**

Address inquiries to 3636 West 106 St., Cleveland 11, Ohio

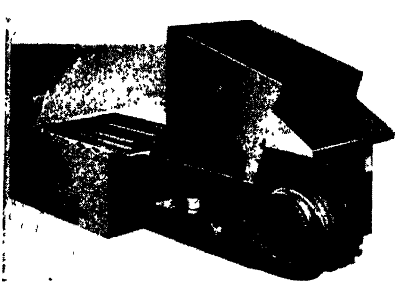
Retail Branches or Subsidiaries: CHICAGO, ILL.; MILWAUKEE, WIS ; ST LOUIS, MO.; NEW YORK, N. Y.; BROOKLYN, N. Y.; TORONTO, CANADA; MONTREAL, CANADA

Dealers in Principal Cities and Towns in the United States and Canada

Representation in numerous foreign countries

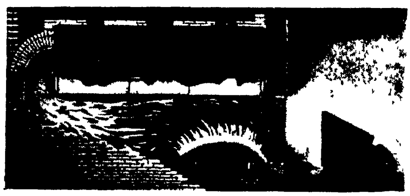
COMMERCIAL AND INDUSTRIAL STOKERS

STANDARD HOPPER MODELS



Commercial Installation—Hopper Model

This series of stokers is the standard of value in equipment for automatically firing boilers ranging in size up to 350 hp. Available in a wide range of coal feeding capacities, lengths and grate arrangements, to fit varied requirements.



Hopper Model Iron Fireman in Operation in Horizontal Return Tubular Boiler

STANDARD COAL FLOW MODELS

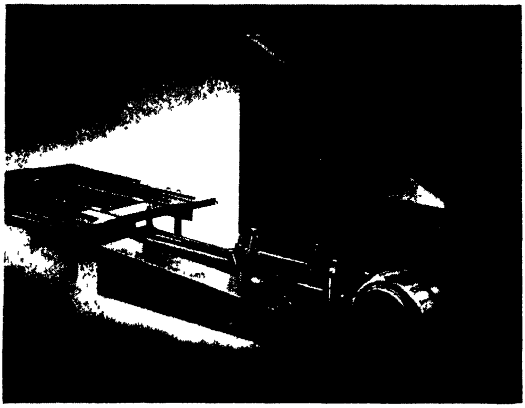
Commercial—Industrial Installation—Coal Flow model that carries coal direct from bunker to fire

A heavy duty stoker which combines Iron Fireman's well known firing efficiency with the automatic conveying of coal direct from bunker to fire. An integral coal conveying mechanism eliminates the labor and expense of manual coal handling The IRON FIREMAN COMMERCIAL-INDUSTRIAL COAL FLOW stoker fires boilers developing up to 350 horsepower. PNEUMATIC SPREADER STOKERS also available in Coal Flow models

MODEL	OUTPUT RANGE		
	Boiler Horsepower	Equivalent Direct Radiation	
		Steam (240 Btu)	Hot Water (150 Btu)
Coal Flow (available in all models)	3 to 500*	400 to 70,000	650 to 110,000
Commercial and Industrial Standard Underfeed	3 to 350	400 to 50,000	650 to 75,000
Commercial and Industrial Poweram Underfeed	30 to 400	4,000 to 56,000	6,000 to 90,000
Commercial Anthracite	30 to 130	4,000 to 18,000	6,000 to 29,000
Pneumatic Spreader. .	50 to 1,000*	7,000 to 140,000	11,000 to 225,000

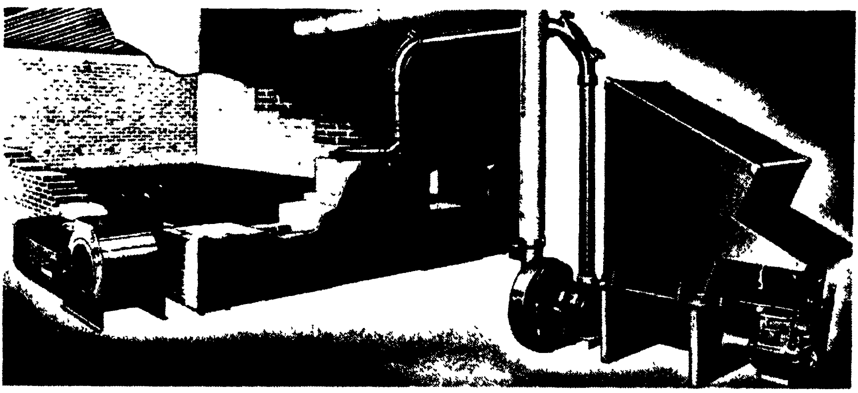
*Multiple units available for larger boilers.

POWERAM STOKERS



Combines ram-type coal distributor system in retort with free-running worm conveyor from coal supply, and has these advantages: Delivers the fuel to the fire bed in a loose, easily aerated condition; the reciprocating pusher blocks insure proper fuel distribution, and make possible the successful and efficient burning of many types of coal which otherwise are impractical to fire automatically with underfeed stokers. Designed for boilers developing up to 400 hp.

PNEUMATIC SPREADER STOKERS



As shown in the illustration above, the IRON FIREMAN PNEUMATIC SPREADER STOKER meters steam size coal from hopper or main coal bunker to transfer housing, where coal is picked up by pneumatic conveyor and delivered to furnace. The conveyor nozzle accurately spreads the larger particles of coal over the entire grate in a shallow, uniform fuel bed. The preheated fines burn in suspension, reducing the cinder carry-over and greatly improving the combustion efficiency and responsiveness, as compared with other stokers which do not preheat fuel. The conveying air provides the overfire air which is essential for efficient combustion. Entering at right angles to the flow of burning gases from the fuel bed, the conveying air produces *maximum turbulence, another requisite of efficient and smokeless combustion.* THE IRON FIREMAN PNEUMATIC SPREADER STOKER was designed to burn efficiently such economical fuels as the lower rank bituminous, and sub-bituminous coals and also lignite. It provides reliability of operation, physical adaptability, ease of operation, and low maintenance which is not afforded by other types of automatic coal burning systems. Pneumatic Spreader stokers are particularly adaptable to operation at high ratings, and as a result are greatly stepping up steam output in many plants throughout the United States and Canada. IRON FIREMAN PNEUMATIC SPREADER STOKERS are made in both hopper and Coal Flow models; the latter carry coal direct from the bunker to the fire.

AMERICAN RADIATOR & Standard Sanitary
New York CORPORATION Pittsburgh

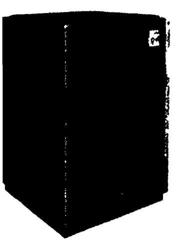
WAR TIME RESTRICTIONS

The manufacture of certain Boilers and Radiators and the complete line of Sunbeam Warm Air Furnaces and Winter Air Conditioners has been restricted temporarily by WPB Limitation Orders, and by urgent demands of war production. However, with one or two exceptions, the products shown on these pages are available for essential needs, but in most cases boilers must be furnished without jackets, probably for the duration.

We invite inquiries as to the status of any heating products under the conditions prevailing at time of inquiry.

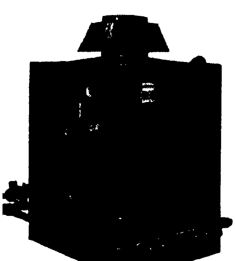
SEVERN BOILER FOR COAL (stoker or hand-fired), or OIL

An exceptionally efficient Boiler with many new features for convenience and economy. Ratings: Steam—350 to 780 sq ft, Water—560 to 1250 sq ft, installed radiation.



EMPIRE GAS BOILER

Designed by experts to burn gas efficiently, economically. All controls concealed. Ratings: Steam—163 to 1097 sq ft, Water—135 to 1755 sq ft, installed radiation.

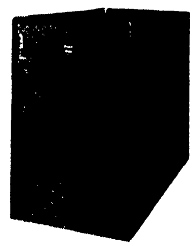


STANDARD GAS BOILER

Basically the same as the Empire Gas Boiler shown above but without jacket. Ratings: Steam — 400 to 11,905 sq ft, Water — 135 to 19,048 sq ft, installed radiation.

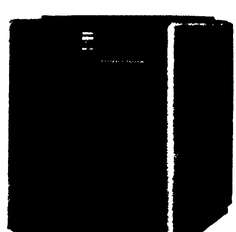
ARCOLINER FOR COAL (stoker or hand-fired) or OIL

An attractive Boiler of advanced design for heating smaller homes inexpensively and well. Ratings: Steam—180 to 460 sq ft, Water—290 to 740 sq ft, installed radiation.



OAKMONT OIL BOILER

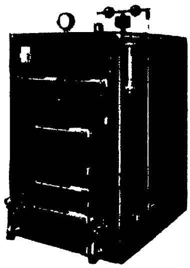
A highly efficient moderate priced Boiler for small homes. Also supplied as complete boiler-burner unit with Arcoflame Burner. Ratings: Steam—390 to 810 sq ft, Water—625 to 1295 sq ft, installed radiation.



MOHAWK Winter Air Conditioner Gas-Fired

Provides complete, automatic gas-fired Winter Air Conditioning in small, medium or large sized homes. Capacities range from 60,000 to 300,000 Btu input per hour.

AMERICAN RADIATOR & Standard Sanitary
New York CORPORATION Pittsburgh



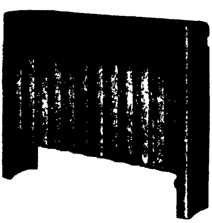
IDEAL REDFLASH BOILERS—For Coal (hand-fired)

Economical heat for any size or kind of building. Attractive red jacket, fully insulated. Ratings: Steam—770 to 9900 sq ft, Water—1230 to 15,840 sq ft, installed radiation.

Arco Radiator

ARCO RADIATOR

The modern, slim type radiator that occupies less space and gives more heat. It comes in four narrow widths and in four heights—19, 22, 25 and 32 inches.

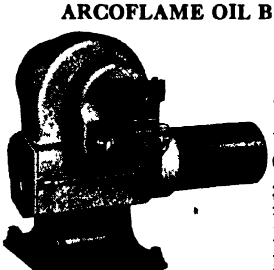


Sunrad

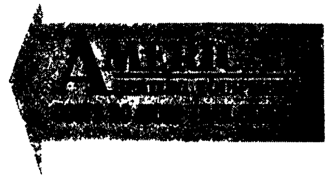
SUNRAD RADIATOR

Installed recessed or free-standing. Requires no enclosure. Combines radiant and convected heat. Sections come in two sizes: 20 in. high x 5 in. wide = 2 sq ft, and 23 in. x 7½ in. = 3 sq ft capacity.

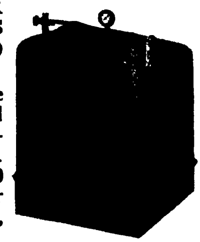
ARCOFLAME OIL BURNERS



The Model "C" Arcoflame has a capacity of up to 3 gallons per hour. The Model "L" (not shown) from 3 to 7 gallons per hour. Both embody unusual and highly efficient features.



IDEAL WATER TUBE BOILERS Coal (stoker fired) or Oil



For medium to large size buildings. Noted for efficient performance and economy. Ratings: Steam—930 to 4,600 sq ft, Water—1,490 to 7,360 sq ft, installed radiation.

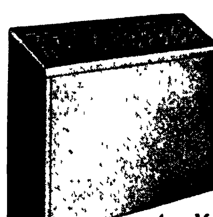
ARCO CONVECTOR

For convection heating at its best. Available in four widths and in virtually any desired length.

Arco Convector

ARCO MARINE CONVECTOR

Especially designed for Marine use. Non-ferrous. Highly efficient. For all systems except one-pipe steam.



Arco Marine Convector

No. 861 Arco Detroit Hurivent Valve (for main)

No. 300 Arco-Detroit Multiport Valve (for radiators)

No. 999 Arco Packless Steam Radiator Valve

THE BABCOCK & WILCOX COMPANY

85 Liberty Street **New York 6, N. Y.**

Manufacturers of

Water-Tube Boilers
Oil Burners

Chain-Grate Stokers
Seamless Steel Tubing and Pipe

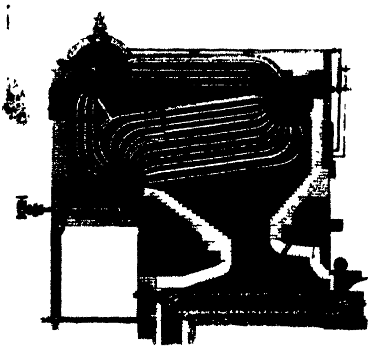
Branch Offices and Representatives in all Principal Cities

Type H Stirling Boiler

The Babcock & Wilcox Type H Stirling Boiler is a highly efficient unit built for moderate pressures at moderate prices.... and is designed to occupy minimum floor space and head room for the heating surface required.

This boiler is built in four classes and 36 sizes ranging from 691 to 6225 sq ft of heating surface, and can be designed for operation with any fuel and every method of firing.

The moderate price is due only to the simplicity of design, efficient production methods and superior shop equipment.



Type H Stirling Boiler with Babcock & Wilcox Chain-Grate Stoker

Advantages of the Babcock & Wilcox Type H Stirling Boiler:

Unusual steaming capacity for the floor space and head-room required.

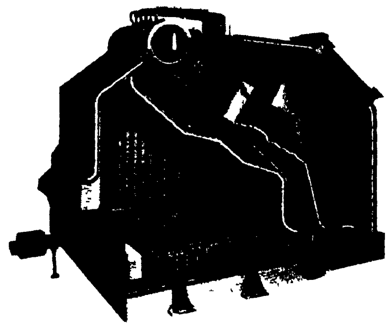
The choice of three locations for gas exit reduces cost of flues and breeching.

Distribution baffles make effective all of the heating surface.

Tube renewal is facilitated by correct tube spacing, and a tube removal door.

The boiler is supported by a structural-steel framework entirely independent of the brickwork.

A complete table of sizes and dimensions will be sent upon request. Simply ask for Bulletin G-8-C.



B&W Integral-Furnace Boiler, Type FF

Many of the advantageous features incorporated in large B&W central-station boilers are now available for the first time in the B&W Integral-Furnace Boiler, Type FF, which is offered in sizes ranging from 1353 to 6506 sq ft heating surface.

Distinguishing features include:

A completely water-cooled furnace. The construction provides water cooling for front and rear (or bridge) walls, as well as side walls and roof.

A furnace arrangement in which the primary combustion zone is followed by an open pass, thus making use of a principle of combustion that was first developed and used successfully in the B&W Open-Pass Boiler for central stations. This design insures mixing of the gases while at high temperatures, thereby aiding efficient and smokeless combustion.

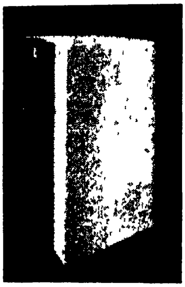
Cyclone Steam Separators, which provide dry steam at high boiler-water concentrations independently of normal variations in water level, and increase circulation by eliminating steam from the water.

These, with related features, result in a boiler that is outstanding for economy of fuel and maintenance and for ease of operation. Write for Bulletin G-34.

Burnham Boiler Corporation

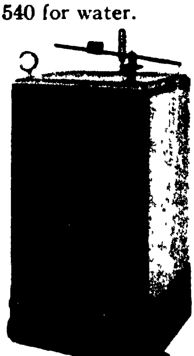
Irvington-on-Hudson, N. Y.—Zanesville, Ohio

***There's a Burnham for Every Purpose*—Catalogs Sent on Request**

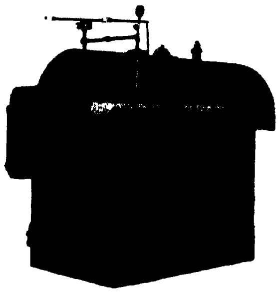


Yellow Jacket Boiler—All Fuel Convertible. 305 to 935 sq ft for steam and 490 to 1495 for water.

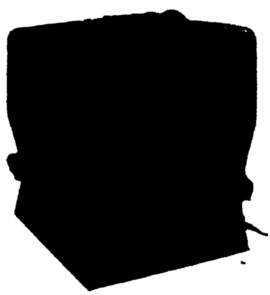
DeLuxe Gas or Oil Boiler—250 to 960 sq ft steam and 415 to 1540 for water.



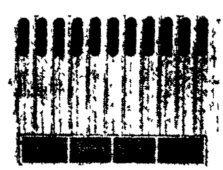
No. 1, 2, 3 and 36 in. Series—All Fuel. 230 to 4920 sq ft steam and 370 to 7880 for water.



Welded Steel Boiler — Residence type with jackets. Capacities from 400 to 1750 sq ft steam and 640 to 2800 for water. Commercial type. Capacities from 1800 to 42,500



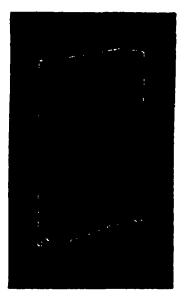
50 Inch Twin Section—4500 to 14,600 sq ft steam and 7200 to 23,360 for water.



Cabinet Type Radiant Radiator — Two heights. 20 and 23 inches.

Junior Yello-Jacket—For Oil Only. 360 sq ft steam and 580 sq ft for water.

Round Sectional, All Fuel—275 to 830 sq ft steam and 440 to 1330 for water.



Burnham Slenderized Radiator—Made in 3 to 7 tubes in heights of 14 to 32 inches.

Crane Co.

BOILERS, RADIATORS, FURNACES, VALVES, FITTINGS, PIPE, STEAM SPECIALTIES, PLUMBING AND HEATING MATERIALS

General Offices: 836 South Michigan Avenue, **Chicago 5, Illinois**

Nation-Wide Service Through Branches, Wholesalers, Plumbing and Heating Contractors

A STATEMENT REGARDING CRANE CO'S. POSTWAR HEATING LINE

WHILE THE WAR has placed severe restrictions upon the manufacture of heating equipment, it also provided the opportunity for extensive development and refinement of heating equipment designs. In Crane laboratories and engineering departments, the war years have been used toward that end to the best advantage.

AS A RESULT, when war restrictions are lifted, Crane will be ready with the most complete and advanced heating line in its history, for all residential installations, large and small, for apartment buildings, schools, stores, hospitals, churches, etc.

INCLUDED WILL BE everything for every heating system—for any type fuel:

1. **Steam and Hot Water Boilers.**
2. **Gravity and Forced Circulation Warm Air Furnaces.**
3. **Oil and Gas Boiler-Burner Units.**
4. **Stokers, Oil Burners, Unit Heaters.**
5. **Controls for Every Purpose.**
6. **Newly Developed Crane Radiant Baseboard Panel, Modern Slim-Tube Radiators, Convectors.**
7. **Valves, Fittings, Pipe and Piping Accessories.**

WHEN THE "GO" SIGN is given, Crane Co. will announce complete details of this new line. It will solve every heating problem and insure the kind of performance intended in every specification. Keep in touch with your Crane Branch or Wholesaler.

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ALABAMA

Birmingham	2 S. Twentieth St.
Mobile	300 N. Royal St.

ARIZONA

Phoenix	233 S. First Ave.
Tucson	35 E. Toole Ave.

ARKANSAS

Ft. Smith	221 S. Ninth St.
Little Rock	120 Commerce St

CALIFORNIA

Long Beach	824 W. Anaheim St.
Los Angeles	321 E. Third St.
Oakland	346 Ninth St.
Sacramento	1227 Front St.
San Bernardino	110 South "E" St.
San Diego	1138 India St.
San Francisco	301 Brannan St.
Santa Ana	919 Poinsettia St.

COLORADO

Denver*	1631 Fifteenth St.
Grand Junction*	601 Pitkin Ave.
Pueblo*	216 W. Third St.

CONNECTICUT

Bridgeport	302 John St
Hartford	710 Windsor St.
New Haven	365 Orchard St.

DISTRICT OF COLUMBIA

Washington	1225 Eye St , N.W.

FLORIDA

Jacksonville	1007 W Bay St.
Miami	85 N.W. Tenth St.
Tampa	1405 Twiggs St.
West Palm Beach	601 Lantana Ave.

GEORGIA

Atlanta	Washington St Viaduct
Macon	500 Broadway
Savannah	14 W. Broad St.

IDAHO

Boise	214 S Fifth St.
Pocatello	756 S. First Ave.

ILLINOIS

Aurora	544 S. Lake St.
Chicago	156 N. Jefferson St.
Southside Branch	231 West 79th St.
Rockford	800 S. Main St.
Springfield	921 E. Monroe St.
Waukegan	1219 Glen Rock Ave

*(Crane-O'Fallon Co.)

LIST OF CRANE BRANCHES (Cont.)

INDIANA

East Chicago ... 1004 Chicago Ave.
Evansville ... 118 S.E. Eighth St.
Indianapolis ... 333 W. Market St.
Muncie ... 342 N. Pershing Dr.
South Bend ... 1616 S. Franklin St.
Terre Haute ... 209 N. Ninth St.

IOWA

Davenport ... 217 E. Second St.
Sioux City ... 223 Jackson St.

KANSAS

Wichita ... 624 E. First St.

KENTUCKY

Lexington ... 165 Midland Ave.

LOUISIANA

New Orleans ... 1148 S. Peters St.
Shreveport ... 1320 Winston St.

MAINE

Portland ... 70 St. John St.

MARYLAND

Baltimore ... 626 W Pratt St

MASSACHUSETTS

Boston 48 W. First St., South Boston
Springfield ... 60 Cypress St

MICHIGAN

Detroit ... 150 Randolph St.
Flint ... 155 Lewis St
Grand Rapids 56 Grandville Ave, S W.
Jackson ... 409 Hupp Ave.

MINNESOTA

Duluth 102 Lake Ave., S.
Mankato ... 510 S Pike St.
Minneapolis 400 Third Ave, N.
St. Paul ... 356 Broadway

MISSISSIPPI

Jackson ... 176 N Gallatin St

MISSOURI

Kansas City ... 1328 W. Twelfth St.
St. Louis ... 30 S Sixteenth St

MONTANA

Billings ... 30th St and First Ave., S
Great Falls ... 2 Third Ave, S

NEBRASKA

Grand Island 115 E Front St
Omaha 323 S Tenth St
Scotts Bluff* 1617 Avenue A

NEVADA

Reno 401 E Fourth St.

NEW JERSEY

Newark ... 90 South St.
Trenton ... 50 Escher St.

NEW MEXICO

Albuquerque* ... 612 N. First St.

NEW YORK

Albany ... North Broadway
Binghamton ... 21 Washington St.
Buffalo ... 201 Church St
Hempstead 209 Main St.
New York, 4730 Twenty-Ninth St., Long Island City
Rochester ... 200 South Ave.
Syracuse ... 760 W. Genesee St.
Utica ... 326 Broad St

NORTH CAROLINA

Asheville ... 531 McDowell St.
Charlotte ... 1307 W. Morehead St.
Greensboro ... 205 S. Lyndon St.

NORTH DAKOTA

Fargo ... 636 Northern Pacific Ave.

OHIO

Akron ... 425 S. High St.
Canton ... 210 Piedmont Ave., S.E.
Cincinnati ... 824 Broadway
Cleveland ... 6215 Carnegie Ave.
Columbus ... 67 N. Front St.
Dayton ... 901 S. Perry St.
Lima ... 207 S. Central Ave.
Toledo ... 384 S. Erie St.

OKLAHOMA

Muskogee ... 302 Commercial St.
Oklahoma City ... 705 W. Main St.
Tulsa ... 623 E. Third St.

OREGON

Portland ... 710 N.W. Fourteenth Ave.

PENNSYLVANIA

Philadelphia ... 245 Master St.
Pittsburgh ... 40 Twenty-Fourth St.
Reading ... 407 N. Front St.

RHODE ISLAND

Providence ... 245 W. Exchange St.

SOUTH CAROLINA

Greenville ... 223 W. McBee Ave.

SOUTH DAKOTA

Aberdeen ... 324 Railroad Ave., E.
Sioux Falls ... 326 E. Eighth St.

TENNESSEE

Chattanooga ... 1317 Chestnut St.
Knoxville ... 523 W Jackson Ave.
Memphis ... 254 Court Ave.
Nashville ... 532 Eighth Ave., S.

TEXAS

Beaumont ... 720 Fannin St.
Corpus Christi ... 1212 N. Tancahua St
Dallas ... 814 Young St.
El Paso* ... 1609 Texas St.
Harlingen ... 201 West "C" St.
Houston ... 2205 McKinney Ave.
San Antonio ... 1200 E. Houston St.

UTAH

Ogden ... Twentieth St. and Wall Ave.
Salt Lake City ... 307 W. Second, South

VIRGINIA

Norfolk ... 2 E. Twenty-Second St.
Richmond ... 1223 W. Broad St.

WASHINGTON

Seattle ... 419 Second Ave., S.
Spokane ... 126 S. Post St.
Tacoma ... 1209 South "A" St.

WEST VIRGINIA

Charleston ... 502 Broad St.

WISCONSIN

Madison ... 641 Williamson St.
Milwaukee ... 321 E. Chicago St.
Oshkosh ... 37 Market St.

WYOMING

Casper* ... 802 East "C" St.

*(Crane-O'Fallon Co.)

NATION WIDE SERVICE THROUGH BRANCHES, WHOLESALERS, PLUMBING AND HEATING CONTRACTORS

Fitzgibbons Boiler Company, Inc.

Established 1886

General Offices: Architects Bldg., 101 Park Avenue

New York 17, N. Y.

Works: OSWEGO, N Y

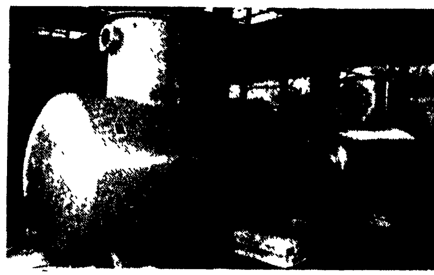
Branches and Representatives in Principal Cities

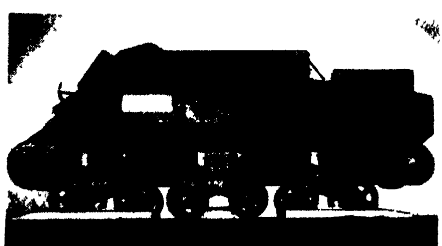
PRODUCTS—STEEL HEATING and POWER BOILERS for all fuels and all heating systems. Capacities to meet requirements of any building. Built and rated according to *A.S.M.E.* and *S.H.B.I.* Codes.—AIR CONDITIONERS for "Split-Systems" and for Direct-Fired installations in residences of all sizes.

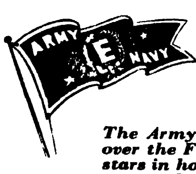
Member, Oil Heat Institute and Indoor Climate Institute.

As in the case with all of the country's industrial leaders, the full capacity, employment, facilities, and engineering skill of Fitzgibbons has been enrolled in the service of the United States, for the production of vital equipment used by the U. S. Army Ordnance, Transportation Corps, U. S. Navy, the Maritime Commission, and Lend-Lease. Fitzgibbons is proud to record this recognition of its outstanding skill and experience in engineering and manufacturing production, and to carry out to the utmost its determination to add all the force of its modern shops to the country's Axis-smashing effort.

The equipment built by Fitzgibbons in this program includes such front line fighting units as hulls for M-3 and M-4 tanks, and for M-7 tank destroyers, as well as gun platforms, gun shields and other fabricated parts for aircraft carriers. Also locomotive boilers for the U. S. Army and Russia, equipment for synthetic rubber manufacture and high octane gasoline production. And of course, many steel boilers for hospitals, cantonments and bases.



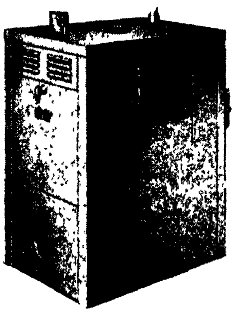




The Army-Navy "E" flag which flies proudly over the Fitzgibbons plant at Oswego, carries stars in honor of the continued success of the men and women of Fitzgibbons in their determination to produce the materials of mechanized war in the quantities that will bring peace to the farthest corners of the globe.

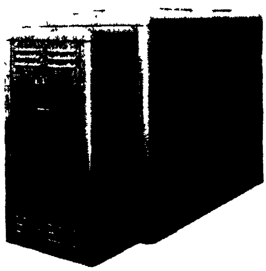
Coming—New Developments in Steel Heating Boilers and Airconditioners

Rather than make premature detailed announcement, Fitzgibbons prefers to complete the final check in laboratory and service tests on new equipment now pending. This equipment has behind it the full weight of Fitzgibbons invaluable experience and engineering skill, plus the manufacturing ability perfected in the fire of speedy wartime production of highest standard fighting equipment. Until the new Fitzgibbons products are placed upon the market, we are describing below the Steel Boilers and Airconditioners which have made the name Fitzgibbons known for fuel economy, and satisfactory comfort in thousands of homes.



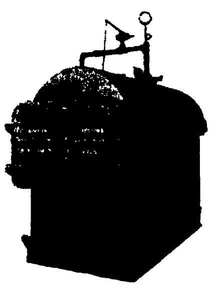
Steel Heating Boiler 400 Series

Typical FITZGIBBONS Residential Steel Heating Boilers and Airconditioners



Direct-Fired Airconditioner The "Directaire"

These modern all welded steel units are bringing comfort to American homes everywhere. Fitzgibbons steel boilers use the Fitzgibbons principle of a single pass of many small tubes, which assures highest operating efficiencies, results in important fuel savings, and makes cleaning easy. Fitzgibbons air conditioners enjoy the superior advantages of Fitzgibbons skill in design and fabricating, insure positively leakproof operation and give the maximum of comfort per unit of fuel.



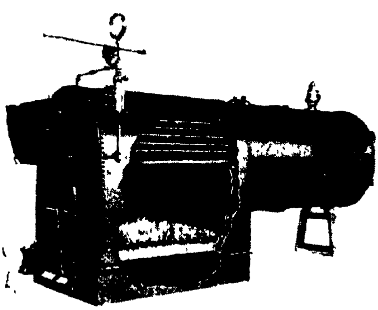
"D" Type

FITZGIBBONS "D" TYPE

Welded Steel Firebox Boilers

The "D" Type arranged for rear smoke outlet. Built for 15 lb w.s.p.—*A.S.M.E.* Code.

Up-Draft Type	1800 to 35,000 sq ft steam
Oil, Gas, Stoker	2190 to 42,500 sq ft steam
Smokeless Type	1800 to 35,000 sq ft steam



700 and "P" Series

FITZGIBBONS "P" SERIES

Portable Riveted Firebox Boilers

For 100 and 125 lb w.s.p —*A.S.M.E.* Code.

Ratings, horsepower—25 to 250 hand fired.
30 to 261 mech. fired.

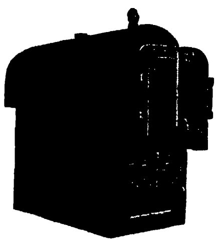
Oil, Gas, Stoker, and hand-fired types.

Farrar & Trefts

Incorporated

20 Milburn Street, **Buffalo 12, N. Y.**

Atlanta, Ga.	Grand Rapids, Mich	New York, N Y	St Louis, Mo
Buenos Aires, S A.	Honolulu, Hawaii	Nutley, N J	San Antonio, Texas
Cambridge, Mass	Milwaukee, Wis	Philadelphia, Pa	San Francisco, Calif
Charlotte, N C	Minneapolis, Minn	Pittsburgh, Pa	Tampa, Fla
Chicago, Ill	Nashville, Tenn	Richmond, Va	Toledo, Ohio
Cleveland, Ohio	New Orleans, La.	Rochester, N Y	Washington, D C
Detroit, Mich			Wilmington, Calif



The Bison Compact Boiler Series 100 and 900

The **F&T Bison Compact Welded Heating Boiler** is more than just another boiler. It has been designed carefully so as to have a large furnace volume, the proper volume of water, just the right amount of steam liberating surface, the correct volume for steam storage and a balanced circulation. The result is a remarkably steady water line—**A Balanced Boiler.**

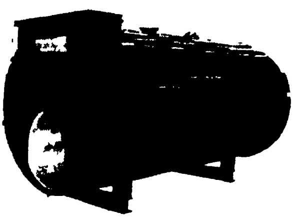
This boiler requires a minimum amount of floor space and is easy and inexpensive to install. It is reasonable as to first cost and economical in operation. Construction is in accordance with the *A.S.M.E.* Code for 15 lb working pressure and boilers are designed for hand firing with anthracite or bituminous coal or for mechanical firing with oil, gas or stoker. There are various sizes available from 1,800 to 35,000 sq ft of steam radiation, all ratings as required by the *Steel Heating Boiler Institute.*

The **Bisonette Compact Boiler** has the same characteristics as the larger Bison Compact Boiler. It has been designed for installation in large residences and small business establishments where the advantages inherent in a **Steel** boiler are desired

Firebox Return Tubular Boiler Series 500 and 600

Firebox Return Tubular Heating Boilers are **Quality Boilers.** They are constructed to measure up to the high standards set by Heating Engineers and will give unfailing service under all conditions Being economical to install and operate, they are highly favored by Architects and Engineers for heating Schools, Hospitals, etc

There are two types of Firebox Boilers, the Up-Draft Type and the Down-Draft Type. Both types are made of welded or riveted construction for heating purposes at 15 lb working pressure and riveted, or, Class 1 fusion welded x-rayed and stress-relieved for power purposes at 100, 125 and 150 lb working pressure in accordance with the *A.S.M.E.* Code Sizes from 4 500 to 35,000 sq ft of steam radiation, as rated by the *Steel Heating Boiler Institute*, are designed for hand firing with coal or for mechanical firing with oil, gas or stoker



Scotch Dry Back Boiler Series S-200

The **Bison Low Pressure Scotch Dry Back Boilers** are *carefully proportioned* and *balanced.* They are designed for hand, oil, gas or stoker firing, for ratings from 15 to 250 hp These boilers operate efficiently and carry sustained overloads. The *Front Smokebox Doors Open Sideways* giving easy access to the tubes.

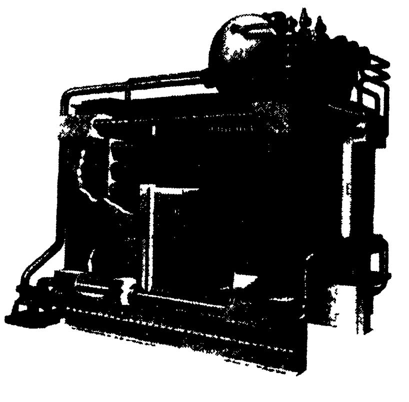
F & T boilers are designed so that the round furnace is always longer than the tube length which increases the furnace volume. This gives a large combustion volume in proportion to horsepower rating which makes the boilers very *economical to operate* and exceedingly "*Quick Steamers.*"

"Catalogs on Request"

The International Boiler Works Company

350 Birch St., **East Stroudsburg, Pa.**

SALES OFFICES IN PRINCIPAL CITIES



INTERNATIONAL-LAMONT Boilers Type LFS*, are designed for firing with oil, gas, pulverized coal or stokers. They are available in capacities from 4,000 to 30,000 lb of steam per hour and in all standard working pressures. Designs for larger sizes and special pressures or superheat requirements on request.

LFS boiler units provide power and process plants with high rates of heat transfer, greater steam production in the same space, quicker response to changing loads, a wider margin of operating safety, reduced maintenance and a lower overall steam cost.

Distinctive advantages of LFS boilers are:

The boiler is in full circulation from zero to maximum load.

The steam drum is an unfired pressure vessel and can be located below the top of the boiler structure or supported on the wall of the boiler room, where the headroom is limited.

Shop tested sub-assemblies permit quick erection on the job

Increased Capacity from Existing Boiler Installations

The furnace water-wall prefabricated and tested elements of the LFS boiler may be added as an external furnace to existing boilers When so applied the furnace is known as an LFB Boiler Booster

PACKAGED UNITS

Forced recirculation units from 20 to 100 boiler horse power, shipped complete for connection to electric, fuel oil and fresh water supply line

International "FUEL-SAVER" Water Tube Steel Heating Boilers offer the same quick steaming and economy that have long been accepted as most efficient in marine and industrial service "FUEL-SAVER" Water Tube Boilers are available for large and small heating requirements in a wide range of types and capacities **For Office and Apartment Buildings, Schools, Hotels, Theaters, Institutions and Industrial Plants.**

TYPE C

18 standardized sizes:
2680 to 42,500 sq ft,
mechanically fired
2200 to 35,000 sq ft,
hand fired.

Water tube design assures full absorption of intense heat of modern firing units. Efficiency is maintained under loads considerably above rated capacities Quick steaming Easy to keep clean

TYPE KD

15 standardized sizes:
5850 to 56,470 sq ft,
mechanically fired
4810 to 46,510 sq ft,
hand fired

Designed for replacements in large buildings Delivered disassembled to boiler room to avoid costly alterations. International erects or assumes responsibility for erection work.

TYPES DD AND K - FOR RESIDENTIAL HEATING AND HOT WATER SUPPLY

The tube arrangement assures thorough distribution of hot gases and rapid circulation allows quick heat transfer. A submerged oversized copper coil generates adequate hot water for continuous home use without need for storage tanks. Type DD is suitable in the modern split system of air conditioning. Type K has the added feature of an emergency wood or coal compartment for use when oil supply fails and for every day use as an incinerator.

*Licensed under LaMont patents.

KEWANEE BOILER CORPORATION

Kewanee, Illinois

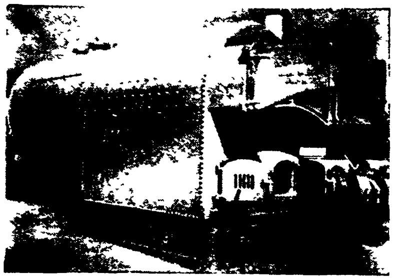
BRANCHES IN 64 PRINCIPAL CITIES

Steel Heating and Power Boilers, Water Heating Garbage Burners, Tabasco Heaters and Tanks.

KEWANEE STEEL HEATING BOILERS

Kewanee offers a dependable line of Steel Boilers built for heating every size building with high efficiency burning any kind of fuel. There are 380 standard sizes and 33 types of Kewanee Boilers most of which are kept in stock ready for immediate delivery.

Seventy-seven years of intensive study and effort are back of Kewanee Boiler designs. They are all constructed in our extensively equipped factory at Kewanee, Illinois, in conformity with these Codes: *American Society of Mechanical Engineers* for construction, and for rating with the *Steel Heating Boiler Institute* Simplified Practice



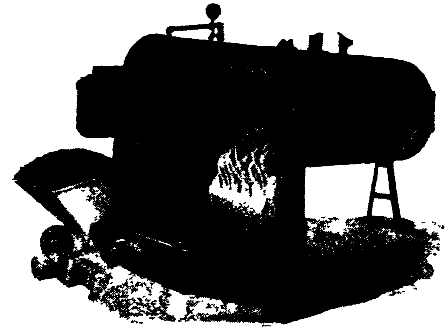
Firebox Boiler Portable Up-draft Type "400" and "500" Series hi-Pressure

The Kewanee series include:

HEAVY DUTY RIVETED FIREBOX TYPES: 1,240 ft to 42,500 ft. Brickset and portable settings, Updraft and Downdraft Smokeless Furnace, Single-pass tubes for rear smoke outlet; Two-pass tubes for front smoke outlet.

WELDED BOILERS: 2,200 ft to 42,500 ft. Direct Draft or Smokeless Arch with Corrugated Crown Sheet. Rear Smoke outlet and Weld + Rivet for front Smoke outlet.

RESIDENCE STEEL BOILERS: 225 ft to 2,924 ft. Square and Round Type "R" with and without Jackets and Hot Water Heating Coils for Storage Tank or Instantaneous flow



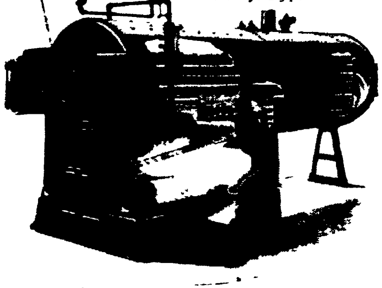
Firebox Boiler for Stoker "400" and "500" Series

SPECIFICATIONS—PORTABLE UP-DRAFT BOILER

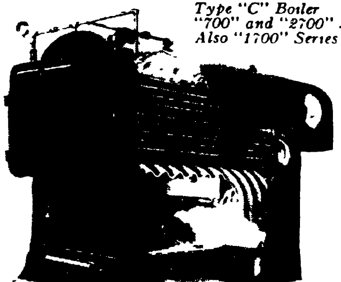
Boiler No.		576	577	578	579	580 480	581 481	582 482	583 483	584 484	585 485	586 486	587 487	588 488	589 489	590 490
Rated Steam Capacity																
Coal	Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker	Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width and Length In x Ft In.		42x8-7	42x9-6½	48x8-10	48x9-6½	54x11-5	54x13-1½	60x13-4½	60x15-5½	66x15-6½	66x18-0½	72x17-0	78x17-7½	78x21-3½	84x20-7	84x23-4
Overall Height Shell	In	80	80	86	86	94	94	101	101	107	107	113	115	115	125	125
Height of Water Line	In	70	70	73	73	79½	79½	84½	84½	89½	89½	94	96½	96½	105	105
Approximate Weight: Coal	Lb	6100	6700	7300	7900	10200	11400	13300	14800	17300	19700	22000	24200	28400	32300	36100
Oil	Lb	5500	6100	6600	7100	9300	10400	12200	13600	15800	18100	20300	22300	26400	30200	33800

Rated Capacity for Water Boiler is 60 per cent greater than Capacity for Steam Boiler

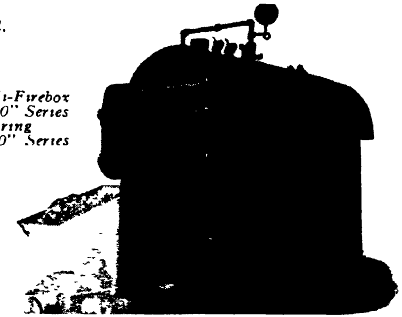
Table for two series of Boilers lists maximum dimensions only.



Smokeless Boiler Portable. Down-draft Type "300" Series



Type "C" Boiler "700" and "2700" Series for Coal. Also "1700" Series for Oil



At Right Type "C" Hi-Firebox Boiler, "7L70" Series for Stoker Firing. Also "27L70" Series Hand Fired

SPECIFICATIONS—SMOKELESS DOWN-DRAFT BOILER

Boiler No.		376	377	378	379	380	381	382	383	384	385	386	387	388	389	390
Rated Steam Capacity:																
Coal	Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker	Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width and Length	In. x Ft In.	42x8-3	42x9-3	48x8-7½	48x9-5	54x11-2½	54x12-11	60x12-8½	60x14-9	66x14-11	66x17-4	72x16-5	78x17-0½	78x20-7½	84x19-11	84x22-8
Overall Height Shell	In.	80	80	86	86	94	94	101	101	107	107	113	115	115	125	125
Height of Water Line	In.	70	70	73	73	78	78	85	85	88½	88½	94½	95	95	107½	107½
Approximate Weight: Coal	Lb	6800	7500	8100	8800	10000	11100	13600	15300	18000	20500	22900	25100	29500	33600	37500
Oil	Lb	6200	6900	7400	8000	9100	10100	12500	14100	16500	18900	21200	23200	27500	31500	35200

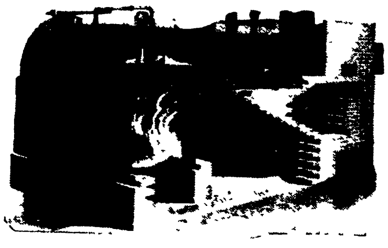
SPECIFICATIONS—TYPE "C" WELDED BOILER

*Boiler No.		2773	774 2774	775 2775	776 2776	777 2777	778 2778	779 2779	780 2780	781 2781	782 2782	783 2783	784 2784	785 2785	786 2786	787 2787	788 2788	789 2789	790 2790
Rated Steam Capacity:																			
Coal	Sq Ft	2200	2600	3000	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker	Sq Ft	2680	3160	3650	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width & Length, In.xFt	In.	36x5-10	36x6-4	36x6-10	36x7-9	42x7-10½	42x8-6½	42x9-2	48x9-4½	48x10-7	54x9-11	54x11-2½	60x11-6½	66x12-3½	72x12-1½	72x13-4½	78x14-9½	84x14-2½	84x15-11½
Overall Height Shell	In.	77½	77½	77½	77½	83½	83½	83½	86½	86½	99	99	108½	112	118	118	122	135	135
Height of Water Line	In.	69	69	69	69	72	72	72	73	73	85	85	94	95	101	101	103	114	114
Approximate Weight:																			
700 Series, Coal	Lb		3900	4400	5000	5500	6000	6500	7500	8400	9700	11000	12900	14900	16600	18400	22000	25200	28400
2700 Series, Coal	Lb	3400	3800	4300	4800	5300	5800	6300	7200	8100	9400	10600	12500	14400	16100	17900	21200	24400	27500
Oil	Lb	2900	3300	3700	4100	4600	5000	5400	6100	6900	8000	9100	10700	12300	13800	15400	18100	20900	23400

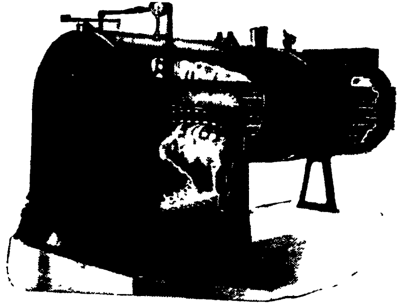
SPECIFICATIONS—TYPE "C" HI-FIREBOX WELDED BOILER

*27L70 Boilers have hand-fired coal grates and ratings, with 7L70 dimensions

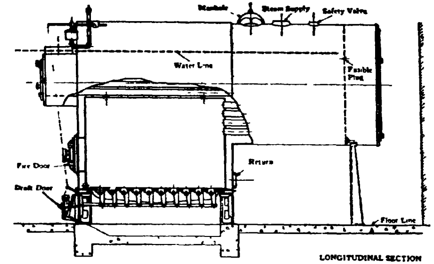
Boiler No.		7L73	7L74	7L75	7L76	7L77	7L78	7L79	7L80	7L81	7L82	7L83	7L84	7L85	7L86	7L87	7L88	7L89	7L90
Rated steam capacity,																			
Stoker	Sq Ft	2680	3160	3650	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width x Length, In.xFt	In.	36x5-10	36x6-4	36x6-10	36x7-9	42x7-10½	42x8-6½	42x9-2	48x9-4½	48x10-7	54x9-11	54x11-2½	60x11-6½	66x12-3½	72x12-1½	72x13-4½	78x14-9½	84x14-2½	84x15-11½
Overall Height	In.	82½	82½	82½	82½	88½	88½	88½	94½	94½	108	108	119½	123	131	134	140	157	161
Height of Water Line	In.	74	74	74	74	77	77	77	81	81	94	94	105	106	114	117	121	136	140
Approximate Weight	Lb	3100	3500	3900	4400	4900	5400	5800	6700	7600	8800	10000	11800	13600	15300	17000	20100	23200	26100



Brickset Boiler Up-draft Type



Type "K" Up-draft Boiler



Weld + Rivet Firebox Boiler "5000" Series Up-draft. Also "6000" Series Smokeless Down-draft

SPECIFICATIONS—BRICK-SET AND TYPE "K" PORTABLE UP-DRAFT BOILERS

Boiler No.		3 3K	4 4K	5 5K	6 6K	8K	8 9K	9 10K	10 11K	11 12K	12 13K	13 14K	14 15K	15 16K	17 18K	18 20K	19	20
Rated Steam Capacity:																		
Coal	Sq Ft	1240	1380	1800	2200	3000	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000
Oil, Gas or Stoker	Sq Ft	1770	2020	2190	2680	3650	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290
Width and Length	In. x Ft In.	30x9-10	36x8-10	36x10-4	36x11-10	42x11-4	42x12-10	48x11-10	48x13-4	48x14-10	54x15-3	54x17-9	60x17-0	60x19-6	66x19-6	72x19-6	72x16-0	72x18-0
Overall Height Shell	In.	59	65	65	65	71	71	77	77	77	83	83	93	93	99	105	107	107
Height of Water Line	In	52	55	55	55	58½	58½	61	61	61	66	66	75	75	80	85½	85½	85½
Approximate Weight: Coal	Lb	3500	3900	4200	4700	5700	6400	7000	7600	8200	9500	10700	12600	14500	17600	20700	19400	21900
Oil	Lb	3130	3550	3800	4200	5100	5800	6400	6900	7400	8600	9700	11500	13300	16100	19100	17700	20000

SPECIFICATIONS—WELD + RIVET FIREBOX BOILER "5000" SERIES

Boiler No.		5076	5077	5078	5079	5080	5081	5082	5083	5084	5085	5086	5087	5088	5089	5090
Rated Steam Cap: Coal	Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, gas or stoker	Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Diam. x Length	In. x Ft In	42x7-3	42x8-0	42x8-11	42x9-10	48x9-9	48x11-1¼	54x11-1	54x12-9	60x13-2½	60x15-3	66x14-5½	66x16-0½	72x16-8½	78x17-5½	78x19-8½
Overall Height	In.	81	81	81	81	86½	86½	92½	92½	100½	100½	105	105	112	117	117
Height of Water Line	In.	70	70	70	70	73½	73½	78	78	84	84	86	86	94½	96	96
Approx. Weight: Coal	Lb	5100	5600	6100	6600	7500	8400	9600	10800	12600	14400	16100	17900	21200	24400	27500
Oil	Lb	4500	5000	5400	5800	6600	7400	8500	9600	11100	12800	14400	16000	19200	22300	25200

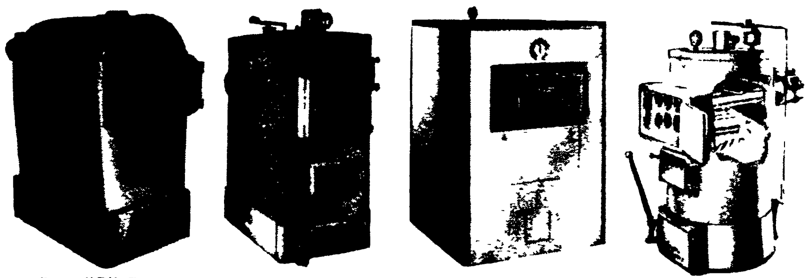
SPECIFICATIONS—WELD + RIVET SMOKELESS BOILER—"6000" SERIES

Boiler No.		6077	6078	6079	6080	6081	6082	6083	6084	6085	6086	6087	6088	6089	6090
Rated Steam Capacity: Coal	Sq Ft	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker	Sq Ft	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Diameter x Length	In x Ft In	42x8-2	42x8-8	42x9-6	48x9-7	48x11-0	54x10-8½	54x12-4½	60x12-9½	60x14-9½	66x14-0½	66x15-7½	72x16-4½	78x17-1	78x19-4
Overall Height	In	81	81	81	86½	86½	92½	92½	100½	100½	105	105	112	117	117
Height of Water Line	In	70	70	70	73½	73½	78	78	84	84	86	86	94½	96	96
Approximate Weight Coal	Lb	5800	6300	6800	7700	8600	9900	11200	13100	14900	16600	18300	21500	24500	27300
Oil	Lb	5100	5600	6100	7000	7800	8900	10100	11900	13500	15100	16700	19800	22700	25300

KEWANEE TYPE "R" RESIDENCE BOILERS

Kewanee Type "R" Boilers are especially designed and constructed to meet all heating and hot water requirements for homes and small buildings Every kind of solid fuel, coke, all grades of hard or soft coals and their briquette or treated forms are burned with excellent results Also, any liquid fuel, oil, and natural or commercial gas can be used with high efficiency.

Standard snug fitting jackets, Rex or Regal extension styles enclosing burners are available for Round "R", Square "R" or 83R Boilers. Hot Water Copper Coil. 55 and 65 gal for Round "R". Capacities up to 720 may be ordered for Square "R" and 83R Boilers.



Type "R" Boilers Square "R", 83R, and Round "R"—in Flush, Round or Extended Jackets

SPECIFICATIONS—RESIDENCE BOILERS SQUARE "R" ROUND TYPE "R"

*Boiler No.		742	743	745	746	747	748	733	734	735	736	1737
Net Load Steam, Coal.	Sq Ft	790	1000	1350	1600	1780	1960	225	246	320	423	
Oil, Gas or Stoker	Sq Ft	840	1120	1470	1900	2160	2380	275	400	540	680	900
Width and Length	In x In.	32x39½	32x45½	32x45½	32x51½	32x57½	32x63½					
Diameter Firebox Inside	In							17	20	23	26	26
Overall Height Shell Top	In	59½	59½	70¼	70¼	70¼	70¼	54	54¾	55¼	59½	61
Height of Water Line	In.	48	48	58½	58½	58½	58½	46	44	44	46	49
Approximate Weight. Coal	Lb	2150	2360	2800	3050	3300	3550	800	880	1040	1280	
Oil	Lb	1900	2060	2500	2730	2920	3125	600	800	920	1130	1400
Standard Jacket, Crated	Lb	205	225	175	190	200	225			190	215	

Boiler No.	83R1	83R2	83R3	83R4	83R6	83R7	83R8	83R9	Oil and Gas; or Stoker 083R.
N. Load Steam	901	1105	1326	1513	2091	2363	2652	2924	
Wt. Jacketed	1600	1800	2000	2200	2800	2900	3100	3300	

*Boiler series 1742-1748, 1733-1737 for oil, gas or stoker; 2742-2748, 2733-2736 for anthracite Kewanee Indirect Hot Water Heating Coils for Type C, Square and Round "R" Boilers; 55 sizes, 55 to 1520 gal

KEWANEE STORAGE WATER HEATERS

Kewanee Storage Heaters—use exhaust or live steam. 15 standard Coil Elements in 29 standard size storage tanks. Capacities 95 to 2240 gals.

KEWANEE SCOTCH MARINE BOILERS

Kewanee Scottie Junior for High Pressure Steam in small industrial usage. 5 sizes 9.9 to 30 hp at 100 lbs steam pressure.

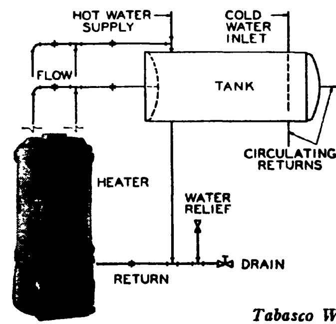
Kewanee Welded Scotch Marine, Low Pressure Steam. 18 sizes, 2680 to 42500 sq ft, mechanically fired.

KEWANEE HI-TEST BOILER

Kewanee Hi-Test Fusion Welded Series for High Pressure Steam in power or industrial process. 6 stock sizes, 50 to 150 hp, 125 and 150 lbs steam working pressure. All **A.S.M.E.** Code.

KEWANEE WATER HEATERS AND TANKS

Tabasco All-Weld Water Heaters, 10 sizes to heat 130-700 gal 50 deg per hour. Extra Heavy W. P. 100 lb per sq in. 200 lb Test.



Also Water Heating Garbage Burners.

Tanks 66 to 1700 gals.

Tabasco Water Heater

THE NATIONAL RADIATOR COMPANY

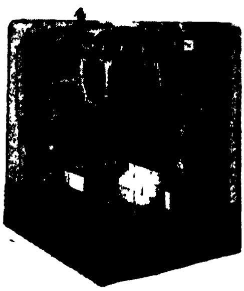
MODERN DESIGN — HEATING EQUIPMENT

Johnstown — Pennsylvania

Branch Offices:

BALTIMORE	2622 Matthews Street	PHILADELPHIA	401 North Broad Street
BOSTON	167 Bridge Street, Cambridge	PITTSBURGH	304 Arrott Building
CHICAGO	Daily News Building	RICHMOND	Richmond Trust Building
NEW YORK	60 East 42nd Street	WASHINGTON	4034 Georgia Ave., N W.

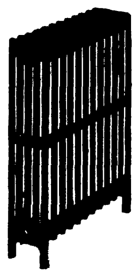
No. 1-2-3 SERIES C.I. BOILERS



Latest boiler design in pre-war equipment. Six flue-ways, extended heating surface on firebox crown-sheet and surfaces of flue-ways. Baffles on both sides of each section precipitate excess moisture from steam. Front flow tapping. Submerged tank or tankless hot water heaters.

RATING (I-B-R)·
Hand, Stoker, Oil-fired
Steam —170–2,300 sq ft
Hot Water—270–3,680 sq ft

ART RADIATORS

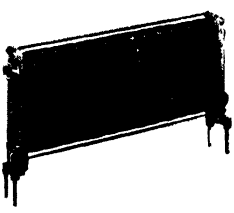


The National "Art" radiator was the first of the compact, thin tube radiators to be introduced with a complete range of sizes. Fluted shoulders add to the appearance, giving attractive lines. Can be furnished legless, with high legs or special harmonizing pedestals.

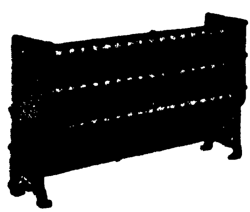
OIL HEATING UNITS—Cast-Iron and Steel Design

GAS HEATING UNITS—Compact; Efficient Heat Absorbtion

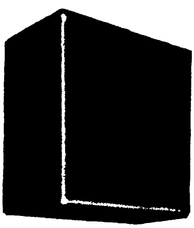
AERO CONVECTORS



The exclusive design of sloping sections exposes a greater amount of heating surface to the flow of air traveling through the convector. More heating surface can be arranged in less space with this design and additional sections are easily added. A variety of Convector and Enclosure designs and coupled arrangements provide flexibility for installation requirements.



UNIT HEATERS

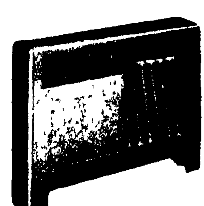


Where a large volume of air is to be heated in stores, garages and workshops, the NRC Unit Heaters will give efficient diffusion of warmth to all areas. Available in horizontal or vertical shaft types of heaters. For steam or hot water.

ACCESSORIES

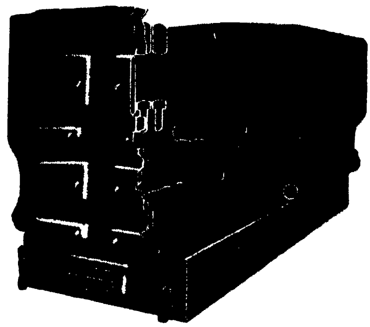
Floor plates, valves, traps, circulating water pumps, controls, etc., are furnished for complete installations.

ENCLOSURES



SERVING THE HEATING INDUSTRY FIFTY YEARS

No. 40-42-48 SERIES—Super-Smokeless and Standard



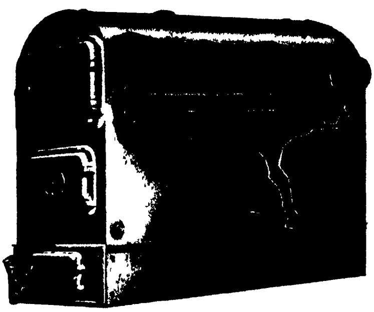
Twin sections facilitate assembly. Large amount of heating surface and adequate flue-ways. Super-Smokeless Boiler has air duct separated by ribs from section to which attached, permitting air to circulate around it. Air is preheated, delivered in thin streams into gases to burn unconsumed fuel. Bridgewall Section is furnished in 11 or more sections.

RATINGS (Bonded):

Steam —2,200–11,200 sq ft.
Hot Water—3,625–18,400 sq. ft.

NATIONAL STEEL BOILERS

Residential Series—Unjacketed



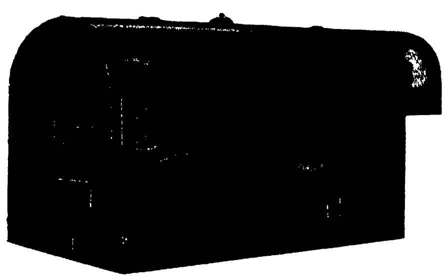
The National Steel Boilers are low pressure type. The Residential Series is also installed in commercial buildings. Electrunite tubes assure longer, trouble-free service. True-arch crown provides maximum firebox height, sludge will not accumulate over hottest part of boiler, better heat-absorbing surface.

RATINGS (SHBI)·

Type DB	Hand-fired	—Steam	— 485-2,200 sq. ft.
		Hot Water—	775-3,520 sq ft.
MB	Stoker-fired }	—Steam	— 735-2,680 sq. ft.
OB	Oil-fired }	—Hot Water—	1,175-4,280 sq. ft.

Commercial Series

Hand-fired series illustrated. Shakers and bridgewall eliminated on OB and MB types, special front base frame (plate) supplied. Recessed smoke box for flush front. Three-pass design. Special ½ in. or ¼ in. opening grates as ordered. Indirect hot water heaters located below water line, front connections for domestic hot water supply; also tank enclosed hot water heaters. Smokeless arch and detachable, water-cooled bridge wall can be furnished (SB type). Rear and front smoke outlets.



Tankless Hot Water Heaters.

Tank Enclosed Hot Water Heaters.

RATINGS (SHBI):

Type DB-SB	Hand-fired	—Steam	—1,800–35,000 sq. ft.
		—Hot Water—	2,880–56,000 sq. ft.
DF-SF	Hand-fired	—Steam	—3,000–35,000 sq. ft.
	(Front Outlet)	—Hot Water—	4,800–56,000 sq. ft.
MB	Stoker-fired }	—Steam	—2,190–42,500 sq. ft.
OB	Oil-fired }	—Hot Water—	3,500–68,000 sq. ft.
MF	Stoker-fired	—Steam	—3,650–42,500 sq. ft.
OF	Oil-fired (Front)	—Hot Water—	5,840–68,000 sq. ft.

CATALOGS WITH COMPLETE DATA AND DIMENSIONAL INFORMATION WILL GLADLY BE FURNISHED

Spencer Heater

Division—The Aviation Corporation

Williamsport, Pa.

Sales Representatives in Principal Cities

AN Avco PRODUCT

Spencer Automatic Magazine Feed Heaters are furnished in cast iron sectional types—and steel tubular types for larger buildings—for steam, vapor and hot water heating. There is a size and capacity for every type of building, to provide economical and convenient heat—safe, dependable, sure.

COMFORTABLE HEAT AT LOW COST

Spencer Jacketed Heater L-1 Series

Why Spencer Heaters perform so satisfactorily can best be explained by an inspection of their design and construction. The Spencer principle, illustrated in the cross-sectional view, is simple:

Once a day fuel (No. 1 Buckwheat Anthracite or small size by-product coke) is put into the magazine. It fills the sloping grate to the level of the magazine mouth. The fire bed always stays at the proper level, for as fast as fuel burns to ash, it shrinks and settles on the sloping grate; and more fuel rolls down automatically over the top of the fire bed. Fuel feed is by gravity alone, in just the right amount to keep the fire always burning at its most efficient combustion point.

This explains why a Spencer Automatic Magazine Feed Heater always gives the same uniform, satisfying heat, and burns less fuel. These exclusive Spencer advantages are available in all types of the magazine feed heaters and boilers.

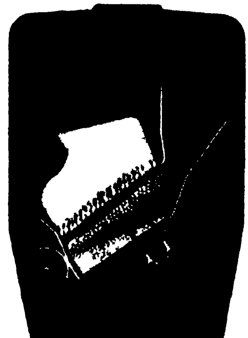
Coal — Coke — Gas — Oil — Spencer J and L series heaters and M series boilers are primarily designed to burn low cost No. 1 Buckwheat Anthracite or small size coke.

If at any time a property owner desires to burn more expensive fuels—oil or gas—his Spencer Heater can be readily converted and will show a high efficiency

Thermostats—Thermostats and electric damper motors are furnished as optional equipment.

Jacketed Covering—Illustrated in the attractive metallic jacket of the deluxe enclosing type for Spencer Cast Iron Heaters, either with or without the enclosing jacket doors.

Spencer Heavy Duty Tank Heaters—With the automatic magazine feed construction, they provide ample domestic hot water at lowest cost, and with a minimum of tank heater attention.

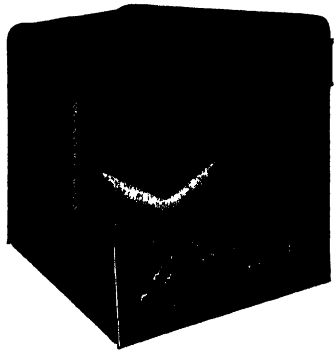


Cutaway sectional view Spencer Cast Iron Heater

SPENCER ALL YEAR SYSTEM

In addition to the excellent heating facilities afforded by Spencer Magazine Feed Heaters, Year Round Domestic Hot Water Service can also be provided and assures at all times an ample supply of domestic hot water at lowest cost. Complete data for installation and operation upon request.

SPENCER STEEL TUBULAR MAGAZINE FEED BOILERS



Steel Tubular Magazine Feed Boiler

For large buildings we recommend Spencer Steel Tubular Magazine Feed Boilers, burning low cost No. 1 Buckwheat Anthracite or coke.

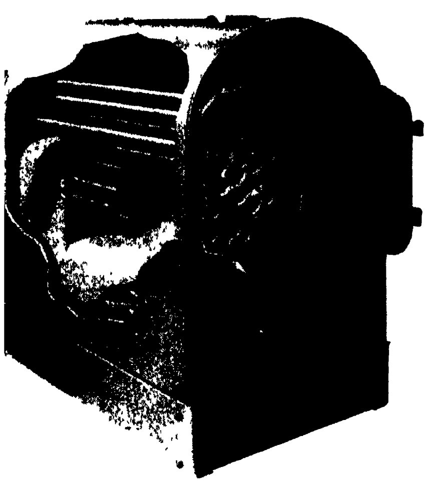
In the cross-section diagram, part of the fire bed is cut away to show the sloping grates and the two magazines filled with fresh coal, ready to feed down automatically by gravity to the fire. These boilers are built in two vertical sections for ease in handling and installation—a great advantage on replacement jobs, eliminating the necessity of costly tearing out of walls or partitions. Combination water and fire tube construction, built to *A S M E* standards

SPENCER STEEL TUBULAR BOILERS
For Oil, Stoker, Gas or Hand-Firing

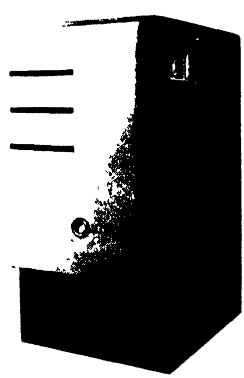
For more than 55 years, Spencer has been building, in the opinion of experts, one of the most efficient, economical and dependable automatic coal burning boilers on the market. With this background of experience, Spencer Engineers developed the Spencer Steel Tubular Boiler for oil, gas, stoker and hand-firing—the "K" and "C" series for residential use, and the Type "A" for larger buildings. They are better boilers both for the property owner and for the architect or engineer who specifies them

The high sustained efficiency of these boilers means adequate heat for a lower fuel cost. Design is of the three pass type. Combustion chamber is amply large. Built of best quality open hearth steel boiler plate, and steel tubes. Can be furnished with domestic hot water heating coils, storage tank or instantaneous type.

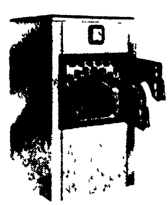
A complete range of sizes from 400 sq ft SHBI net steam rating up. They meet or exceed in every particular the requirements of the A.S.M.E. and S H. B. I. Codes.



Type "A" Steel Boiler



"C" Series Steel Boiler



"K" Series

Every Spencer Boiler is guaranteed to carry more than its full rated load giving the installer a definite factor of safety.

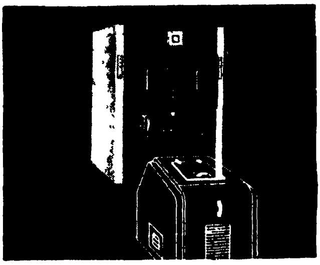
These boilers have all the advantages of the Spencer exclusive design and are readily adapted to mechanical oil or stoker firing—or hand-fired coal or coke.

The H. B. Smith Company, Inc.

Westfield, Mass.

Branch Offices and Sales Representatives in Principal Cities

A complete line of modern cast iron sectional boilers for residential, commercial and industrial heating and for domestic hot water supply



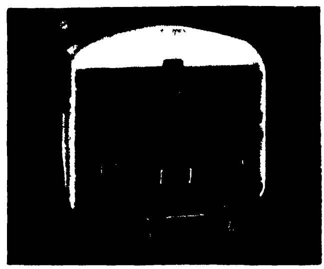
20 Mills

15-20-25 SMITH-MILLS BOILERS

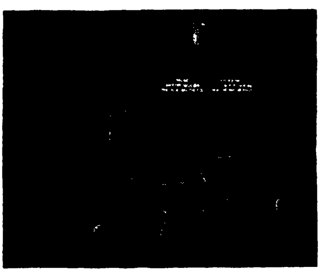
Capacities 200 sq. ft. to 2275 sq. ft. steam radiation. This complete line of modern push nipple boilers is available in models for oil, gas, stoker and hand firing Provisions have been made for built-in domestic hot water heaters and controls.

MILLS WATER TUBE BOILERS

Capacities 900 sq. ft. to 13,380 sq. ft. of steam radiation. Independent header type construction—literally thousands of these famous Mills boilers have been installed in camps, cantonments, bases, war housing and other essential construction during the past two years. Models for hand and all types of automatic firing



44 Mills



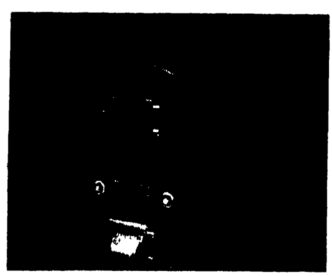
60 Smith

42 AND 60 SMITH BOILERS

May be used in batteries for heating loads up to and over 100,000 sq ft. steam radiation Many of these large units installed in industrial plants furnish steam for process requirements as well as for heating and domestic hot water.

SMITH HY-TEST BOILERS

Smith Hy-Test Boilers for hot water supply, are available in several models and many sizes for tank capacities to 20,000 gal. Constructed of the finest quality grey iron castings, these Hy-Test units are carefully tested at high pressures before shipment. The No. 17 series, for example, is tested at 350 lbs. hydrostatic pressure—the highest test pressure of any cast iron boiler made.



17 HY-Test

Complete catalogue information describing Smith boilers is filed in current issues of Sweet's "Engineering" and Domestic Engineering Catalogue Directory

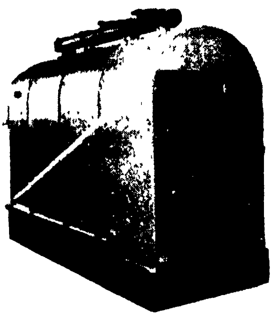
Terre Haute Boiler Works Co.

Established 1857

Main Office and Works:
Terre Haute, Ind.

Branches and Representatives in PRINCIPAL CITIES

CLIFF LOLINE SECTIONAL STEEL HEATING BOILERS



For Oil, Gas, and Coal Stoker or Hand Firing

Designed especially for boiler replacement work. Efficient Tube Heating Surface and Generous Firebox Proportions Only four sections are required for maximum rating. Compactness and accessibility for replacement work in old buildings, etc., eliminates expensive and hazardous excavation or foundation wall removal. Each section will easily pass through the ordinary door opening.

Assembly of Front Header made easy with a Strap and Slip Bolt. Watertight Joint made with Composition Gasket on Boiler Side of Header. Yoke slotted for easy Slip Bolt Assembly. Small lugs on Yoke provide quick alignment of sections. Heavy stay-bolting throughout.

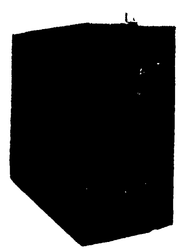
NOTE: Rock wool insulation and baked enamel, steel jacket furnished at extra cost.

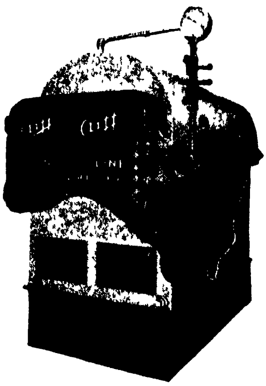
RATINGS -Hand Fired -2100-29820; Oil, Gas, Stoker—2550-36210 sq. ft. Steam

CLIFF STEEL RESIDENTIAL BOILERS

For Oil, Gas, Stoker or Hand Firing

A Three-Pass electric welded Steel Boiler furnished with Red Enameled Steel Jacket and 1½ in thick rock wool insulation for Residence, Apartments and Smaller Buildings Will pass thru 31 in doorway. Designed for 15 lbs working pressure on steam and 20 lbs pressure on water Rated in accordance with S.H.B.I. Code. Submerged type hot water coils can be furnished installed inside of boiler; either tank-heater or tank-less type Ratings 800-3160 sq ft steam





CLIFF HEAVY DUTY LOLINE BOILERS

For Oil, Gas, Stoker or Hand Firing
Also Hi-Firebox Type
A.S.M.E. Code Construction

Three-Pass compact Fire-Box Boilers. Designed and constructed to A.S M.E. Standard for 15 lbs. steam or 30 lbs. water working pressures. All boilers comply with the Industry's simplified practice recommendations R157-35 as issued by the U. S. Department of Commerce and the S H.B.I. Code for the rating of low pressure heating boilers

Can also be furnished with a Cliff Patented Steel Jacketed Red Enamel Finish having 1½ in. Rock Wool Insulation.

RATINGS—Hand Fired—2600-35000; Oil, Gas, Stoker—3160-42500 sq. ft. Steam.

Write for complete information on CLIFF Boilers

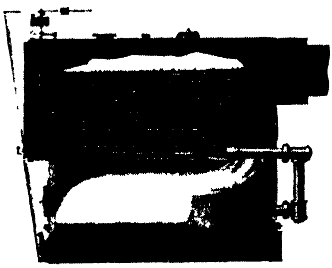
Pacific Steel Boiler Division

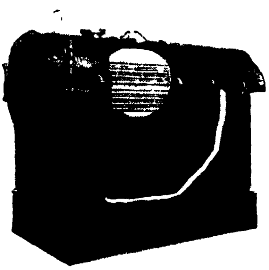
United States Radiator Corporation

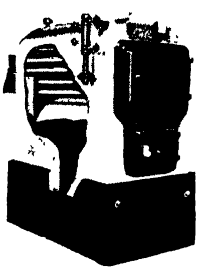
General Offices: **Detroit, Michigan**

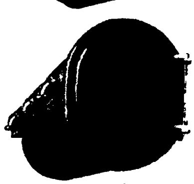
Sales Offices in Principal Cities

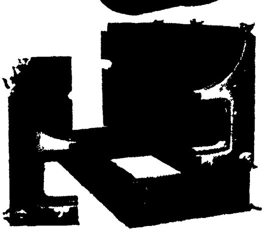
A Complete Line of Low Pressure Steel Heating Boilers











All Pacific Boilers are built using the *A.S.M E.* Boiler Code Standards as minimums.

DIRECT DRAFT AND SMOKELESS SERIES FOR COAL FIRING

Built in the following capacities for steam. 2200 to 35,000 sq ft.

All Pacific Boilers are built, inspected, and tested under the supervision of the Hartford Steam Boiler Inspection and Insurance Company.

HIGH FIRE BOX SERIES FOR MECHANICAL FIRING, STOKER, OIL OR GAS

Built in the following capacities for steam 2680 to 56,830 sq ft

All Pacific Boilers are made of steel with each joint and seam electrically arc-welded —built to last a life-time.

PACIFIC RESIDENTIAL SERIES FOR COAL, STOKER, OIL OR GAS

Built in the following capacities for steam: 320 to 2890 sq ft

PACIFIC THREE-PIECE CONSTRUCTION

Made up of three parts, shell, firebox and base, Pacific Boilers are particularly adaptable to replacement work. Where necessary Pacific fireboxes can be split (as illustrated) allowing the boiler to be taken into the building in four pieces and erected without welding on the job.

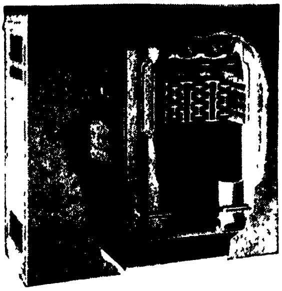
Descriptive Bulletins on Pacific Steel Heating Boilers will be mailed on request.

UNITED STATES RADIATOR CORPORATION

General Offices: Detroit, Michigan • Branches and Sales Offices in Principal Cities

Detroit, Michigan

CAPITOL SUNRAY No. 2 SERIES OIL FIRED BOILER

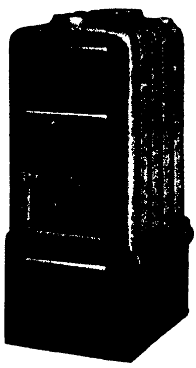


Boiler No	Net I-B-R Sq Ft Direct Cast-Iron Radiation		Net I-B-R Btu	Gross I-B-R Output Btu
	Steam	Water		
2-03	350	560	84 000	128,000
2-04	500	800	120,000	180,000
2-05	650	1040	156,000	232,000
2-06	800	1280	192,000	284,000

"DEEPFIRE" HOT WATER SUPPLY BOILERS

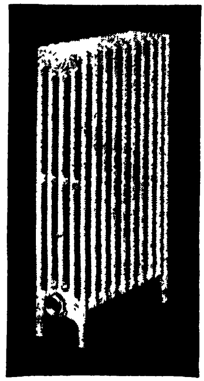
Coal Fired

Tested to 250 lb Hydrostatic Pressure for 100 lb Working Pressure



Boiler No.		Capacity—Gallons		
		100° Rise 6 Hours	85° Rise 1 Hour	100° Rise 1 Hour
30-D	30	792	155	132
40-D	40	1512	297	252
50-D	50	2160	423	360
60-D	60	2700	529	450

*CAPITOL THINTUBE RADIATORS

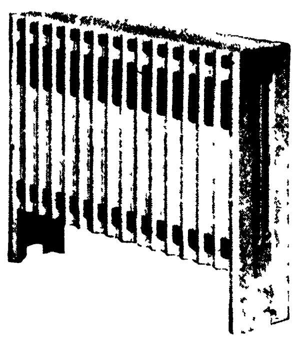


Heights In.	Per Section Heating Surface
3-Tube	
25	1.6 Sq Ft
4-Tube	
19	1.6 Sq Ft
22	1.8 Sq Ft
25	2.0 Sq Ft
5-Tube	
22	2.1 Sq Ft
25	2.4 Sq Ft
6-Tube	
19	2.3 Sq Ft
25	3.0 Sq Ft
32	3.7 Sq Ft

*1¾ in Centers

40 per cent less space needed for these graceful, efficient Capitol ThinTube Radiators.

U. S. SUNRAY RADIATOR



The **U. S. Sunray Radiator** forms its own cabinet enclosure. Used as a free standing radiator or recessed in the wall. Can be furnished in 22½ in. heights with removable grilles or cast-iron corner plates. Manufactured with outside end tappings or inside drop leg tappings or tappings on the bottom of intermediate sections.

No 5 Depth 5⁹⁄₁₆ In.	Per Section Heating Surface 1.8 Sq Ft
No. 6 Depth 6¾ In.	Per Section Heating Surface 2.3 Sq Ft

LITERATURE UPON REQUEST

UNITED STATES RADIATOR CORPORATION

General Offices: Detroit, Michigan • Branches and Sales Offices in Principal Cities

Detroit, Michigan

CAPITOL RED TOP BOILERS

"B" Series Unjacketed Boiler

"A" Series—All Fuels

Boiler No.	I-B-R Sq Ft Direct Cast-Iron Radiation	
	Steam	Water
A-7	340	545
A-8	440	705
A-9	540	865
A-10	640	1025
A-11	740	1185

"B" Series—All Fuels

B-7	740	1180
B-8	920	1470
B-9	1110	1770
B-10	1275	2040
B-11	1440	2300
B-12	1580	2530
B-13	1730	2770
B-14	1880	3000

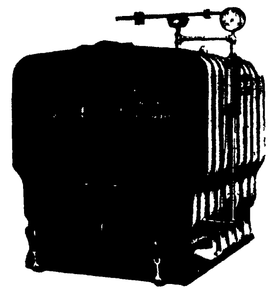
"C" Series—All Fuels

Boiler No.	Direct Cast-Iron Radiator Loads Sq Ft	
	Steam	Water
C-12	2250	3600
C-14	2715	4345
C-16	3180	5090
C-18	3645	5830
C-20	4110	6575
C-22	4575	7320
C-24	5040	8065
C-26	5450	8720
C-28	5800	9280
C-30	6110	9775

U. S. Sunray No. 3 Series

Boiler Number	I-B-R Sq Ft Direct Cast Iron Radiation					
	Hand Fired		Stoker Fired		Oil Fired	
	Steam	Water	Steam	Water	Steam	Water
23-S or W	150	240			260	415
33-S or W	220	355			310	500
43-S or W	300	480	400	640	400	640
53-S or W	450	720	550	880	550	880
63-S or W	600	960	700	1120	700	1120

CAPITOL SQUARE SECTIONAL BOILERS



"50" Series (All Fuels)

"50" Series—All Fuels

Boiler No	Direct Cast-Iron Radiator Loads Sq Ft	
	Steam	Water
950	5640	9025
1050	6560	10,500
1150	7435	11,895
1250	8300	13,280
1350	9140	14,625
1450	10,000	16,000
1550	10,840	17,345
1650	11,660	18,655
1750	12 460	19,935
1850	13240	21,185

"WN" Series—All Fuels

WN-276	3110	4975
WN-277	3885	6215
WN-278	4765	7625
WN-279	5640	9025
WN-280	6560	10,500
WN-281	7435	11,895
WN-282	8300	13,280
WN-283	9140	14,625
WN-284	10,000	16,000

"WNO" Series—For Oil Firing

WN-0277	4935	7895
WN-0278	5680	9090
WN-0279	6425	10,280
WN-0280	7170	11,470
WN-0281	7910	12,655
WN-0282	8775	14,040
WN-0283	9605	15,370
WN-0284	10,455	16,730
WN-0285	11,300	18,080
WN-0286	12,135	19,415
WN-0287	12,965	20,745
WN-0288	13,800	22,080
WN-0289	14,630	23,410

LITERATURE UPON REQUEST

Weil-McLain Company

Manufacturing Division: **Michigan City, Ind. and Erie, Pa.**

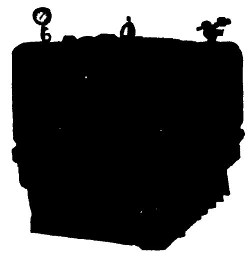
General Offices: **641 W. Lake Street, Chicago 6, Ill.**

NEW YORK OFFICE: **501 Fifth Avenue**

Weil-McLain Boiler and Radiator service is made conveniently available through local stocks carried by Weil-McLain Distributors in most of the important distributing centers.

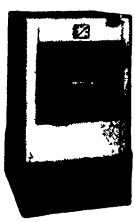
Round-Type Boiler

Unjacketed Round Boiler with corrugated heating surfaces for economical home heating Connected Load Ratings Steam 275 to 1,000 sq ft, Water 440 to 1,600 sq ft



Square-Type Boilers

Sectional boilers for larger installations. Complete range of sizes Connected Load Ratings· Steam 1,815 to 11,300 sq ft, Water 2,900 to 17,900 sq ft.

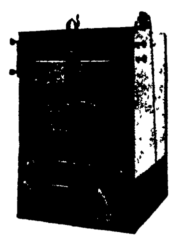


No. 67 -No. 77
All-Fuel Boilers

Conversion type boilers (no jackets furnished for duration). For hand or automatic firing. Connected Load Ratings: Steam 290 to 1000 sq ft, Water 465 to 1600 sq ft.

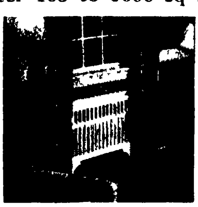
Self-Feed Boiler

Magazine type Boilers for small inexpensive sizes of hard coal or coke. Connected Load Ratings: Steam 240 to 785 sq ft, Water 385 to 1260 sq ft.



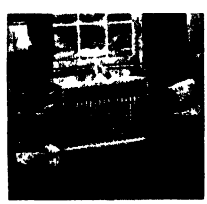
"C" Series Boiler

Designed for larger installations (furnished without jacket for duration). For hand or automatic firing. Connected Load Ratings: Steam 710 to 2525 sq ft, Water 1130 to 4040 sq ft.



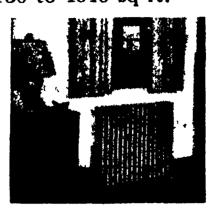
Raydiant "Concealed"

A Raydiant convector type all cast-iron Radiator. Made in "Concealed," also Partially Recessed, Cabinet and Humidifying types.



Solray Radiator

Free standing Cabinet type Radiator in a lower price range than Raydiant Cabinet Radiators. Available in three depths in 21, 24 and 27 in. heights.



Junior Radiator

Smaller Tubular type Radiation which conserves space. Available in 1¾ in. centers in 3, 4, 5 and 6 tube widths and 13 to 32 in. heights.

The Vinco Company, Inc.

305 East 45th Street **New York 17, N. Y.**

Only a clean boiler can be an efficient boiler. A clean boiler means saving fuel, as well as safeguarding boiler metal. Both are now a patriotic duty.

Boiler Cleanser
3 and 5 lb Cans

A positively harmless insoluble powder cleaner for new, remodeled and old heating systems. A unique, scientifically processed compound on a special formula not to be confused with other powder boiler cleaners.

What Vinco Boiler Cleanser Does

VINCO removes oil, grease, scale, rust and dirt from the internal surfaces and from the boiler water *without the labor, expense, and uncertain results of blowing boilers over the top or of wasting returns.*

By this thorough cleaning Vinco prevents or cures foaming, priming, surging, and slow steaming.

How Vinco Boiler Cleanser Works

Each minute grain of VINCO powder adsorbs several times its own weight of oil, rust and dirt. These larger grains of adsorbed impurities then settle and are drained through the bottom according to directions on each can.

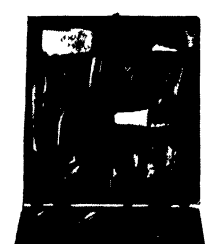
Vinco Guarantees

1. VINCO contains no potash, lye, soda of any kind, oil, acid, or other harmful ingredients.
2. Purchase price is refunded if results are not as claimed when VINCO has been used according to directions.

Rust Preventer
1 qt cans only

VINCO RUST PREVENTER

When used after VINCO Boiler Cleanser has removed oil, grease, rust, scale and dirt, it will add and keep the rust inhibiting factors at the optimal constant for a year or more (Test kit below has complete instructions and chart)



Vinco Test Kit No 10
(Patent applied for)

VINCO TEST KIT No. 10

for Testing Heating Boiler Waters

The kit enables the layman to make simple, rapid tests to diagnose and prescribe correct treatment of boiler waters right on the job

A new time saving method that permits valid conclusions heretofore requiring complicated and often lengthy laboratory analysis and technique.

Each kit has sufficient material for complete tests on 100 jobs

Refills cost about 2 cents per test.

HELP WIN THE WAR BY SAVING FUEL AND BOILER METAL.

SPECIFICATIONS FOR COMPLETE VINCO TREATMENT OF NEW OR REMODELED STEAM, VAPOR, OR HOT WATER SYSTEMS

Do not use as a cleaning agent soda or any alkali, vinegar or any acid. Use Vinco.

1. AFTER THE SYSTEM IS TESTED AND TIGHT, USE THE PROPER QUANTITY OF VINCO LISTED.

After this first clean-out of any new or remodeled heating system, Vinco Boiler Cleaner need be used only if more piping, radiation, or another boiler is added to the original installation, or if the system is fouled by unwise cleaning or leak-sealing experiments.

2. After using Vinco Boiler Cleaner, Vinco Field Test Kit should be used to determine and apply the proper quantity of Vinco Rust Preventer. Vinco Rust Preventer should be applied annually or whenever the boiler water is drained for necessary repairs to the system.

SPECIFICATION FOR OLD HEATING SYSTEMS THAT DO NOT PERFORM PROPERLY

Diagnose and treat according to Vinco Field Test Kit. If a test kit is not available, consult table of quantities on this page and follow directions on Vinco cans.

SPECIFICATION FOR HOT WATER SYSTEMS

Use half quantities listed for treatment of steam systems to remove impurities. Then use test kit to determine proper quantity of Vinco Rust Preventer

CONSULT THIS TABLE FOR NEW AND REMODELED HEATING SYSTEMS AND When a Vinco Field Test Kit No. 10 is not available if cleaning old heating systems.

QUANTITIES OF VINCO (IN POUNDS) REQUIRED FOR HEATING SYSTEMS

•

(Note that quantities are based on actual installed radiation, not on boiler capacity.)

Sq Ft of Radiation	For Steam or Vapor Systems, to prevent or cure priming or foaming. Also for Hot Water Heating Systems Maintained at approx. 200 F or above	Annually, to remove rust scale, dirt and for Hot Water Systems below 200 F.
up to 350	3	1½
351 " 600	5	2½
601 " 1100	8	4
1101 " 1400	10	5
1401 " 1800	13	6½
1801 " 2100	15	7½
2101 " 2700	18	9
2701 " 3100	20	10
3101 " 3700	23	11½
3701 " 4200	26	13
4201 " 4600	28	14
4601 " 5000	30	15
5001 " 5300	31	15½
5301 " 5600	32	16
5601 " 5900	33	16½
5901 " 6200	34	17
6201 " 6500	35	17½
6501 " 6800	36	18
6801 " 7100	37	18½
7101 " 7400	38	19
7401 " 7700	39	19½
7701 " 8000	40	20
8001 " 8300	41	20½
8301 " 8600	42	21
8601 " 8900	43	21½
8901 " 9200	44	22
9201 " 9500	45	22½
9501 " 9800	46	23
9801 " 10100*	47	23½

*Above 10100 sq ft use an additional pound Vinco for each additional 300 sq ft of actual installed radiation

Soot-Off—1 lb cans 60 and 100 lb drums

VINCO SOOT-OFF

Safely and thoroughly removes the insulating blanket of soot on fire pot, flues and chimney. It also insures against external corrosion (caused by dampness and soot forming sulphuric acid during summer layoff.) No dangerous chemicals.

REMOVE SOOT WITH VINCO SOOT-OFF SEVERAL TIMES A YEAR

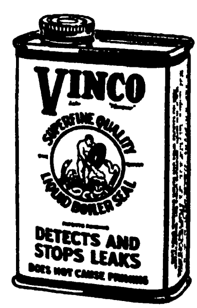
VINCO SUPERFINE LIQUID BOILER SEAL

A different liquid seal. Unique in that it does not induce priming and foaming. It has no unpleasant smell. Makes speedy and permanent repairs of boiler and heating system leaks Fine to tighten up new jobs. Directions simple.

Quantities

Steam and Vapor Systems—Use 1 quart Vinco Liquid Boiler Seal to each 6 sq ft *grate* area.

Hot Water Systems—Use 2 quarts Vinco Liquid Boiler Seal to each 6 sq ft *grate* area.



Liquid Boiler Seal 1 qt. cans only

McDONNELL & MILLER

Safety Devices for Steam and Hot Water Boilers and Liquid Level Controls

General Offices: Wrigley Building, **Chicago 11, Illinois**

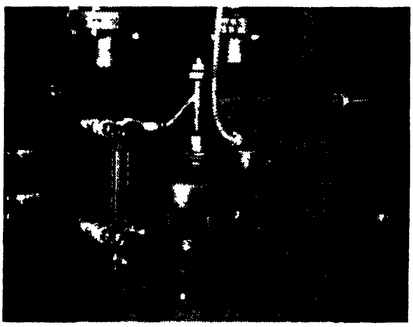
"Doing one thing well"

PRODUCTS:

Boiler Water Feeders; Feeder Cut-off-Combinations; low Water Fuel Cut-offs; Pump Controls; Low Water Alarms; Humidifier Water Level Controls; Safety Relief Valves for hot water heating boilers and storage tanks; High and Low Oil Switches and Liquid Level Controllers for a wide range of services.

The service range and types of installations covering most common applications of McDonnell Boiler Water Level Controls are described here. For facts concerning protection of process boilers, or any special information, consult our engineering department.

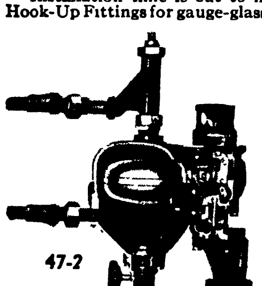
McDonnell Combined Boiler Water Feeder and Low Water Cut-off



McDonnell No 47-2 for heating boilers under 5000 sq ft capacity Maximum steam pressure, 25 lbs.

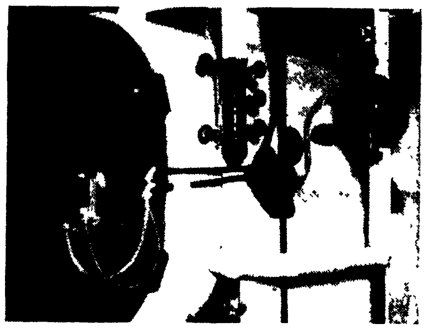
A typical installation of the McDonnell No. 47-2 Combined Boiler Water Feeder and Switch is illustrated above. The water feeder maintains safe level by feeding when water line drops. In case of foaming and priming or failure of water supply, causing water level to drop to ¼ in. in the water glass, low water switch comes into action. For automatically fired boilers switch may be wired to cut current to burner. For hand fired boilers switch may be wired to complete alarm bell circuit. Also furnished for hand fired boilers without switch—designated No. 47.

Installation time is cut to minimum by Quick-Hook-Up Fittings for gauge-glass installation Valve mechanism is isolated from the heat of the float chamber. Large area straight-through blow-off valve is standard equipment



47-2

The McDonnell No 2 Switch used in the No 47-2 is Underwriters' approved Electrical rating: a-c, ¾ hp 110-220 V; d-c, 10 amp. 125 V.

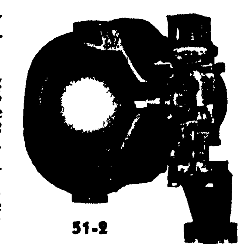


McDonnell No 51-2 for heating boilers over 5000 sq ft capacity Maximum steam pressure, 35 lbs.

The illustration above shows a typical installation of the McDonnell No. 51-2 Combined Boiler Water Feeder and Switch used for larger boilers. This combination functions the same as the No. 47-2, the feeder taking care of normal requirements and the switch safeguarding against emergencies by cutting off the burner in automatically fired boilers or completing a low water alarm circuit for hand fired installations. The feeder is also furnished for hand fired boilers (as No. 51) without low water cut-off and alarm switch.

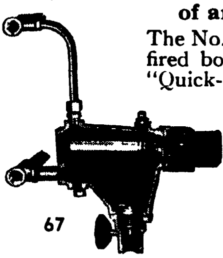
Basic features of the No. 51-2 are the same as in the No. 47-2, except that it is installed with 1 in. equalizing pipe instead of the McDonnell "Quick-Hook-Up" and has the larger feeding capacity required for larger boilers.

For boiler operating at pressures from 35 lb to 75 lb use the McDonnell No. 53-2 (with switch) or No. 53 (without switch) Electrical rating of switches used in No. 51-2 or 53-2 is the same as previously given for the No 47-2



51-2

McDonnell No. 67 Low Water Cut-off for automatically-fired steam boilers of any size. Maximum steam pressure, 25 lb



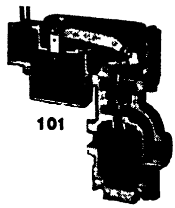
67

The No. 67 takes care of low water cut-off requirements of automatically fired boilers with steam pressure below 25 lbs. It has the McDonnell "Quick-Hook-Up" for quick, easy and trouble-proof installation in the gauge glass tappings; deep sediment chamber with large quick-opening blow-off; packless, non-binding construction; adjustable terminal box to make wiring neat and easy; dependable, snap-action, twin switches.

One switch **closes** on small drop without stopping burner. It can be used to complete an alarm bell circuit, warning of approaching low water, or can be used to complete circuit to the McDonnell No. 101 Electric Water described below. Second switch cuts current to burner if water level drops to ¼ inch in gauge glass.

Underwriters' ratings for both switches· a-c, ¾ hp, 110-220 V; d-c ¼ hp, 115 V.

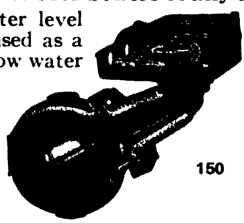
McDonnell No. 101 Electric Boiler Water Feeders for boilers up to 5000 sq. ft. capacity



101

The No. 101 is an electric water feeder for use with No. 67 Low Water Cut-off or with McDonnell "built-in" Low Water Cut-offs which are standard equipment on many modern heating boilers. It converts the cut-off into a combination boiler water feeder and low water cut-off as described on the opposite page.

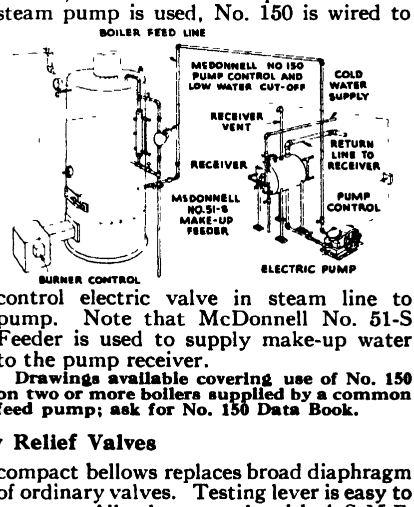
McDonnell No. 150 Pump Control, Low Water Cut-off and Alarm Switch for boilers of any size. Maximum steam pressure, 150 lb



150

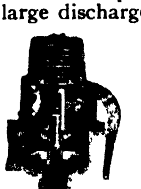
This boiler water level control may be used as a pump control, a low water cut-off, a low water alarm switch, or any combination of these functions Electrical ratings are: a-c., 1 hp, 110-220 V., d-c, ¼ hp, 110-220 V.—as pump control or cut-off; 1 amp. 110 V, a-c. or d-c—as low water alarm. No. 150 has automatic reset low water cut-off switch but may be ordered with manual reset as No. 150-M.

A typical hook-up of the McDonnell No. 150, controlling an electric pump and providing low water cut-off, is shown opposite. When water level drops ¾ in. below normal, No. 150 starts pump and then stops it when normal water line has been restored. If emergency occurs and water level falls to arrow mark on body of control, cut-off switch stops burner. If steam pump is used, No. 150 is wired to control electric valve in steam line to pump. Note that McDonnell No. 51-S Feeder is used to supply make-up water to the pump receiver.



Drawings available covering use of No. 150 on two or more boilers supplied by a common feed pump; ask for No. 150 Data Book.

McDonnell Safety Relief Valves



No. 29

The No. 29 series of Safety Relief Valves have "snap action"—a mechanical means of opening full orifice capacity at precise instant set pressure is reached. Resulting large discharge capacity makes it feasible to rate them in heat dissipating capacity so they can be matched to Btu. output of unit on which they are installed, thus preventing over-pressures and protecting against explosions.

Stainless steel cone replaces conventional composition disc. Long-lived, compact bellows replaces broad diaphragm of ordinary valves. Testing lever is easy to operate. All valves comply with *A.S.M.E.* code.

(1) For Hot Water Heating Boilers

Valve No.	Opening Pressure	Btu. Output
29	29 lbs	156,000
129	29 lbs	350,000

(2) For Domestic Hot Water Heaters and Tanks

Valve No.	Opening Pressure	Max. Water Supply Pres.	Btu. Output
229	75 lbs	50 lbs	316,000
329	100 lbs	75 lbs	380,000
429	150 lbs	100 lbs	432,000

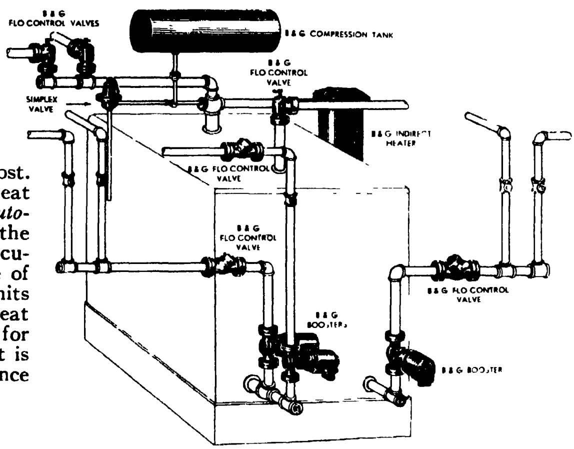
Bell and Gossett Company

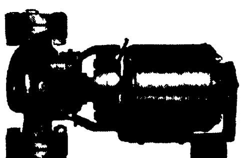
Morton Grove, Illinois

HOT WATER SYSTEMS AND SPECIALTIES

B & G Forced Hot Water Heating Systems

To a postwar world demanding improved ways of doing old things, B & G Forced Hot Water Heating contributes better heating at lower installation and operating cost. It is a system in which the heat input is *accurately and automatically controlled* to equal the heat loss. Water can be circulated through a long range of temperatures, hence permits close adjustment of the heat supply to the actual need for heat. Operating equipment is extremely simple—an assurance of dependable performance.

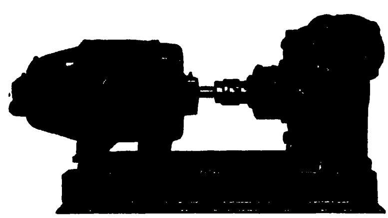




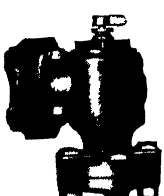
←— B & G Boosters

An electrically-driven centrifugal pump, used to mechanically circulate water through B & G Forced Hot Water Heating Systems. It is distinguished by genuine oil lubrication, patented water-tight seal, exceptionally quiet operation and precision manufacture throughout

B & G Universal Pumps —→



The B & G Universal Pump is designed for large forced hot water heating systems in apartment buildings, office buildings, factories, schools, etc. The installation can be operated as a large single zone or divided into several zones in which circulation of the pumped water in each circuit is controlled by a B & G Motorized Valve, operated by a zone thermostat.



B & G Angle Flo-Control ←— Valves

This valve, installed in the main, shuts off circulation to radiators when heat is not needed, permitting summer operation of a B & G Indirect Water Heater. It also helps maintain a uniform room temperature during the heating season.

B & G Motorized Valves —→

Thermostatically operated valves used for controlling boiler water flow through the individual circuits of zoned heating systems.

←— B & G Monoflo Fittings

B & G Monoflo Fittings permit the use of a *single* pipe main instead of the conventional flow and return lines. It is installed at the junction of the radiator risers to the single main and assures the diversion of the proper amount of heated water into each radiator. Savings in space, labor and materials are obviously effected.

SEE THE B & G HANDBOOK FOR COMPLETE DESIGNING DATA

HOT WATER SYSTEMS AND SPECIALTIES

B & G Relief Valves

For relieving excess boiler pressures in hot water heating systems, and in the lines of service water systems. B & G Relief Valves have the design features which assure dependable service.

B & G Reducing Valves

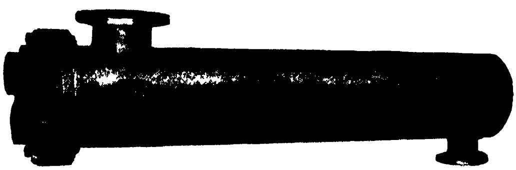
Fast operating valves for keeping hot water heating systems properly filled. Easily adjusted to meet varying building heights. Also high pressure reducing valves for protection of plumbing fixtures.

←B & G Compression Tanks

Essential equipment for closed hot water heating systems. Expansion of heated water is taken up by the tank, providing a cushion of compressed air against sudden pressures and water hammer shocks. On high temperature installations, the Tank develops sufficient pressure to prevent boiling of the water in the system.

B & G Indirect Water Heaters→

Any steam, vapor or hot water heating boiler can be equipped with a B & G Indirect Water Heater With the proper electrical controls, the Heater will furnish an ample supply of hot water, *winter and summer*, at very low operating cost Heater must be used with a storage tank of suitable capacity.

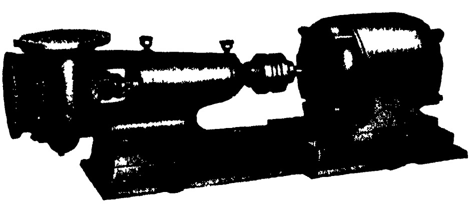


←B & G Type "SU" Instantaneous Water Heaters

For heating water with steam. Ideal for industrial plants or wherever large volumes of hot water are required continuously for service water supply or process work. No storage tank required—the large heat transfer surface in these units heats water instantly as needed.

B & G Steam Convertors ↓

Extensively used when steam is required in the factory for power or process work, but where the benefits of *mechanically circulated hot water* are desired for the heating system. Steam is passed through the convertor shell, heating the water circulated through the tubing.

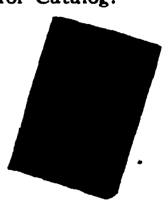


←B & G Centrifugal Pumps

Design and construction based on years of experience in the industrial field. Rugged, compact units—built to stand up under the strain of continuous operation. Available with semi-open or enclosed impellers—motors flexible coupled or integral with pump. Send for Catalog.

Send for Your Copy of the B & G Handbook→

A single authoritative source of information on the design and installation of Forced Hot Water Heating Systems and Service Water Heating Systems. Packed with data and tables of every day value. Your copy will be sent on request.



Taco Heaters, Incorporated

342 Madison Avenue, **New York 17, N. Y.**

TACO HEATERS OF CANADA, LTD., 24 Adelaide St , W., **Toronto**

TACO ABBOTT SYSTEM Taco **HEATERS**

THERE is a storage or tankless, Biltin or external, type Taco indirect water Heater for every job—for use with patented Taco-Abbott System.

Biltin Taco Heaters, which are standard equipment on leading heating boilers, are catalogued only in the boiler catalogues. For additional information on Biltin Tacos write boiler manufacturer or Taco.

In addition to this abbreviated information on external residential Taco Heaters, complete catalogue information is available for apartment house, hotel and housing projects.

TANKLESS TACO Nos. 12, 14, 15 and 18

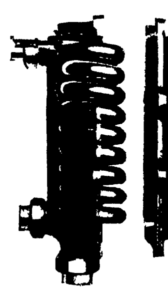
Size of Taco	Capacity Gallons Heated from 40° F to 140° F.			Boiler Conn. Inches	Dom. Water Conn Inches	Height by Length Inches
	†Boiler Water at 180° F.		Boiler Water at 212° F. Boiler Steaming			
	Per Min.	Baths	Per Min.			
12	3½	1	5	2	½	9½ x 16
14	4	1-2	6	2	½	14 x 17 9/16
15	5	1-3	7½	2	½	14½ x 20¼
18	6	2-4	9	2½	¾	17 x 22

†Automatically fired installations—Taco-Abbott System for year 'round operation

STORAGE HEATERS

Size of Taco	CAPACITIES				
	Gallons in 3 Hours Heated from 40° F. to 140° F.			Sq. Ft Hot Water Radiation 175° Water	
	Taco Below Water Line		Taco on Steam of 0 Lbs. Gauge Pressure	Taco* Below Water Line Boiler at 212° F.	Taco** on Steam of 0 Lbs Gauge Pressure
	Boiler Water at 212° F.	†Boiler Water from 160° to 190° F.			
A	B	C	D	E	F
		DOMESTIC TACO			
1	40	.	75	30	50
1A	52	30	100	40	65
1B	66	30 to 40	125	50	85
2	82	40 to 52	150	60	100
2A	100	52 to 66	175	75	125
2B	120	66 to 82	225	90	150
2C	144	82 to 100	275	105	175
3	160	100 to 120	300	120	200
3A	200	120 to 144	375	150	250
		MULTI-COIL TACO			
		Boiler Water 180°			
MC150	250	150	475	210	310
MC180	300	180	560	250	375
MC240	400	240	750	330	500
MC300	500	300	940	420	625
MC360	600	360	1150	500	750
MC450	750	450	1410	630	940
MC600	1000	600	1620	710	1130

Tankless Taco No 14

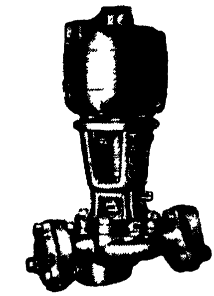


Domestic Taco

*Based on 120° F. inlet temp. to heater and 160° F. outlet temp.

**Based on 140° F. inlet temp. to heater and 180° F. outlet temp.

TACO WARM WATER HEATING SPECIALTIES



Taco Hy-Duty Circulator

Taco Warm Water Heating Specialties

Used with both one and two-pipe warm water heating systems. Layouts and specifications gladly furnished.

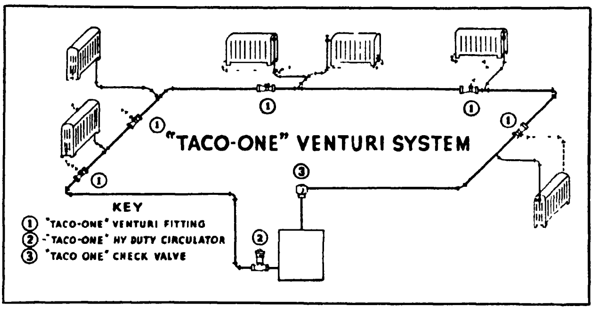
"Taco-One" Venturi System

A forced circulating warm heating system using a single pipe main from boiler to radiators and back again. Small pipes and one pipe main make a neater job, reduce installation costs. Radiators can be placed above and below main. 85 per cent of residential jobs need only one circuit which requires no balancing valves. This revolutionary heating system is made possible by the Taco Venturi Fitting and remarkable Taco Hy-Duty Circulator.

Taco Hy-Duty Circulator—Is more efficient (and slightly more expensive) than conventional circulators Designed to work with the Venturi Fitting in "Taco-One" Venturi System, it costs less to operate and lasts longer than circulators "built at a price."

Taco Venturi Fitting—Acts as a suction pump in which the only moving part is water. Water does the trick by producing a vacuum pull that draws water through each radiator, giving positive uniform circulation.

Taco Venturi Fitting
(Cross Section)



DESIGN TABLE FOR "TACO-ONE" VENTURI SYSTEM

Maximum Radiation for Various BTU Emissions and Water Temperatures			Maximum Length of Longest Circuit in Feet	Flange Size of Taco Hy-Duty Circulator	Maximum Number of Venturi Fittings to be Used on Systems		Main Sizes			Size of Venturi Fittings	Radiator Branch and Riser Pipe Size See Note A	Maximum Amount of Radiation per Venturi Fitting for Various BTU Emissions and Water Temperatures					
							Single Circuit	Double Circuit				Single Radiator Above Main			Total Two Rads Above Main or One Below Main		
150 BTU 175° Water	200 BTU 200° Water	240 BTU 215° Water			One Circ	Double Circuit	Circuit	Trunk	Circuits			150 BTU 175°	200 BTU 200°	240 BTU 215°	150 BTU 175°	200 BTU 200°	240 BTU 215°
500	375	308	100	1"	12	10	1"	1¼"	1"	1"x½	½"	75	56	46	60	45	38
1000	750	625	125	1¼	12	10	1¼"	1½	1¼"	1¼x½	½"	100	75	63	80	60	50
1000	750	625	125	1¼"	12	10	1¼	1½	1¼	1¼x¾	¾"	150	112	94	120	90	75
1000	750	625	150	1½"	12	10	1½"	2"	1½"	1½x½	½	100	75	63	80	60	50
1000	750	625	150	1½"	12	10	1½"	2"	1½"	1½x¾	¾	150	112	94	120	90	75

Increase laterals exceeding 8 ft. one size from main to heel or riser. In most cases, not more than half total radiation should be installed below main.

In no case should laterals to radiators below main exceed 9 ft.

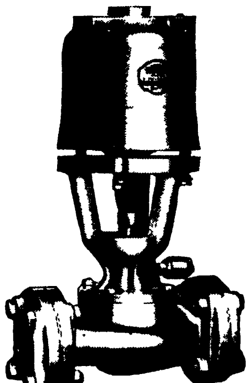
Care must be taken with radiators below main on two-circuit jobs. Send complete information on such jobs to Taco Heaters, Inc., for layout.

H. A. Thrush & Company

Peru, Indiana

Representatives in Principal Cities

FORCED CIRCULATING THRUSH FLOW CONTROL SYSTEM OF HOT WATER HEATING AND HEATING SPECIALTIES

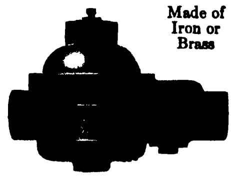


Patent Nos 2,054,009, 2,111,441, 2,257,867, 2,356,482

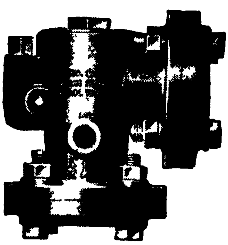
Flexible, economical of fuel, forced circulating Thrush Summer-Winter System of Hot Water Heat is the most satisfactory way to heat buildings. Thrush Systems, Water Circulators, Water Heaters, Pressure Relief and Pressure Reducing Valves and other fuel-saving heating specialties are available. Write today for information or engineering assistance.

THRUSH WATER CIRCULATORS

Five sizes, 1 in., 1¼ in , 1½ in., 2 in. and 3 in., for circulating water in Heating or Domestic Water Systems Save fuel, insure uniform heating.



Made of Iron or Brass



Patent Reissue No 19,873

THRUSH FLOW CONTROL VALVES

For use with Thrush Water Circulators

Six sizes, 1 in., 1¼ in., 1½ in., 2 in., 2½ in. and 3 in. Work automatically by pressure head, generated by Thrush Circulator. This patented valve prevents circulation when not required

THRUSH PRESSURE REDUCING VALVES

Types for High and Low Pressures

Sizes, ¼ in., ⅜ in., ½ in., ¾ in. and 1 in. High pressure reducing valves for protecting house plumbing and heating equipment from excessive city line pressures. Low pressure reducing valves to reduce pressure of water entering heating system and maintain water supply in system automatically

Number 4 Illustrated

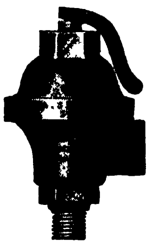
THRUSH LOW PRESSURE RELIEF VALVES

Protect heating boilers from excess pressures. Made in angle or straight types, of iron or brass, sizes ½ in., ¾ in. and 1 in. Unfailing dependability has been proved by over a quarter of a century of successful operation.

THRUSH AIR-TIGHT PRESSURE TANKS

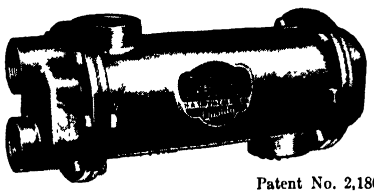
An essential part of every hot water heating system. Conserves water and fuel, because the heated water expands into the Thrush Pressure Tank and returns to the system as it cools Adds to the continued operating efficiency of the heating system over a long period of time.

ALL THRUSH RELIEF VALVES COMPLY WITH A.S.M.E. CODE AND LISTED AS STANDARD BY UNDERWRITERS' LABORATORIES



THRUSH HIGH PRESSURE RELIEF VALVES

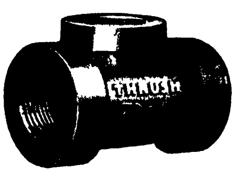
Protect water supply or range boilers and gas or electric water heaters from excess pressure. Made in angle or straight types, of iron or brass, sizes, ½ in., ¾ in. and 1 in., for pressure relief only or combination pressure and temperature relief.



Patent No. 2,180,620

THRUSH WATER HEATERS

Highly efficient heat exchangers or converters. Sixteen sizes, for Hot Water or Steam. Pressure up to 150 lb Straight tubes readily cleanable Provide Domestic Hot Water at low cost. Also used industrially for heating or cooling liquids.



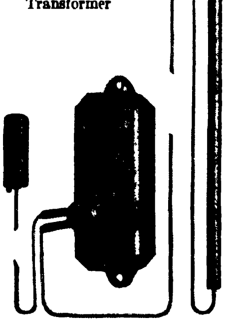
THRUSH SUPPLY TEES FOR ONE PIPE SYSTEMS

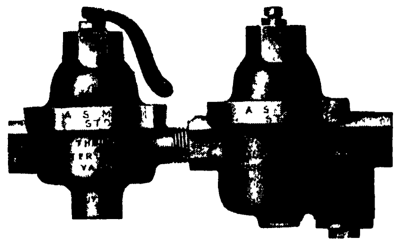
Assure positive diversion to radiators. Made in four sizes, 1 in., 1¼ in., 1½ in. and 2 in., with branch outlets of ½ in., ¾ in. or 1 in.

No. 210 THRUSH DIFFERENTIAL TEMPERATURE CONTROL

Special dual acting thermostat—one bulb inside and one bulb outside building. Automatically maintains correct indoor temperature with any weather change. A real fuel- and money-saver, especially for apartment houses, or groups of industrial, tourist camp or other buildings.

Requires No 198 Thrush Relay Transformer

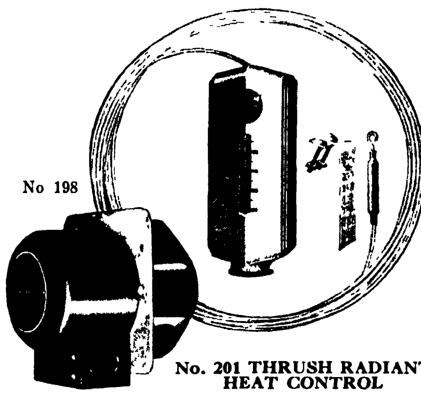




THRUSH DUAL CONTROL UNITS

Provide automatic pressure relief, reduce line pressure, automatically fill and maintain water supply in hot water heating system. Made in four types, brass or cast iron, ½ in. or ¾ in.

No 198



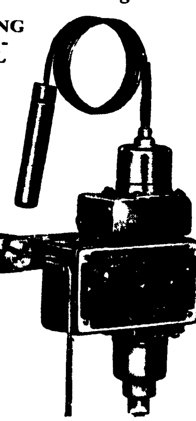
No. 201 THRUSH RADIANT HEAT CONTROL

Automatically maintains room temperature within a fraction of a degree. Controls both room and water temperature in the radiators, compensating to prevent variation in room temperature or a lack of radiant heat. Requires No. 198 Thrush Relay Transformer for low voltage.

THRUSH MODULATING BOILER WATER TEMPERATURE CONTROL

No. 203 with dual bulbs, automaticcally varies boiler water temperature to meet all demands for heat. Maintains correct boiler water temperature, thus saving fuel.

Other electrical water temperature controls, No. 205 immersion and No. 206 clamp-on type, available.



WRITE FOR COMPLETE CATALOG

Buffalo Pumps, Inc.

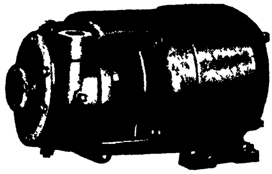
450 Broadway, Buffalo, N. Y.

Sales Representatives

ALBANY 7, N Y, R B Taylor, 1303 Standard Bldg.
ATLANTA, GA, J J O'Shea, 305 Teckwood Drive
BALTIMORE 2, MD, C A Conklin, III 508 St Paul St
BOSTON 76, MASS, E D Johnson, 507 Main St Melrose Station
CHICAGO 6, ILL, Emmert & Trumbo, 20 N Wacker Drive
CINCINNATI 2, OHIO, F W Twombly, 626 Broadway
CLEVELAND 13, OHIO, T A Weager, 418 Rockefeller Bldg
DAVENPORT, IOWA, D C Murphy Co, Inc, 305 Security Bldg
DALLAS 1 TEXAS, T H Anspacher, 1801 Tower Petroleum Bldg
DENVER 17, COLO, Stearns Roger Mfg Co, 1718 California St
DES MOINES 9, IOWA, D C Murphy Co, Inc, 214 Old Colony Bldg
DETROIT 16, MICH, Coon DeVisser Co, 2051 W Lafayette Blvd
KITCHENER, ONTARIO, Canada Pumps, Ltd
LOS ANGELES 13, CALIF., F Halladay, 804 Pershing Sq Bldg
MINNEAPOLIS 2, MINN, E F Bell, 2102 Foshay Tower
NEW ORLEANS 12, LA, Devlin Brothers, 1003 Maritime Bldg
NEW YORK 7, N Y, Koithan & Johnson, 39 Cortlandt St.
NEWARK 2, N J, G C Norman, 27 Washington St
OMAHA 2, NEBR, Wain Engineering Co, 300 Brandeis Theatre Bldg
PHILADELPHIA 2, PA, Davidson & Hunger, 702 Cunard Bldg
PITTSBURGH 22, PA, H Lee Moore, 431 Fulton Bldg
SAN ANTONIO 6, TEX, Langhammer Rummel Co, 436 Main Ave
SEATTLE 4, WASH, Consolidated Service & Supply Co, 71 Columbia St
ST LOUIS 2, MO, J W Cooper, 2726 Locust St
TOLEDO 2, OHIO, Carl N Eyster Co, 1922 Linwood Ave
WASHINGTON 5, D C, G S Frankel, 512 Woodward Bldg.

PRODUCTS—A complete line of Single and Multi-stage Centrifugal Pumps and Special Pumps for use in all types of heating and air conditioning installations.

Buffalo Single Suction Close-Coupled Pumps

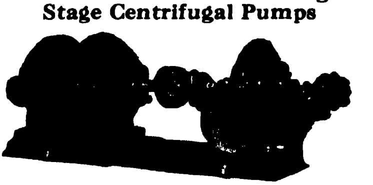


This pump is close-coupled to electric motor, eliminating the necessity for bearings. The impeller is overhung on the motor shaft, providing a compact, easily-serviced unit. Permanent alignment is assured and the pump mounted in this manner requires very little space.

Buffalo Close-Coupled Pumps are suitable for handling hot water with low submergence on suction, or for operating with suction lift as high as 25 ft.

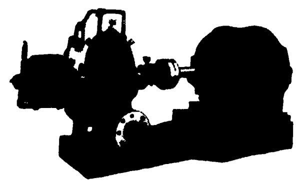
These pumps are also available in special alloys.

Buffalo Double Suction Single Stage Centrifugal Pumps



For general service where clear water is handled you will get top performance with these pumps. They embody all of the accepted modern features of centrifugal pump design. Capacities range from 10 to 50 thousand U.S. gallons per minute.

Buffalo Self-Priming Single and Double Suction Centrifugal Pumps



Now available with positive self-priming device built with the pump. This primer is built under license from the Nash Engineering Company and is fully covered by patent.

Buffalo Self-Priming Pumps offer these advantages: (1) All working parts are above the liquid to be pumped. (2) There is complete access to all parts of installation. (3) Rotors are balanced—vibrationless. (4) Buffalo Self-Priming Pumps are very quiet—no long shafts to vibrate and fewer bearings. (5) Constant positive prime obtained without foot valves

Buffalo Automatic Sump Pumps

Buffalo Sump Pumps are self-contained and have unusually high efficiencies thus permitting the use of small motors. Ball bearing thrust and enclosed shaft especially adapt these pumps for their service.

Chicago Pump Company

2330 Wolfram Street **BRUnswick 4110** **Chicago**

PRODUCTS—Return Line Vacuum Heating and Boiler Feed Pumps, Condensation, House, Booster, Fire Pumps, Circulating, Brine, Sewage, Bilge, Sludge, Pneumatic and Tankless Water Supply Systems and Automatic Alternator for Duplex Sets of Pumps.

"CONDO-VAC"

Return Line Vacuum and Boiler Feed Pump for Heating Systems

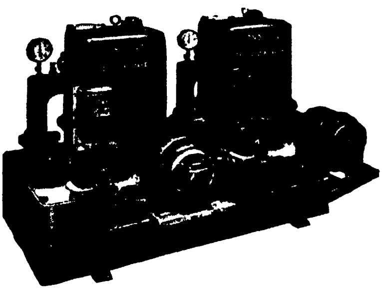


Fig 2102—Duplex "Condo-Vacs" with Duplex Double Automatic Control

No vacuum on stuffing boxes, ample clearance in rotating member. It costs less to operate a "Condo-Vac." "Condo-Vac" reduces corrosion in piping and boiler to minimum—because pump does not take in air from atmosphere and entirely eliminates all air coming back from system. "Condo-Vac" is quiet, has a low inlet, entirely automatic, fool-proof, easy to maintain. *Ask for bulletin 270.*

Close-Coupled Pumps

Boiler Feed, Circulating, Tank Filling, Water Supply

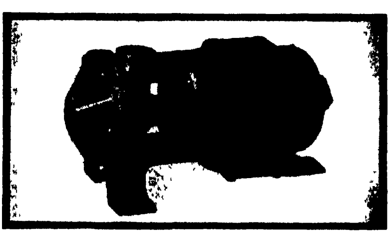


Fig 2130—Close-Coupled, side suction pump Capacities range from 3 to 600 Gpm against heads up to 189 ft Motors from 1/6 to 20 Hp. Discharge 1 to 3 in. Closed and open type impellers. Bulletin 108.

"Sure-Return" Condensation Pump

for Low and Medium Pressure, and Systems up to 75,000 Sq Ft Radiation

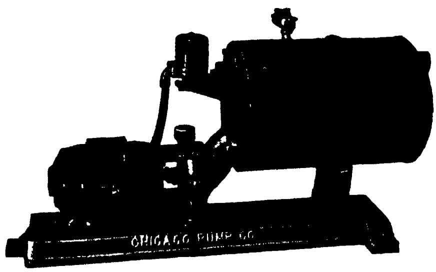


Fig 1946

"Sure Return" Condensation Pumps and Receivers are built for systems up to 75,000 sq ft of direct radiation and for low and medium pressures. Built in either single or duplex units Duplex units are alternated in their operation by the Automatic Alternator. *Complete data in Bulletin 250.*

Vertical Condensation Pumps

for Low and Medium Pressure for Systems from 500 to 100,000 Sq Ft Radiation

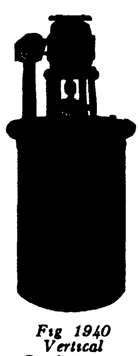


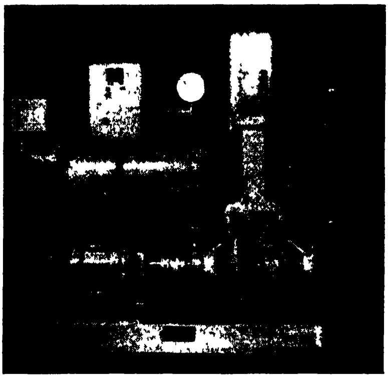
Fig 1940 Vertical Condensation Pump

The vertical condensation pump is designed to receive returns from lowest radiation. The receiver is placed underground—an ordinary hole sufficing if necessary — and requires very little floor space. Unit is shipped complete, easy to install, assembled so as to prevent steam leaks. Special bearings will stand up under hot water for several years. A special float mechanism is guaranteed not to leak or stick in stuffing box. *Complete data and description in Bulletins 245, 253 and 255.*

The Nash Engineering Company

234 Wilson Road **South Norwalk, Conn., U. S. A.**

Sales and Service Offices in all Principal Cities



Return Line Vacuum Heating Pump

Standard with the heating industry for over seventeen years. Removes air and condensation from return lines of vacuum steam heating systems, discharging air to atmosphere and returning water to the boiler.

Two independent units are combined in a single casing—an air unit and a water unit. Impellers of both are mounted on the same shaft. Pump is bronze fitted throughout.

Supplied direct connected to standard electric motors, for belt drive, or for steam turbine drive. For continuous or automatic operation. Standard in capacities up to 300,000 sq ft E.D.R. Larger units special. Bulletins Nos. 307, 308, 309 and 310 on request.

Vapor Turbine Vacuum Heating Pump

Jennings Vapor Turbine Heating Pumps combine all advantages of the standard return line heating pump with a new type of drive, a specially designed low pressure turbine which operates directly on steam from the heating mains on any system, requiring a differential of only 5 in. of mercury, and returns that steam to the heating system with practically no heat loss.

This pump affords the safety and economy which goes with continuous condensation return and steady vacuum, and at no cost for electric current. Furnished standard in capacities up to 65,000 sq ft E D R. Larger units special. Bulletin No 290 on request

Condensation Pump and Receiver

Removes the condensation from radiators in return line steam heating systems, particularly radiators set below the boiler water line level, and pumps the condensation back to the boiler. Pump is bronze fitted with enclosed centrifugal impeller of improved design. By making the pump casing a part of the return tank, and bolting the motor base to the tank, floor space is conserved. The rectangular construction permits installation in a corner against the wall.

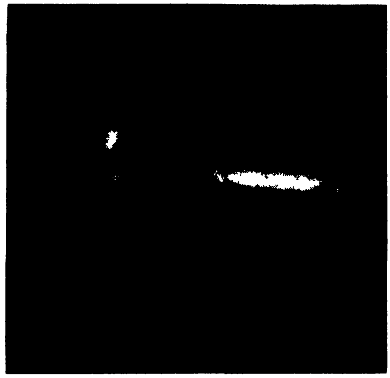
These pumps are furnished in standard sizes with capacities ranging from 1½ to 225 gpm of water. For serving up to 150,000 sq ft of equivalent direct radiation. Bulletin No. 319 on request

The Nash Engineering Company

234 Wilson Road **South Norwalk, Conn., U. S. A.**

Sales and Service Offices in all Principal Cities

Centrifugal Pump



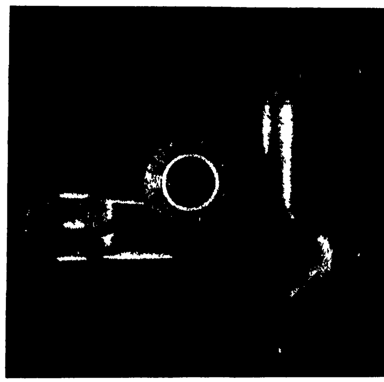
Made in standard and suction (self-priming) types. For circulating hot and cold water; boosting city water pressure; handling water in air washing and conditioning; handling ash sluicing water, etc.

Compact—motor armature and pump impeller are mounted on the same shaft. Simplified—no bearings in pump casing, one stuffing box. Accessible—impeller removable without disturbing piping or shaft alignment.

Self-priming types will handle air or gas continuously with liquid being pumped, and can be operated intermittently without foot valve.

Supplied in 1, 1¼, 1½, 2, 3, 4, 6, and 8 in. sizes, with capacity up to 2000 gpm. Heads up to 300 ft. Bulletin No. 322 on request

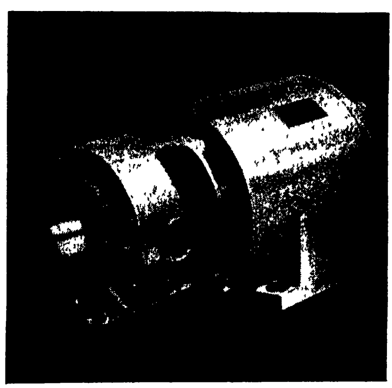
Suction Sump and Sewage Pumps



Jennings Sump Pumps are self-priming centrifugals for handling seepage water and liquids reasonably free from solids. Sewage Pumps are equipped with non-clog type impeller for liquids containing solids. Suction piping only is submerged Centrifugal impeller and vacuum priming rotor are mounted on same shaft that carries rotor of the driving motor, forming a single moving element, rotating without metallic contact.

Will handle air or gas with liquid being pumped, and because of self-priming feature are installed entirely outside of pit, affording perfect accessibility for inspection or cleaning. Capacities to meet all requirements. Bulletins Nos 159, 161, and 338 on request.

Air Compressor and Vacuum Pump



Nash Air Compressors operate on a unique and different principle. The one moving part rotates in casing without metallic contact. There is nothing to wear, and no internal lubrication.

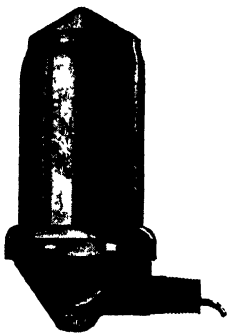
Nash Compressors deliver absolutely clean air; ideal for agitation of liquids, pressure displacement, and handling gases. Vacuum pumps ideal for priming pumps, blood sucking pumps in hospitals, and wherever non-pulsating vacuum is required.

Pressure 75 lb or vacuum 27 in. of mercury. Furnished for any capacity; special for higher vacuums and pressures. Bulletins Nos. 282, 325, 331 and 337 on request.

ANDERSON PRODUCTS
INC.
CAMBRIDGE 39, MASSACHUSETTS

Vent-Rite Controlled Venting Radiator Valves · Vent-Rite No. 66 Control Valves · Vent-Rite Unit Heater Valve

KEEP RADIATORS "ALIVE" IN '45 WITH VENT-RITE AIR and VACUUM RADIATOR VALVES



No.2

REPAIRABLE

All Vent-Rite Valves are Repairable

VENT-RITE air and vacuum radiator valves are built for both one and two pipe steam services in a wide variety of types, sizes, outlets and venting capacities.

Heating studies of the last few years have proved that a great percentage of valve trouble comes from the accumulation of dirt within the valves. Every Vent-Rite valve can readily be taken apart, cleaned, reassembled, and adjusted, and the process is a simple one. In the rare case where a part may have become defective, the valve can be sent to the factory for complete reconditioning Only Vent-Rite offers these advantages

Because every Vent-Rite Valve has an exceptionally wide and complete range of venting rates, it is possible to obtain the correct venting rate in each radiator in a system, regardless of size or location. Balanced Radiation results. Adjustment of the venting rate at the individual valves is easily accomplished through a stream-lined, convenient, adjusting device The Vent-Rite Line includes Nos. 1, 51, 3, 5A and 55 (Non-Vacuum), 2, 62, 4, 6A, 66.

VENT-RITE PIONEERED

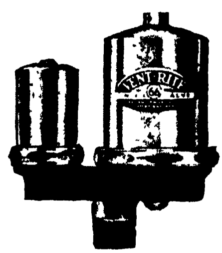
- CONTROLLED VENTING
- BALANCED RADIATION
- REPAIRABLE VALVES
- VENT-VAC METHOD

THE VENT-VAC METHOD

The Vent-Vac Method provides more even room temperatures by continuing the distribution of steam between firing periods The steam is available through the use of heat left in the boiler, and it is distributed to the points of greatest heat loss. To insure fast, uniform distribution of steam during the firing periods, it breaks the vacuum used between firing periods for this purpose This "breaking" of the vacuum occurs as soon as firing starts, restoring the system to atmospheric pressure Vent-Rite Vacuum Valves, and a Vent-Rite Control Unit are used. The system is simple, economical, and amazingly effective. Vent-Rite Control Units not only create vacuum in the system between firing periods, but also limit the amount of vacuum that can be created to the point beyond which the distribution of excessively expanded vapor would be inefficient.

VENT-RITE CONTROL VALVES

Vent-Rite Control Valve No. 66 is the heart of the Vent-Vac Method of steam control for automatically-fired, one-pipe systems. It takes the place of a main line vent, limits the amount of vacuum created and breaks the vacuum at the beginning of the firing period. It is entirely mechanical. With the Vent-Vac Method, using a No. 66 Control Valve, a system is "Vacuum" between Firing periods, "Non-Vacuum" during Firing, combining the best of both systems assuring "Balanced Radiation."



No. 66

Barnes & Jones
INCORPORATED

129 Brookside Avenue **Boston, Mass.**

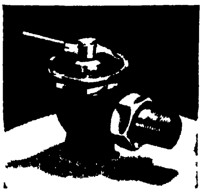
New York Office: 101 Park Avenue

Barnes & Jones Vapor and Vacuum Systems of Steam Heating; Modulation Valves; Adjustable-Orifice Radiator Valves; Packless Quick-Opening Radiator Valves; Thermostatic Radiator Traps; Thermostatic Trap Replacement Units; Condensators (Boiler Return Traps); Float and Thermostatic Traps; Strainers; Damper Regulators; Gages; Systems of Zone Control for Steam Heating.

Complete Catalog on Request

To comply with limitation orders of W P B, cast-iron has been substituted for critical materials wherever possible B & J quality has been maintained where required changes have been made.

Modulation Valves, Type K—Packless Quick Opening Valves, Type F



Types K and F Valves have non-tarnishable indicating dial, non-rising stem, renewable disc seat. Tail piece extra heavy. Extra long to facilitate installation. Three models: lever handle, wheel handle, lock shield. Type F Valve furnished with wheel handle only.

Type K Valve

Size	½″	¾″	1″	1¼″
Cap. Sq Ft Rad.*	30	60	100	180

Type F Valve

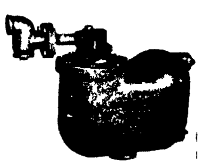
Size	½″	¾″	1″	1¼″	1½″	2″
Cap Sq Ft Rad *	30	60	100	180	270	400

*Based on 2 oz pressure differential.

Adjustable Orifice Valves, Type H

May be adjusted for different capacities after installation. At all times provides indication of the adjustment Operation is quiet. Unauthorized tampering with adjustment is virtually impossible.

Condensators



For returning water of condensation to boiler from open return line systems independently of boiler pressure, without change in operating conditions, air binding, or admitting steam to the return side.

No.	31	32	33	34	35	36
Cap. Sq Ft Rad *	700	1600	3500	6000	10,000	16,000

Thermostatic Radiator Traps

Sturdily made to precision standards. Sensitive in operation. Provide instant discharge of air and water, prevents passage of steam. Contains unique Cage Type Thermostatic Unit, which carries its own thermostatic element, valve piece and valve seat, factory calibrated and locked in correct adjustment.

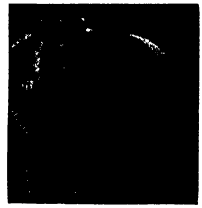
Trap No.	120	12	123	124	134	13	14
Inlet Tapping	½″	½″	½″	½″	¾″	¾″	1″
Outlet Tapping	½″	¾″	½″	¾″	¾″	¾″	1″
Cap. Sq Ft Rad.*	200	200	400	400	400	700	1200

*Based on 1½ lb pressure differential.

Thermostatic Radiator Cage Replacement Units

Offer complete and reliable trap renewal in practically every make of thermostatic trap. You simply (1) remove the old cover and unit, (2) insert the new Barnes & Jones Cage Unit, (3) replace the cover, and the old trap will operate with its original efficiency.

Float and Thermostatic Traps



Handle large and sudden condensation loads. Large air and water capacity. Large float assures instant opening of the discharge valve. Cage Type Thermostatic Unit assures quick elimination of air.

Trap No.	41	42	43	44B	45B
Inlet Tapping	¾″	1″	1¼″	1½″	2″
Outlet Tapping	¾″	1″	1¼″	1½″	2″
Cap. Lg. Water per Hour*	200	500	1200	2400	5000

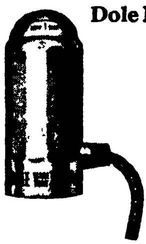
*Based on 2 lb pressure differential.

The Dole Valve Company

Main Offices and Factory: 1933 Carroll Avenue, **Chicago 12, Ill.**

THE ALL STAR LINE **AIR AND VACUUM VALVES**

Selecting the right vent for a particular purpose is your assurance of the utmost efficiency and economy from one pipe steam heating systems. The Dole line covers every venting need and offers a complete choice for every purpose.

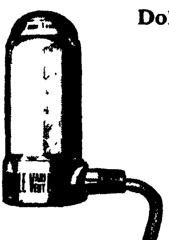


Dole No. 1A Vari-Vent Air Valve

Modern gas, oil or stoker fired one pipe steam systems require QUICK venting. This radiator valve lets air escape twice as fast as ordinary valves and balances the flow of steam at the first "breath" of boiler pressure. Adjustable vari-vent feature gets air out of those "far away" radiators as quickly as those close to the boiler.

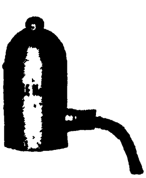
Dole No. 3 Air Valve

Vents radiators of hand fired gravity steam heating systems. Double shell construction provides separate passages for air and condensation—extra large float defeats spitting or water leakage. Complete venting assured at pressures up to 10 lbs



Dole No. 2B Vari-Vent Vacuum Valve

Adjustable radiator valve for "vacuumizing" and balancing gravity steam heating systems. Patented Dole bellows vacuum seal locks out air after it has been once expelled from the system. Easily adjusted vari-vent feature assists in equalizing steam flow to all radiators.

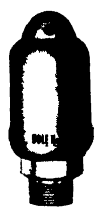


Dole No. 1933 Air Valve

Low cost valve for venting radiators of hand fired systems. Large float provides a seal against condensation to stop spitting.

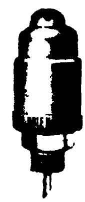
Dole No. 1B Vari-Vent Air Valve

Balances the flow of steam to convectors, either cast iron or copper, of automatically fired systems.



Dole No. 1C Quick Vent Float Valve

Vents mains and speeds flow of steam to radiators of automatically fired systems Extra large venting port.

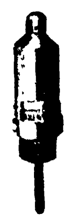


Dole No. 5 Quick Vent Float Valve

Vents steam mains on hand fired systems. Positive seal against water.

Dole No. 4 Quick Vent Valve

For quick venting mains that end 18 in. or more above the boiler water line.

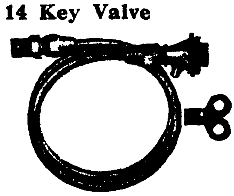


Dole No. 103 Vacuum Valve

For venting convectors, ceiling radiators and pipe coils of "vacuumized" gravity steam systems.

Dole No. 6B Vacuum Valve

Vents the mains of "vacuumized" one pipe steam systems Prevents the return of air. Closes against water



Dole No. 14 Key Valve

Low cost venting device for concealed radiators and convectors of hot water heating systems. Protects panel fronts from rusty water stain.

Write The Dole Valve Company for complete catalog and handy selector chart which indicates the Dole Air or Vacuum Valve most suited for a particular need.

C. A. Dunham Company

Administrative and General Offices

450 E. Ohio Street, **Chicago 11, Ill.**

Factories: MARSHALLTOWN, IOWA; MICHIGAN CITY, IND.; TORONTO, CANADA; LONDON, ENGLAND
TORONTO 4, 1523 DAVENPORT ROAD. **LONDON**, MORDEN ROAD, S.W.19

THE DUNHAM DIFFERENTIAL HEATING SYSTEM

The system is a simple two-pipe system in which all the essentials of circulation, distribution and control are co-ordinated. Control of the temperature of the steam is accomplished by controlling the pressure or vacuum of the steam in the supply piping and radiators to balance exactly the heat input with building-heat-loss.

VAPOR AND VACUUM HEATING SPECIALTIES

Dunham Thermostatic Radiator Trap for Operating Pressures up to 15 Lb Gage

Bronze body and cap. Thermostatic element phosphor bronze. Valve and valve seat cuprous alloys. Thermostatic elements interchangeable in covers without gages. Covers and disc assemblies interchangeable with standard traps.

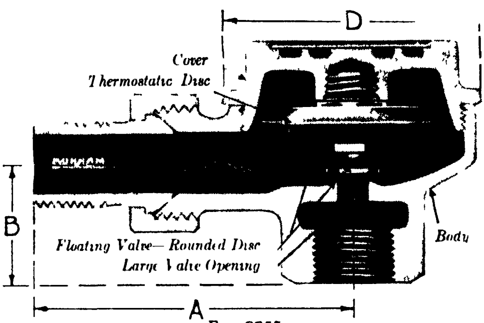


Fig 3355

Size No	Size Conn Inches	Pattern	Cap * Sq Ft EDR	Net Wgt Lb	Dimensions, Inches A	B	D
1B	1/2	A.P.	200	1 5/8	3 1/4	1 1/8	2 7/16
2C	3/4	A.P.	400	2 1/4	3 5/16	1 1/8	3 1/16
3B	3/4	A.P.	700	2 1/4	3 5/16	1 1/8	3 1/16

*Ratings are based upon 1/4 lb condensation per sq ft of equivalent direct radiation per hour and a 1 1/2 lb pressure differential

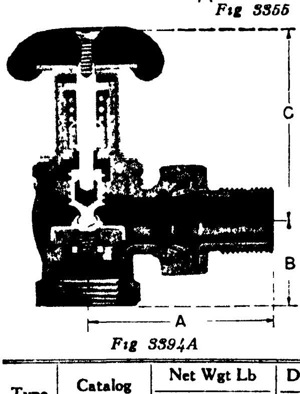


Fig 3394A

740 SERIES DUNHAM RADIATOR VALVE (Spring Packed)

Designed for low pressure steam heating service. Bodies are brass castings, rough finish. The valves are equipped with heavily constructed brass union nuts and nipples. All pipe threads and tappings are carefully machined and checked to standard gages. Non-rising stem, requires less than one turn of handle to open the valve fully. Dial shows direction and amount of opening. Heavy bronze spring keeps a constant pressure on a special graphited asbestos composition ring to maintain a tight seal around the valve stem.

Type	Catalog Number	Net Wgt Lb Angle	SW RH-LH	Dimensions, Inches A	B	C	Type	Catalog Number	Net Wgt Lb Angle	SW RH-LH	Dimensions, Inches A	B	C
740	SWI 1/2A	1 5/8	1 3/4	2 3/8	1 1/8	3 1/8	740	SWI 1 1/4A	3 1/8	3 1/2	3 1/2	1 5/8	3 3/4
741	SWI 1/2S						741	SWI 1 1/4S					
742	SWI 1/2R						742	SWI 1 1/4R					
743	SWI 1/2L						743	SWI 1 1/4L					
740	SWI 3/4A	1 15/16	2 1/8	2 13/16	1 1/4	3 1/8	740	SWI 1 1/2A	4 1/4	4 3/4	3 13/16	1 7/8	3 7/8
741	SWI 3/4S						741	SWI 1 1/2S					
742	SWI 3/4R						742	SWI 1 1/2R					
743	SWI 3/4L						743	SWI 1 1/2L					
740	SWI 1A	2 1/2	2 13/16	3 1/16	1 7/16	3 5/8	740	SWI 2A	6 1/8	7	4 3/8	2 1/4	4 1/8
741	SWI 1S						741	SWI 2S					
742	SWI 1R						742	SWI 2R					
743	SWI 1L						743	SWI 2L					

Note: Dimensions for straightway and corner patterns upon request.

DUNHAM THERMOSTATIC STEAM TRAPS—TYPE TH

For Working Pressures 25 Lb to 100 Lb

Construction—Type TH traps are the thermostatic fluid expansion type. The trap consists of two principle parts, a body with renewable valve seat and a cover containing the expansion thermostat. Permanent adjustment for correct operation is built into the element. The covers may be removed from the trap body while hot without danger to the thermostatic element. The valve is swiveled to insure its seating tightly without causing localized stresses on the element. Body, cover, nut and nipple are of brass; valve and seat are of special heat treated stainless steel; thermostatic element is formed from monel metal sheet.

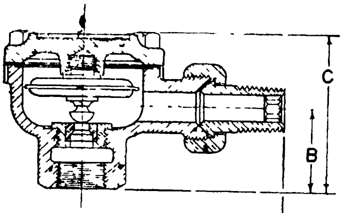


Fig 8427A

Trap Size	Size Conn. Inches	Dimensions, in Inches			Working Pressure (lb per Sq In Gage) *Capacity Pounds Condensate per Hour			
		A	B	C	25	35	50	100
TH1	½	3¼	1⅜	2 7/16	1200	1550	1950	2600
TH2	¾	3 7/16	1 9/16	3¾	1800	2200	2600	3500
TH3	1	3⅜	1¾	3¾	2500	2900	3400	4500

*Traps shall start discharging with a temperature differential (i e difference between temperature of discharging water and that of saturated steam of same pressure as that on the trap) of about 30° to 40° F

DUNHAM TYPE OTS HEATING ELEMENT

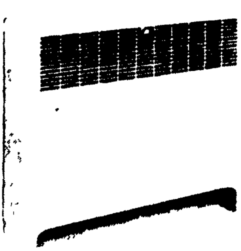
Capacity Ratings 1¼ In. OTS Heating Element

Element Number	"L" Length Inches	1 Lb Steam—65 Deg Entering Air Btu	Rated EDR
OTS-418	18	1880	7 8
OTS-424	24	2525	10 5
OTS-436	36	3815	15 9
OTS-448	48	5105	21 3
OTS-460	60	6395	26 7
OTS-472	72	7685	32 0

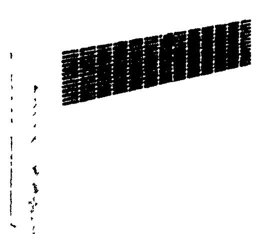
An extended surface heating element made entirely of steel. Light in weight, and has unusual heating capacity Each element is made up of 1¼ in steel pipe with No 22 gage heavy fins mechanically attached, eliminating the use of a solder bond without sacrifice of heat transfer. Each fin, when pressed on the pipe, interlocks with the preceding fin—forming an exceedingly tight and permanent mechanical joint. The complete unit is painted with heat-resisting zinc chromate black enamel See table for standard lengths Lengths up to 12 ft can be supplied by welding two standard lengths together Standard units are threaded at each end with standard pipe threads.

DUNHAM CABINET CONVECTORS

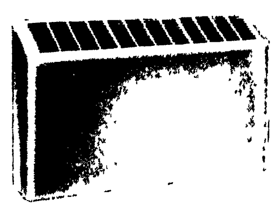
Dunham Convectors can be used on steam or vapor and gravity or forced hot water heating systems. EDR capacities with 1 lb steam, 65 deg entering air range from 14½ sq in. EDR to 110½ sq in. EDR. Made in 5 in., 7 in , and 11 in widths, 22 in., 26 in , 30 in. enclosure heights, casing lengths from 16¼ in to 61¼ in Casings are constructed of No. 18 gage steel with removable fronts Outlet grille is punched in the casing front Front outlet grille dampers can be furnished when so ordered. Heating element is copper fins on seamless drawn, round copper tubes brazed to bronze headers. These elements withstand hydrostatic test pressures of 400 lb per square inch and are suitable for operation on 150 lb steam or water working pressure.



Type FCF

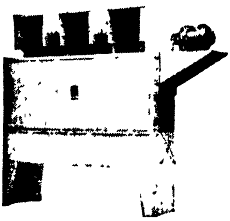


Type WCF

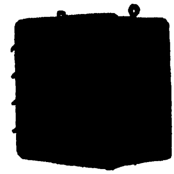


Type WCST

DUNHAM UNIT HEATERS



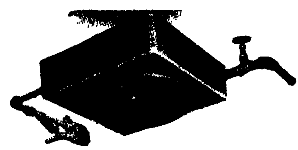
Type R—Floor Type with mixing damper



Type V—Horizontal propeller fan type. Built in various sizes up to 1200 sq ft EDR

Types C and R are essentially industrial type units designed for heating large spaces. Type C, discharging large volumes of heated air downward to working levels distributes heat evenly over large areas.

Type R is available in various types of mounting as shown. Various appurtenances satisfy requirements such as, mixing dampers, by-pass dampers and filter sections for heating and ventilating or for heating only. All units have belt driven centrifugal type fans using constant speed 1750 rpm motors. All essential easily accessible. Complete heating element and complete fan and shaft assembly can be removed through either end of heater casing Heating element is replaceable tube type, made up of seamless drawn copper tubes, wound with a copper fin in the form of a helix and metallically attached. Tubes are securely fastened into one piece semi-steel cast headers by means of brass clamping nuts

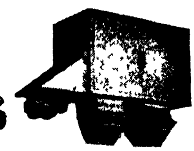


Type C—Built in various sizes up to 2000 sq ft EDR 4 types of diffusers

Type R—Floor Type with elongated nozzle

Type R—Wall Type

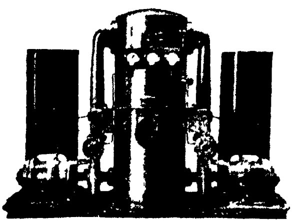
Type R—Ceiling Type



Type R—Inverted Wall Type

DUNHAM PUMPS

Tested and Rated with A S.H.V.E. Code and Code of Vacuum Return Line Heating Pump Manufacturers' Section of Hydraulic Institute.



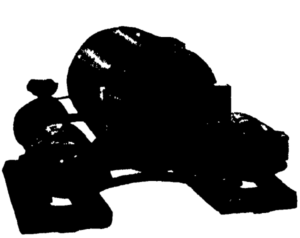
Types DV and VRD

Vacuum Pumps

Types DV and DVD—Capable of maintaining whole systems under vacuums as high as 25 in. Built in 9 sizes. Capacities 2500 to 65,000 sq ft EDR.

Types VR and VRD—Meets all code tests for air and simultaneous air and water handling capacities. No moving parts or close clearances in exhauster unit. Built in 9 sizes. Capacities 2500 to 65,000 sq ft EDR.

Condensation Pumps



Pump and motor assembled on rigid cast iron base. Bronze fitted centrifugal pump has non-corrosive shaft. Enclosed type Impeller. Liberal size ball bearings.

Type CH-Model B, Single and Duplex—Capacities 2000 to 50,000 sq ft EDR; 60 cycle d.c. or a.c. 1750 rpm; 25 or 50 cycle a.c., 1450 rpm.

Type CHH-Model B, Single and Duplex—Capacities same as type CH. Discharge pressures 20 lb for pumps 15,000 sq ft and larger; 20, 30, 40, 50 and 70 lb discharge in all sizes. 60 cycle d.c. or a.c. 3450 rpm; 25 or 50 cycle a.c., 2850 rpm.

Type CV—Capacities 2000, 4000, 6000, 8000, 10,000, 15,000, 20,000 and 25,000 sq ft EDR; and discharge pressures 10 to 50 lb. Motor shaft through cover in liberal size packing gland eliminates possibility of moisture rising into motor float mechanism. Also packless type.

OTHER DUNHAM SPECIALTIES

Float and Thermostatic Traps 8 sizes, 800 sq ft to 20,000 sq ft. **Closed Float Traps** 8 sizes, 800 sq ft to 20,000 sq ft. **High Pressure Traps** (Inverted Bucket) ½ in. and ¾ in. sizes, up to 150 lb. **Bucket Traps** ½ in. to 1¼ in. for maximum of 250 lb; ½ in., ¾ in. for 150 lbs. **Pressure Reducing Valves** Single seated (Type 340) ¾ in. to 3 in. Double seated (Type 300) 1½ in. to 10 in. Standard Flanged. **Strainers** ½ in. to 2 in.

William S. Haines & Company

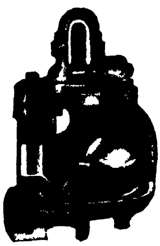
12th and Buttonwood Sts., **Philadelphia 23, Pa.**

Manufacturers of

EQUIPMENT FOR VAPOR AND VACUUM HEATING SYSTEMS

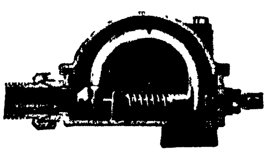
PRODUCTS—Haines Vento Radiator Traps, Medium Pressure and Blast Type Traps, Combined Float and Thermostatic Traps, Air Eliminators, High Pressure Thermostatic Traps, Boiler Return Traps, Radiator Valves.

HAINES F & T TRAPS



Designed to handle large quantities of condensation. For dripping steam mains, unit heaters, hot water generators. etc. Cannot become air bound as it has a thermostatically controlled air by pass. Sizes ¾ in., 1 in., 1¼ in.

HAINES MEDIUM PRESSURE TRAPS

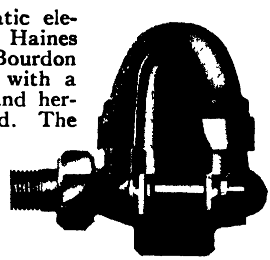


A ruggedly constructed bolted case trap. Ideal for hospital and kitchen equipment and all process work operating on pressure up to 60 pounds.

HAINES MODULATING VALVES

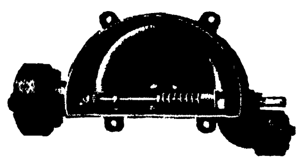
A packless valve assuring positive and leak proof performance. Completely opens or closes on less than a full turn of handle. Can be furnished with wheel or lever handle or lockshield.

HAINES RADIATOR TRAPS



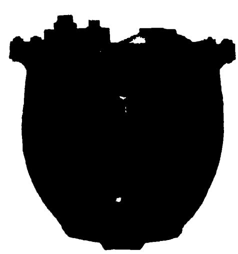
The thermostatic element in all Haines Traps is a Bourdon tube, charged with a volatile fluid and hermetically sealed. The expansion and contraction of the fluid, under varying temperatures, furnishes the operating power. The vertical seat of this trap prevents it from becoming inoperative from scale or other foreign matter.

HAINES HIGH PRESSURE TRAPS



For dripping high pressure mains, laundry equipment and all process fixtures with working pressures up to 125 pounds.

HAINES BOILER RETURN TRAPS



For vapor and atmospheric heating systems. Assures positive circulation by venting the air and returning the water of condensation to the boiler. Has no stuffing boxes or packed joints to leak air or water.

Each device is individually tested, factory adjusted and guaranteed.

Hoffman Specialty Co., Inc.

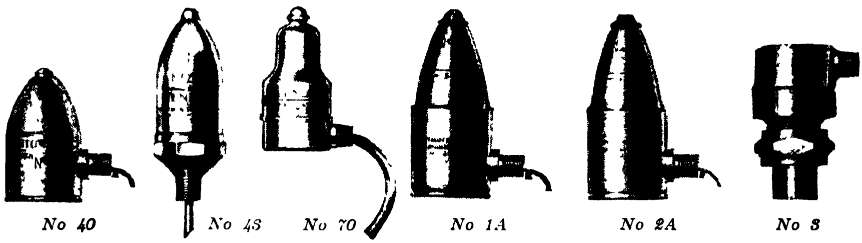
General Office and Factory

1001 York Street, **Indianapolis 7, Ind.**

Sales Representatives in Principal Cities

Manufacturers of Radiator Air Valves, Quick Vents and Air Eliminators for all types of Steam and Vacuum Heating Systems—Steam Traps of all kinds—Radiator Supply Valves—Vacuum and Condensation Pumps—and Hot Water Automatic Heat Control Systems.

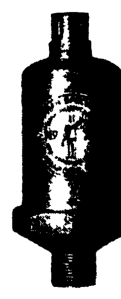
RADIATOR AIR VALVES FOR STEAM AND VACUUM SYSTEMS



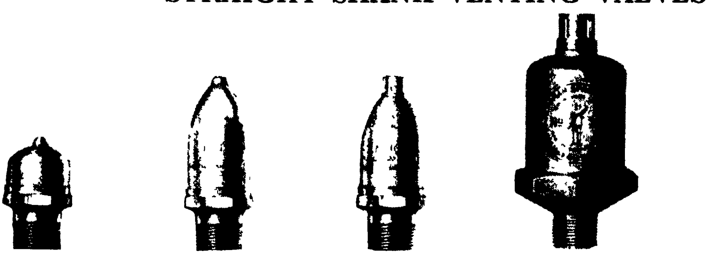
No 40 *No 43* *No 70* *No 1A* *No 2A* *No 3*

No. 40 **Steam**—Hoffman patented tongue syphon—1/8 in. connection—fixed port.
No. 43 **Steam**—Strait shank for convectors—telescopic syphon—1/4 in. connection.
No. 70 **Steam**—Meets Federal spec WW-V-151 Class 1—Long syphon—1/8 in. conn.
No. 1A **Steam**—Tongue syphon—ADJUSTABLE air opening—1/8 in. connection.
No. 2A VACUUM—Tongue syphon—ADJUSTABLE air opening—1/8 in. connection.
No. 3 **Steam**—For Airline or PAUL systems—1/8 x 1/4 in. conn.—union tailpiece.

All radiator air valves have polished brass finish and operate on 10 lb max. press.



STRAIGHT SHANK VENTING VALVES



No 4 *No. 4A* ***No. 16A*** ***No. 75*** ***No. 76***

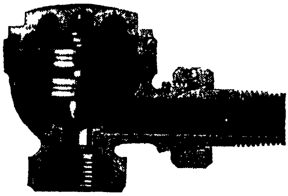
No. 4 **Steam Mains**—Will not close against water—25 lb press 3/4 in. connection.
No. 4A **Steam Mains**—Float closes against water—10 lb press. 3/4 in. connection.
No. 16A VACUUM **Mains**—Float closes against water—10 lb press. 3/4 in. connection.
No. 75 **Steam Mains**—Large systems—has float—10 lb press. 3/4 in. connection.
No. 76 VACUUM **Mains**—Large systems—has float—10 lb press. 3/4 in. connection.
No. 75A **Steam Mains**—Large systems at low pressure—has float—3 lb press. 3/4 in. conn.
No. 76A VACUUM **Mains**—Large systems low pressure—has float—3 lb press. 3/4 in. conn.

Nos. 75-75A-76-76A have cast iron bodies—others brass.

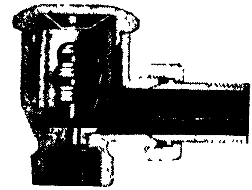
UNIT HEATER VENT VALVE

No. 74—**Operates 0 to 35 lb. Vents all air at any pressure and whether rising or falling—Same appearance as No. 75—Can be used on steam mains—has cast iron body—3/4 in. connection—finished in attractive Hoffman gray-green color.**

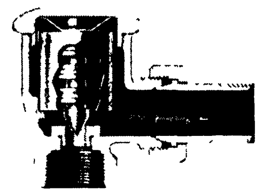
LOW, MEDIUM AND HIGH PRESSURE THERMOSTATIC TRAPS



Low Pressure



Medium Pressure



High Pressure

Low pressure traps have brass bodies, caps, and union nut and tail piece. 17C is made in Angle, Straightway, R. H., L. H and Vertical patterns. 8C is made in Angle and Straightway patterns. 9C is made in Angle pattern only. Thermostat and seat both renewable.

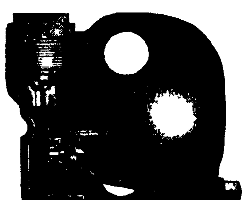
No. 17C Capacity 200 sq ft EDR 15 lb pressure ½ in. connection
No. 8C Capacity 400 sq ft EDR 25 lb pressure ¾ in. connection
No. 9C Capacity 700 sq ft EDR 25 lb pressure 1 in. connection

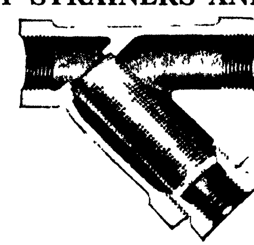
Medium Pressure Nos. 8 & 9 and High Pressure Nos. 8H & 9H have all bronze bodies and caps with union nut and tailpiece. Thermostats are 6 diaphragms of special non-corrosive metal. Thermostats and seats are renewable. ½ in. sizes are furnished in Angle, R.H., L.H., and Straightway patterns, others in Angle only. Medium Press. 50 lb limit. High Press. 125 lb.

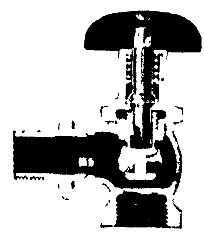
Capacities—Lb Condensate per Hour—Working Pressure—Lb per Sq. In. Gage

Traps		5	15	25	50	Traps		25	50	100	125
8	⅜"	100	180	235	400	8H	⅜"	235	400	550	590
8	½"	125	225	300	490	8H	½"	300	490	650	720
9	¾"	225	350	450	650	9H	¾"	450	650	875	950
9	1"	325	500	625	850	9H	1"	625	850	1125	1250

FLOAT TRAPS, DIRT STRAINERS AND SUPPLY VALVES





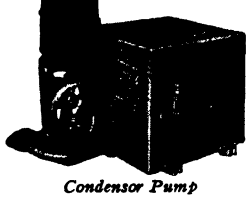


Float Traps are available in large capacities and four pressures, 15, 30, 60 and 125 lb. Used for venting and draining risers, steam mains, unit heaters, blast coils, etc. Hoffman Float Traps are made for easy servicing with all working parts mounted on cover. Remove four bolts to expose all parts. Pipe sizes are from ¾ in. to 2 in.

Radiator Supply Valves are made in sizes from ½ to 1½ in. in Angle and straightway patterns. Brass bodies, union nut and tailpiece. No. 80 has reversible cone disc and bevelled seat. No. 85 is Modulating type. Both are packless.

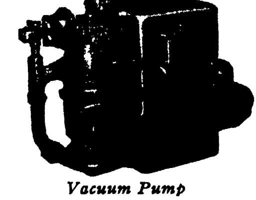
Hoffman Dirt Strainers are self-cleaning Y type. Brass strainer cylinders and cast iron body. Sizes ½ to 2 in. for 125 lb pressure. Should be used in line ahead of all float and thermostatic traps.

CONDENSATION AND VACUUM PUMPS



Condensor Pump

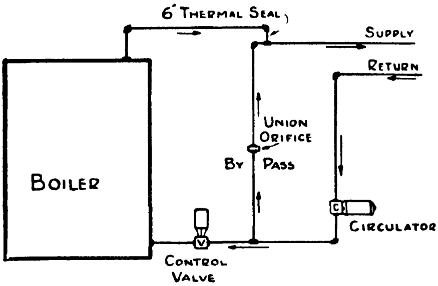
Hoffman-Economy pumps are available in varying capacities, D.C. and A.C. current, single, two, or three phase, and in pressures up to 150 lb. Also made in single and duplex units for different capacities and pressures.



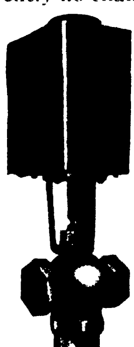
Vacuum Pump

HOFFMAN CONTROLLED HOT WATER HEATING

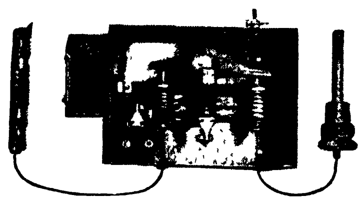
The Hoffman method vastly improves the ordinary forced hot water system by the application of Continuous Circulation. This method accomplishes 4 major improvements. 1. Avoids intermittent bursts of heat to the radiators. 2. Maintains a radiator temperature to exactly offset the heat loss thereby eliminating "cold 70." 3. Conserves fuel by positively preventing overheating. 4. Prevents creeping noises from pipes because temperature change is so gradual.



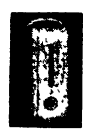
The diagram illustrates the general operation. The circulator (C) is running continuously, circulating the water through the by-pass from return back through the supply line. The water in boiler is kept at a predetermined temperature based on the Radiator Emission rate for which the system is designed. When heat is required the Temperature Controller or Thermostat (depending upon type of Hoffman system used) causes the control valve (V) to open, allowing the return water to pass through the boiler and the hotter boiler water mixes with water passing through the Hoffman orifice, in the by-pass line from the return, to raise the temperature of water entering the supply line. When the heat demand has been satisfied the control valve (V) closes and normal circulation through the by-pass pipe continues The mixing of the water, already somewhat cooled from continuous circulation, with the high temperature boiler water eliminates the "slugs" of extremely hot water so common with forced hot water systems. There is positively no overheating and the temperature is raised so gradually that a comfortable heat is always available with apparently no change in the heating system


Control Valve

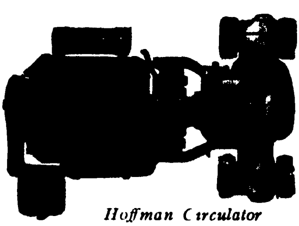
The two types of Hoffman Hot Water Controlled Heat differ only in the manner of control The Circulator, Control Valve and Union Orifice are furnished with both systems



1. Outdoor Thermal Control System—This System is controlled by means of the Hoffman Patented Temperature Controller illustrated above. Capillary tubing is connected to a temperature bulb located on the exterior of the building and another capillary tubing connected to a temperature bulb located in the supply line of the System Variations in temperatures of the outdoor weather and the circulating water are immediately transmitted through this Controller to the Control Valve This System very completely controls the supplying of the exact amount of heat necessary to offset the actual heat loss Made in nine regular pipe sizes from 1 in to 6 in. with a maximum capacity of 4,460,000 BTU



2. Indoor Thermostat Control System—Thermostat, located in that part of the building which is to determine the heat requirements for the entire System, operates the Control Valve through the Control Panel illustrated below. This allows the circulating water temperature to be increased according to the demand of the thermostat and will supply the heat required uniformly and gradually. This System is naturally less expensive than the Outdoor Control type Available in 1 in, $1\frac{1}{4}$ in and $1\frac{1}{2}$ in. sizes with a maximum capacity of 275,000 Btu.


Hoffman Circulator

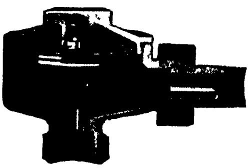
Design and Operating information is available upon request.

ILLINOIS ENGINEERING COMPANY

General Offices
and Factory:
Chicago 8, Ill.

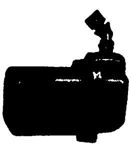
**Representatives
In Principal Cities**

Illinois Thermo Radiator Traps



Series G

Illinois Thermo Radiator Traps for vacuum, vapor and low pressure heating systems. Has cone type valve. Flushes throughly and seats perfectly at all times. Valve and seat are of Nitralloy. The duplex diaphragm is of special phosphor bronze. Scientific design and rugged construction assure flexibility and long life. These diaphragms have withstood over three million strokes on a breakdown test. *Ask for Bulletin*



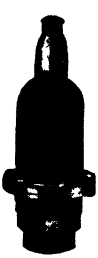
Vapair Vent Trap

Boiler Return Trap

Vapor Gauge

Damper Regulator

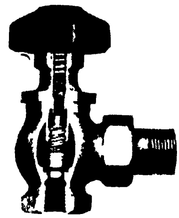
Illinois Selective Pressure Control Systems



Selective Controller

An entirely new and unique method of Steam Circulation Control . . . Heating Systems that set new standards in comfort, economy, simplicity and convenience of operation. Each system individually engineered to meet exact requirements. Recorded fuel savings, without sacrifice of comfort, warrant your investigation. *Ask for Bulletin 16.*

Illinois Modulating Supply Valve



Quick-opening, packless. Steam tight on 50 lb pressure. Large diameter of thread spool and machine cut threads make valve operation easy. Furnished in a complete line of sizes and patterns.

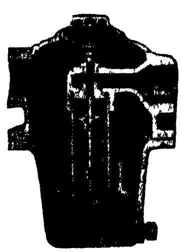
Illinois Vapor System

A two pipe low pressure steam circulating system which may be installed in any type of building, where the condensate can return to the boiler by gravity.

A sensitive damper regulator or other means of automatic control is used to control initial steam pressure above, at or below atmospheric pressure. Steam is regulated at the radiators by Illinois Modulating Supply Valves. Condensate and air are discharged from the radiator through Illinois Thermostatic Radiator Traps. In the boiler room a Vapair Vent Trap and Boiler Return Trap are installed near the boiler. The vent trap eliminates air from the system and the Return Trap insures return of condensate to the boiler.

The system and the piping arrangement are simple. No metering orifices or vacuum pumps are needed. This system will be found suitable for many installations where low first cost and low operating cost are of prime importance. May be used with unit heaters or any type of radiation.

Illinois Combination F & T Traps



Series G

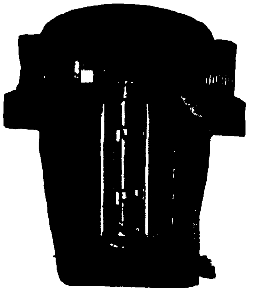
Unsurpassed for draining ventilating units, unit heaters, and for dripping mains and risers -- wherever it is desirable quickly to vent air from the main as well as handle the water of condensation in quantity, whether hot or cold.

ILLINOIS ENGINEERING COMPANY

General Offices
and Factory:
Chicago 8, Ill.

Representatives
In Principal Cities

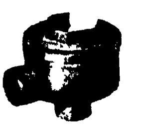
Illinois Steam Trap



Series 30

Valve and stem are separate from the bucket and operated only by the bucket at the extreme top and bottom of travel—result—***valve is always either full open or tight closed*** No wire drawing or cutting of valve and seat which are of stainless steel.

Illinois Thermostatic Traps for High Pressures

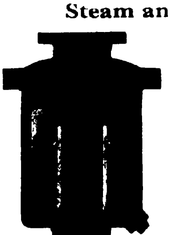


Series HG

Maximum working pressure 150 pounds Used where neat appearance and compactness are desirable, as for trapping sterilizers or water stills in hospitals; steam jacketed kettles, coffee urns, warming tables and for process work. Also used extensively for air vents on blast type drying heaters. Multi-diaphragm of phosphor bronze Heavy duty bronze body. Made in three sizes

These traps are also furnished for medium pressures *Write for literature.*

Steam and Oil Separators

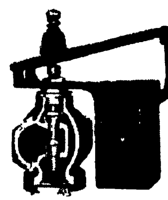


Vertical Standard Separators

Eclipse steam separators are made in both horizontal and vertical type, and also the special receiver separators for standard or extra heavy pressures.

Eclipse oil separators are furnished in the horizontal type and have a removable baffle plate to facilitate cleaning of baffle and keeping the separator's efficiency at the highest point.

Illinois Motorized Valves (on and off)



Type E

For automatic control of steam temperatures and pressures to prevent overheating and conserve steam; to control fluid levels; and to regulate flow in hot water heating systems. May be operated by any automatic contact device or by manual switches.

Furnished in three types

Spring Controlled Regulating Valve

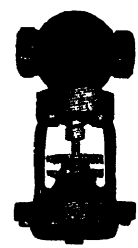


Fig 121

Furnished in either single seat or double seat type as service requires, for the control of steam, air or gas Control spring is completely enclosed, protecting it from dirt and rust Valves are furnished with proper size diaphragm and proper length spring to give satisfactory service under all operating conditions Furnished also in weight loaded type, Fig. 71.

Write for Bulletins.

Master Type Pressure Regulator

Fig 142

Used wherever high pressure steam must be accurately reduced in varying amount to any steady lower pressure, in service such as hospital, laundry, cooking, process, dry kilns and railway steam control. It will reduce initial pressure up to 250 pounds down to any lower pressure. Does not build up pressure on a closed or dead end line. Made of bronze with monel metal valves and seats.

Jas. P. Marsh Corporation

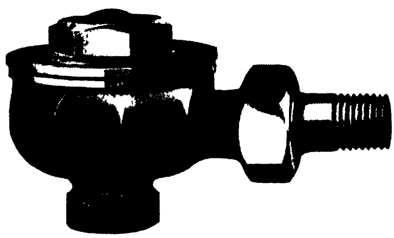
2072 Southport Ave., **Chicago 14, Ill.**

Branches in Principal Cities

Marsh products include: Pressure, Vacuum and Compound Gauges; Dial Thermometers; Steam Traps; Air Valves and Vents; Packless Radiator Valves and other heating specialties.

Thermostatic Diaphragm Radiator Traps

These efficient traps are equipped with a phosphor bronze diaphragm, consisting of two wafers of tinned phosphor bronze, drawn and spun to perfection of temper.

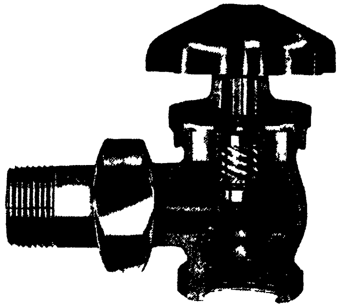


The wafers are spun together and soldered to form a seamless, sensitive, powerful expanding member—not easily fouled by dirt and foreign matter. Diaphragms are charged with a volatile fluid making them self-equalizing for use on pressures below atmosphere to 15 lbs. gauge. Traps are factory adjusted.

(Due to war restrictions some changes are made in the construction of these traps. However, the diaphragm or motive element remains as illustrated.)

Packless Radiator Valves

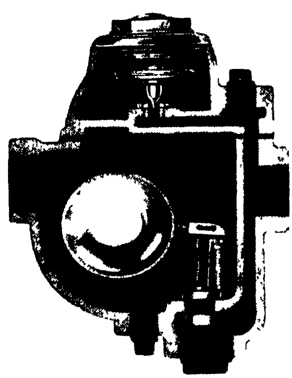
Marsh all-metal packless valves are truly packless. These valves contain no packing to deteriorate, wear or crack, and are simple in design, with ample strength



where strength is required. The principles upon which they are designed have been proved sound over many years of service. Valves operate easily—opening or closing tightly with less than one full turn. All valves are individually tested. Adaptable for use on hot water—forced or gravity systems—as well as all steam heating systems.

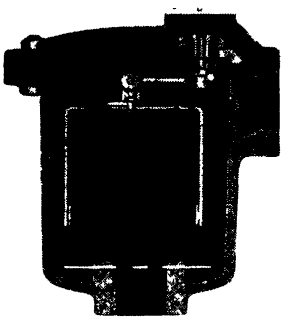
(Due to war restrictions some changes are made in the body construction and available sizes. However, the interior construction remains as illustrated.)

Marsh No. 12
Float and Thermostatic Traps



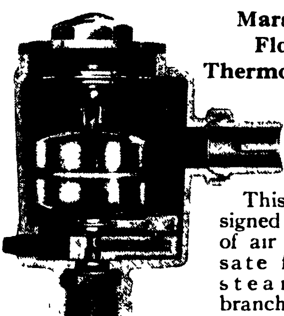
Marsh Heavy Duty Float and Thermostatic Traps are designed for removal of air and condensate from steam mains, branches, or risers, unit heaters, steam coils, etc. The size and weight of the trap permits installation in the piping without any other means of support. Condensation is discharged through a float operated valve located at the lowest point inside the trap body. Air vent is located in a by-pass in the cap or cover of the trap. Air passes through a passageway and out through the trap outlet. Construction permits removal of mechanism without disturbing the piping.

Marsh No. 500
Inverted Bucket Type Trap



This type is ideal for all types of hospital and kitchen equipment or similar service where a considerable volume of condensate is handled. The trap is self-venting, which, combined with the large water capacity assures unusually high efficiency in removing condensate, air and gases.

Marsh No. 17
Float and
Thermostatic Trap

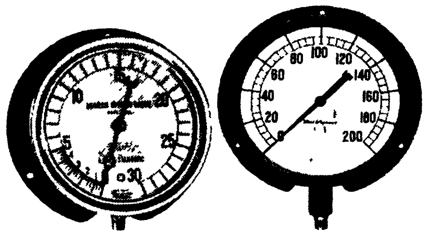


This trap is designed for removal of air and condensate from short steam mains, branches or risers Operating characteristics adapt it for unit ventilators, unit heaters, and other equipment subjected to freezing temperatures when heating system is not in operation Outlet discharge is water sealed at all times Air vent is located in trap bonnet and air is normally discharged through a port directly to the outlet connection. A removable strainer protects mechanism All working parts are made accessible by removing bonnet. The piping is all the support required for the No. 17 Trap.

Marsh Low Pressure Gauge

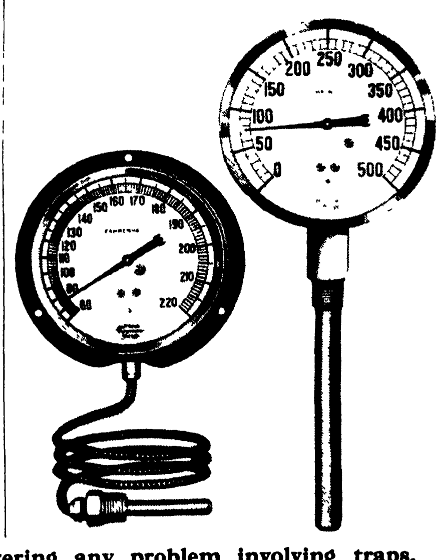
The Marsh A.S.M.E. standard, low pressure gauge will contribute to the economy and improve the operation of any type of steam boiler It is finely built throughout and is available with the Marsh "Recalibrator" for quickly and easily resetting the hand to zero when the gauge is knocked out of adjustment.

Marsh Gauges include vacuum and compound types in a wide range of designs covering all services and pressures. Over 75 years of gauge manufacturing has reached its highest achievement in the Marsh "Mastergauge" for use where high pressures and temperatures are present and where maximum stamina and accuracy are essential.



Marsh Dial Thermometers

Have the same basic refinements found in Marsh Gauges. Typical Marsh Thermometers of bourdon tube type in self-contained and distant reading types are illustrated. They are available in either vapor tension or gas-filled types. Bi-metallic types of dial thermometers are also available Practically all temperature ranges up to 750° F. are covered. The "Recalibrator" is standard in all Marsh bourdon tube type thermometers. The Marsh line also includes recording thermometers



Ask for complete information covering any problem involving traps, vents, gauges, dial thermometers, radiator air valves, packless valves, etc.

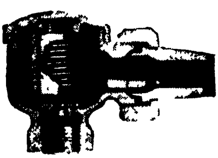
Sarco Company, Inc.

475 Fifth Ave., **New York 17, N. Y.**

Branches in Principal Cities

SARCO CANADA LIMITED, 85 RICHMOND ST., W., TORONTO, ONT.

PRODUCTS—A complete line of Specialities for Vapor, Vacuum and Gravity Steam Heating Systems and Control combined with a competent Engineering Service to architects and heating engineers to assist them in providing modern heating.



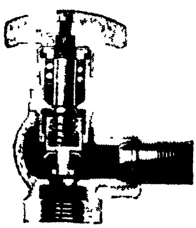
Type HS Radiator Trap

SARCO RADIATOR TRAPS

In accordance with latest WPB regulations, Sarco Radiator Traps for vapor and vacuum heating systems are now furnished in all brass construction, rough brass finish, with renewable bronze seat and self-aligning bronze valve head Union connection is provided on inlet.

The interior construction and, therefore, the efficiency and long life of the trap, remain unchanged. They are still equipped with the Sarco helically corrugated bellows, made from heavy wall bronze tubing drawn in our own plant The hard bronze seat is renewable.

Available in ½ in. and ¾ in. sizes, angle, straightway or corner offset patterns; also 1 in., angle style only *Catalog No. HV-162.*



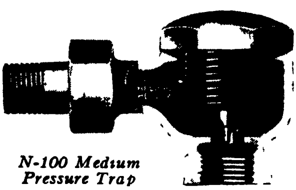
Type R Radiator Valve

SARCO RADIATOR VALVE

While wartime restrictions remain in force, we furnish the semi-packless valve, type R, illustrated.

It is of the quick-opening, non-rising stem type, with permanent, molded packing, spring controlled.

Construction is all brass, with union spud and nut on outlet, Jenkins disc for steam, metal-to-metal for hot water. Sizes ¾ in. and 1 in., with wheel handle or lock shield *Catalog No. HV-162.*

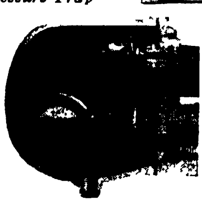


N-100 Medium Pressure Trap

SARCO N-100 TRAP

For high pressure radiators and heating coils in stationary and marine service, and for hospital and kitchen equipment. Has full length protecting shield and stainless steel valve head and seat. Sizes ⅜ in to 1 in pressures to 100 lb *Catalog HV-46.*

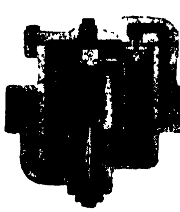
Also S-65 for pressures to 65 lbs.



Float-Thermostatic Trap

SARCO FLOAT-THERMOSTATIC TRAPS

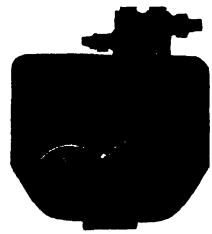
For dripping ends of mains and risers, and for stack or blast heaters, large unit heaters and hot water generators. Automatic thermostatic air vents built in. Available in six sizes with connections ¾ in. to 2 in. Pressures up to 200 lb. *Catalog HV-450*



Inverted Bucket Trap

SARCO INVERTED BUCKET TRAPS

Are recommended for high pressure unit heaters and sometimes preferred for kitchen and laundry equipment Strainers are built right into these sturdy traps. Seats and valves are stainless steel and renewable. Automatic air vents can be furnished for extra rapid removal of air. Available in sizes ½ in. to 2 in. for pressures up to 900 lb. *Catalog HV-350.*



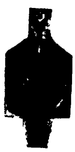
Alternating Receiver

SARCO ALTERNATING RECEIVER

A complete line of boiler return traps for vapor systems.

Returns water of condensation to boiler automatically, thereby assuring positive return of water under all pressure conditions.

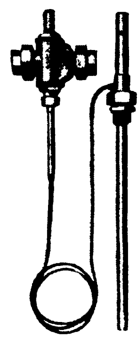
Made in four sizes for up to 14,000 sq ft of radiation *Catalog HV-165.*



SARCO AIR ELIMINATORS

For venting air from vapor systems at one central point in the basement. Available in three sizes, No. 5 for systems up to 2000 sq ft radiation, No. 6 for 3000 sq ft, and No. 12A for 15,000 sq ft. All are equipped with float valves to stop water escaping thru the vent and with check valves to prevent ingress of air when system is under vacuum.

Also several types for hot water heating systems *Catalog HV-170.*



Type TR-21 Standard for hot water storage tanks, fan units, etc

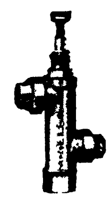
Water Blender Type MB

SARCO SELF-CONTAINED TEMPERATURE REGULATORS

Sarco Temperature Regulators are simple, self-operated valves—the only self-contained units that use the irresistible force of liquid expansion. No stuffing boxes to leak, no auxiliary "power" required; all moving parts are *inside* the equipment. Here again—a type and size for every purpose—for steam, gas, oil, water or brine for temperatures ranging from 0 to 400° F *Catalog HV-600*

SARCO WATER BLENDERS AND TEMPERING VALVES

For mixing hot and cold water to deliver automatically water at any desired temperature. Two models are available, type MB for showers, wash basins, etc, and type DB, a tempering valve for use with submerged heating coils or tankless heaters. *Catalog HV-800*

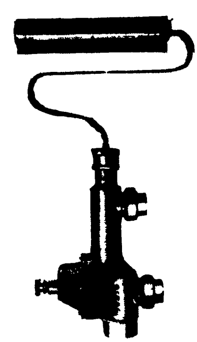


Water Blender Type DB

SARCOTHERM HOT WATER HEATING SYSTEM

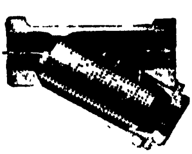
A simple, all-mechanical system for the control of radiator temperatures in direct relation to outside temperatures Radiation is balanced by Sarcoflow fittings in the radiator outlets

The Sarcotherm three-way valve recirculates a varying proportion of water around the boiler and back to the system as dictated by the thermostatic bulb outside the building *Catalog "Sarcotherm" No. 1.*



Sarcotherm Weather Control Valve

SELF-CLEANING STRAINERS



For use in pipe lines carrying brine, steam, oil, gas, water, ammonia or air. Have large free screening area with minimum resistance to flow. Steam or air strainers can be cleaned by blowing through without disassembling. Made in cast iron, bronze or cast steel for pressures up to 500 lb, with brass, iron or monel screens. Available in sizes ¼ to 8 in. *Catalog No. HV-1200.*

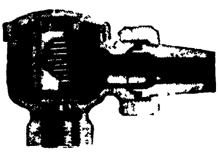
Sarco Company, Inc.

475 Fifth Ave., **New York 17, N. Y.**

Branches in Principal Cities

SARCO CANADA LIMITED, 85 RICHMOND ST., W., TORONTO, ONT.

PRODUCTS—A complete line of Specialities for Vapor, Vacuum and Gravity Steam Heating Systems and Control combined with a competent Engineering Service to architects and heating engineers to assist them in providing modern heating.



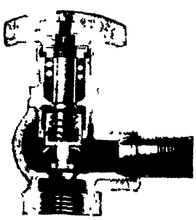
Type HS Radiator Trap

SARCO RADIATOR TRAPS

In accordance with latest WPB regulations, Sarco Radiator Traps for vapor and vacuum heating systems are now furnished in all brass construction, rough brass finish, with renewable bronze seat and self-aligning bronze valve head Union connection is provided on inlet.

The interior construction and, therefore, the efficiency and long life of the trap, remain unchanged. They are still equipped with the Sarco helically corrugated bellows, made from heavy wall bronze tubing drawn in our own plant The hard bronze seat is renewable.

Available in ½ in. and ¾ in. sizes, angle, straightway or corner offset patterns; also 1 in., angle style only *Catalog No. HV-162.*



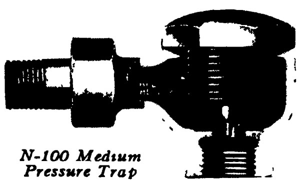
Type R Radiator Valve

SARCO RADIATOR VALVE

While wartime restrictions remain in force, we furnish the semi-packless valve, type R, illustrated.

It is of the quick-opening, non-rising stem type, with permanent, molded packing, spring controlled.

Construction is all brass, with union spud and nut on outlet. Jenkins disc for steam, metal-to-metal for hot water. Sizes ¾ in. and 1 in., with wheel handle or lock shield. *Catalog No. HV-162.*



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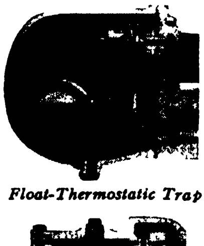
Also S-65 for pressures to 65 lbs



Float-Thermostatic Trap

SARCO FLOAT-THERMOSTATIC TRAPS

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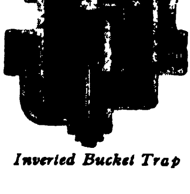

Inverted Bucket Trap

SARCO INVERTED BUCKET TRAPS

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Alternating Receiver

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Type TR-21 Standard for hot water storage tanks, fan units, etc

Water Blender Type MB

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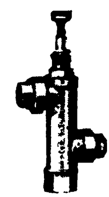
SARCO WATER BLENDERS AND TEMPERING VALVES

For mixing hot and cold water to deliver automatically water at any desired temperature. Two models are available, type MB for showers, wash basins, etc., and type DB, a tempering valve for use with submerged heating coils or tankless heaters. *Catalog HV-800*

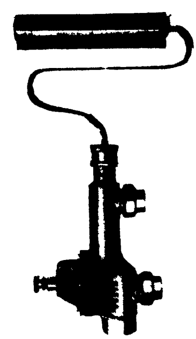


Water Blender Type DB

SARCOTHERM HOT WATER HEATING SYSTEM

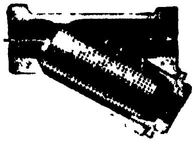
A simple, all-mechanical system for the control of radiator temperatures in direct relation to outside temperatures Radiation is balanced by Sarcoflow fittings in the radiator outlets

The Sarcotherm three-way valve recirculates a varying proportion of water around the boiler and back to the system as dictated by the thermostatic bulb outside the building *Catalog "Sarcotherm" No. 1*



Sarcotherm Weather Control Valve

SELF-CLEANING STRAINERS



For use in pipe lines carrying brine, steam, oil, gas, water, ammonia or air. Have large free screening area with minimum resistance to flow. Steam or air strainers can be cleaned by blowing through without disassembling. Made in cast iron, bronze or cast steel for pressures up to 500 lb, with brass, iron or monel screens Available in sizes ¼ to 8 in. *Catalog No HV-1200.*

WARREN WEBSTER & COMPANY

Pioneers of the Vacuum System of Steam Heating

Webster -since 1888 Systems of Steam Heating

Main Office and Factory:
Camden, New Jersey
Representatives in Principal Cities—
Consult Your Local Phone Directory

Webster Nesbitt UNIT HEATERS

NOTICE—The availability of Webster Equipment described on these pages is subject to restrictions resulting from war and priority regulations and conditions. We reserve the right to change prices, materials, and designs without notice. The low pressure steam heating specialties listed comply with WPB Limitation Order L-42, Schedule VIII.

PRODUCTS AND SERVICES

Webster Systems of Steam Heating including Vacuum and Type "R" (vapor).

Webster Central Control Systems including HYLO and MODERATOR.

Modernization of Obsolete and Faulty Heating Systems.

Webster System Equipment including Light-Weight Concealed Radiation (Gravity Convection Heaters), Radiator Supply Valves, Metering Orifices, Thermostatic Traps, Drip Traps, Heavy Duty Traps, Dirt Strainers, Dirt Pockets, Boiler Return Traps, Vent Traps, Damper Regulators, Boiler Protectors, Lift Fittings, Expansion Joints, Separating Tanks, Steam and Oil Separators, Steam Vacuum Pump Governors, Air Separating and Receiving Tanks, Gages, Water Accumulators.

Webster Series "78" and Series "79" Traps for use at process pressures (10 to 150 lb per sq in.)

Webster-Nesbitt Unit Heaters.

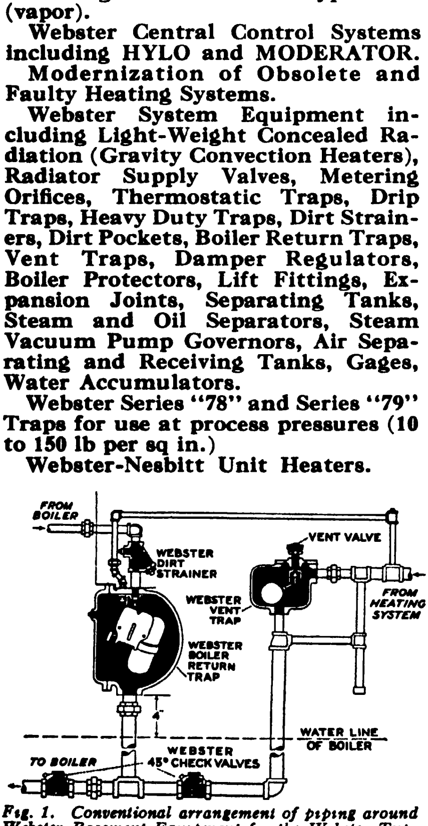


Fig. 1. Conventional arrangement of piping around Webster Basement Equipment for the Webster Type "R" System

WEBSTER SYSTEMS

Webster Systems are low pressure, two-pipe systems of steam circulation *with the addition* of accurately-sized metering orifices at radiator supply connections and, when required, intermediate metering orifices at points in branch mains Metering orifices effect even distribution of steam to all parts of the heating system and permit the successful application of a centralized control. Webster Valves are used at supply of radiators Webster Thermostatic Traps prevent flow of steam into return mains when radiators are filled. Webster Drip Traps and Dirt Strainers are used where needed on steam mains Webster Systems are available for vacuum, open return or "vapor" operation The Type "R" System corresponds to the so-called Vapor type. Fig. 1 illustrates a typical arrangement of Boiler Return Trap, Vent Trap, etc., when low pressure boiler is the source of steam.

WEBSTER CENTRAL CONTROLS

These are patented systems for varying the amount of steam to all radiators according to outside temperature. They provide *continuous heat delivery* with effective *fractional filling of radiators.* The Hylo Systems may be provided for manual control, or if desired, may be semi-automatic by incorporation of inside thermostat or thermostat and schedule clock. The Moderator Systems employ an automatic Outdoor Thermostat supplemented by a manual Variator.

The latter is used for quick heating-up, night load, and unusual weather or occupancy conditions. Use of Webster Central Control Systems results in (1) increased comfort because over-heating and underheating are minimized and (2) lower fuel or steam costs.

WEBSTER SYSTEM RADIATION

Discontinued for the Duration

Concealed, non-ferrous type for use exclusively with Improved Webster Systems. Is unique in that it combines *in a single unit,* a light-weight heating element of high efficiency with an orificed radiator supply valve, a radiator trap and supply and return piping connections. Metal

enclosures for installation within the wall and exposed metal cabinets are available. Webster System Radiation and enclosures are so designed that the entire heating element can be quickly removed without damage to plaster or paint. Space requirements reduced to a minimum and installation greatly simplified.

RADIATOR VALVES

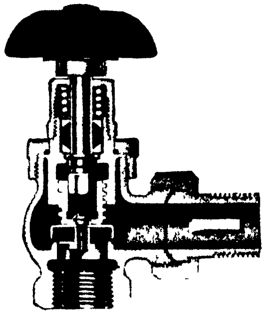


Fig 2 Webster Type "WB-P" Brass Radiator Valve

The Type WB-P Valve meets fully specifications calling for a "spring packless" valve A heavy spring automatically maintains pressure on die-molded metallic ring packing. Although packing seldom requires renewing, this valve is so designed that old packing ring can be removed and new installed while pressure is on the heating system

The Type WB-P Valve opens quickly and easily in less than a turn of the handle Has non-rising stem All brass construction including union nut, nipple, and interior parts Furnished with standard wheel handle as shown in Fig 2 Lockshield type available for institutions

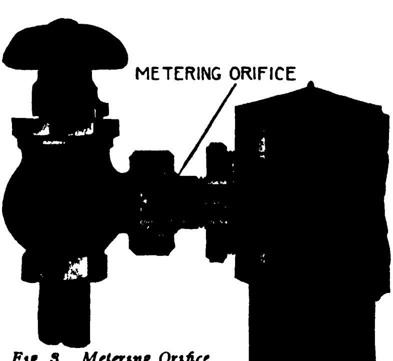


Fig 3 Metering Orifice Inserted in Union Connection of a Webster Supply Valve.

Angle bodies in sizes from ½ in. to 1½ in. Offset and straightway models can be had on order.

While primarily for low pressure steam heating service, the Type WB-P Valves are entirely suitable for hot water heating. Furnished with or without leak hole as desired.

Pressures—For low pressure vapor and vacuum steam heating service. Maximum pressure is 75 lb per sq in.

Metering Orifices—Accurately sized and made of metal to resist erosion and corrosion, amply thick to be free from vibration and shaped for quiet operation.

RADIATOR TRAPS

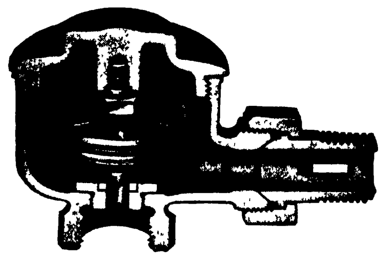


Fig 4 Webster 702H Thermostatic Radiator Trap

The Webster Series 7 Trap is a diaphragm-type thermostatic trap designed for low pressure service. It is recommended for all types of radiators, piping drip points, steam-using equipment, and other applications within its pressure and capacity range. The Series 7 Trap is noted for its high efficiency and for giving many years of trouble-free service.

Construction Features—Body and cap are high quality brass or gray cast iron. Double thermostatic diaphragm is phosphor bronze, individually factory adjusted and tested. Diaphragms are compensated for pressure which means that they function efficiently at *all* pressures within their operating range. They do not close too quickly at certain pressures to hold up condensate while remaining open at other pressures to pass steam.

Stainless steel or brass valve piece is 60 deg cone type, factory adjusted. Brass seat with stainless steel insert is readily renewable.

Pressures—Webster Series 7 Traps are designed for low pressure vapor and vacuum steam heating service. Maximum pressure is 25 lb per sq in.

Table 1. Recommended Ratings in Sq Ft E. D. R.*

The ratings below are conservative and not full-flooded capacities. Applications requiring use of higher ratings should be referred to the Company or its Representatives. When writing give full details of proposed use. Select trap by rating, not by pipe size.

Symbol	Size	Pressure Difference Across Trap in Lb per Sq In.					
		1	1½	2	5	10	15
702H	½"	165	200	235	370	530	640
712H	½"	330	400	465	730	1050	1300
723H	¾"	580	700	810	1280	1840	2300

*Based on 240 Btu per sq ft per hour

FLOAT-AND-THERMOSTATIC TRAPS

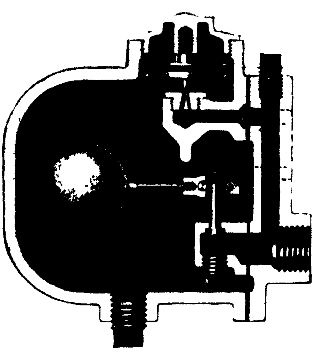


Fig 5 Webster Size 00026-T Drip Trap has rated capacity of 200 lb Water per Hour at 2 lb Pressure Difference

Series "26"—A heavy duty trap for drips of mains, blast radiation, unit heaters, hot water generators and similar applications. A rugged float-type trap available with and without thermostatic air vent. Made in five sizes: 200, 500, 1200, 2400 and 5000 lb water per hour at 2 lb pressure difference. Maximum working pressure is 15 lb per sq in.

PROCESS STEAM TRAPS

Series "78"—thermostatic trap *built for* process steam pressures (10 to 150 lb per sq in.). Monel Metal diaphragm. Stainless Steel or brass valve piece and seat insert. Angle model only. Sizes: ⅜, ½, ¾ and 1 in. Extensively used with laundry, cooking, sterilizing and other process-steam uses.

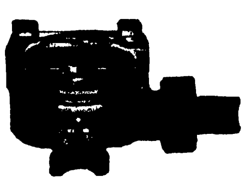


Fig 6 Webster Size 782 Trap

Series "79"—For use where large volumes of very hot condensate form more quickly than can be discharged by thermostatic traps alone. Float and thermostatic traps designed for normal working pressures between 15 and 150 lb per sq in. Water of condensation is passed through a float-controlled seat opening while air is discharged into the return piping by a thermostatically controlled vent Compact and light in weight. Can be readily mounted in a pipe line without other support Available with 1 in. inlet and outlet. Readily bushed for smaller pipe sizes.

Cast iron body, composition gasket and cover bolted together with steel cap screws Monel Metal or brass valve piece and stem. Stainless steel seat. Air vent unit is Monel Metal diaphragm with Stainless Steel or brass valve piece and brass seat with Stainless Steel insert.

DIRT STRAINERS AND POCKETS

Placed in return lines of steam heating systems to prevent dirt, rust and scale from impairing tightness of traps

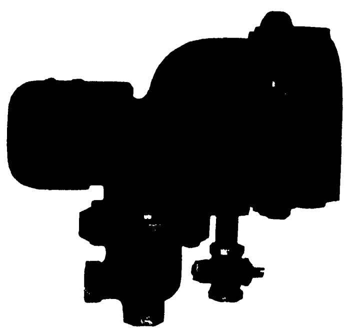


Fig 7 Size 34C-1 Webster Boiler Protector with Low Water Electrical Cut-out Switch Size 34 has no Cut-out Switch

BOILER PROTECTOR

Prevents breakage in low pressure heating boilers when water level becomes inadequate. Automatically supplies raw water to boiler when water level drops to 1 in. above bottom of gage glass

For maximum boiler pressure of 15 lb per sq in. Maximum cold water main pressure should not exceed 150 lb per sq in.: minimum must not be less than 25 lb per sq in.

Made with ¾ in. connections, with or without electrical cut-out switch

WEBSTER-NESBITT UNIT HEATERS

Are manufactured by John J. Nesbitt, Inc., Holmesburg, Philadelphia 36, Pa., and are distributed solely through Warren Webster & Company, Camden, New Jersey. Designed to circulate large volumes of air at comparatively low temperatures, assuring quick heating.

Ratings of Webster-Nesbitt Unit Heaters are based on tests made in accordance with standard test code of Industrial Unit Heater Association and A.S.H.V.E.

PROPELLER FAN UNIT HEATERS

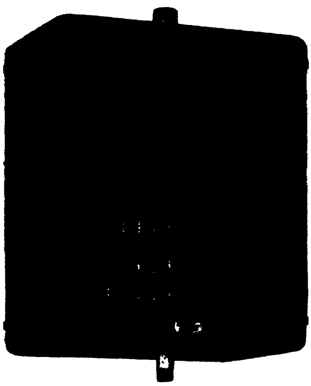


Fig 11 Standard Propeller-Fan Type

Designed to incorporate four characteristics proved by wide experience to be essential to both proper application and satisfactory performance:

1.) *Selective range of sizes.* Manufactured in nine sizes Heating capacities vary from 34,700 to 338,000 Btu per hour. Air deliveries from 470 to 4800 cfm.

2.) *Quiet Operation.* All fans have blades of exceptionally large areas and of a shape to impart a gradual acceleration to the air stream. Ample spacing is maintained between the fan and heating element. Motors are of sleeve bearing type and equipped with isolators.

3.) *Durable lightweight Heating Elements.* Extended fin-and-tube type, constructed of copper condensing tubes and plate-type aluminum fins.

4.) *Modern Casing Design.* Compact suspended type. Catalog W-N 123.

GIANT UNIT HEATERS

Sturdy blower-fan units for the economical heating of large areas.

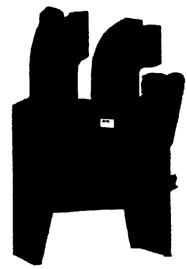


Fig 12 Blower-Fan Type

Standard (Non-Thermadjust) Type. Used principally where heating is by recirculation only, and where constant heat output is desired during period of operation.

Thermadjust Type. Employs dampers in front of casing and over face of heating element to provide mixing of unheated and heated air, producing heat output in accordance with requirements and continuous circulation of air volume.

Valve Controlled Type. Unit is of standard casing arrangement but equipped with Nesbitt Heating Surface with Steam-distributing Tubes which allows for automatic control of heat output.

Floor mounted, wall mounted, ceiling suspended, from 272,000 Btu, 4800 cfm, to 930,000 Btu, 15,000 cfm. Write for details Catalog W-N 116.

LITTLE GIANT UNIT HEATERS

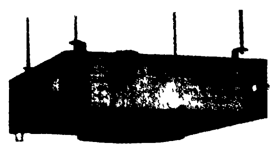


Fig 13 "Little Giant" Down Blow Type

New, light compact draw-through high velocity down blow unit heaters. 181,000 Btu, 3440 cfm to 505,000 Btu, 10,080 cfm at basic rating of 2 lb steam and 60 deg entering air.

Down Blow Type. Generally used when the presence of cranes and other machinery requires that the unit and piping be located well above the floor level. Write for details. Catalog W-N 118.

Armstrong Machine Works

851 Maple Street **Three Rivers, Mich.**

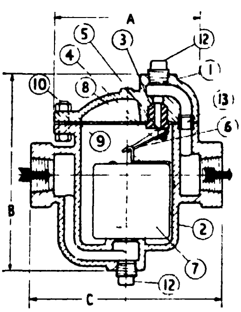
Representatives in All Principal Cities

Armstrong offers two types of traps for heating, air conditioning, and steam distribution service.

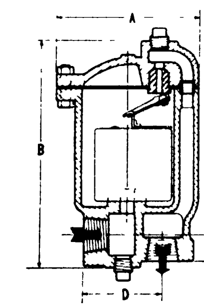
Standard Inverted Bucket Traps, the type originated by Armstrong, are *non-airbinding* and *self-scrubbing*. They are used for low, medium, and high pressure service where relatively little air must be handled along with the condensate. Their free-floating lever design makes it possible to open very large discharge orifices compared with the size of the trap itself.

Armstrong Blast Traps are used where large amounts of air must be vented quickly when steam is first turned on. They have several advantages over the conventional float and thermostatic trap.

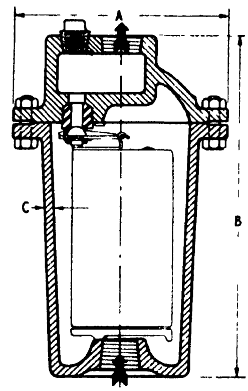
1. The Armstrong Blast Trap has but a *single orifice* to be maintained tight against the full pressure differential.
2. *Positive action.* The discharge valve in an Armstrong Blast Trap is either wide open or tight shut. Fast opening and fast closing prevent wire-drawing.
3. *Handles dirt.* There are no dead spots in an Armstrong Trap in which dirt can settle and interfere with the operation of the trap.



Cross-section of No 800, 811, 812 and 813 traps for straight-through pipe connections



Cross-section of No. 801 trap for standard angle pipe connections



No. 211-216, Bottom inlet Type

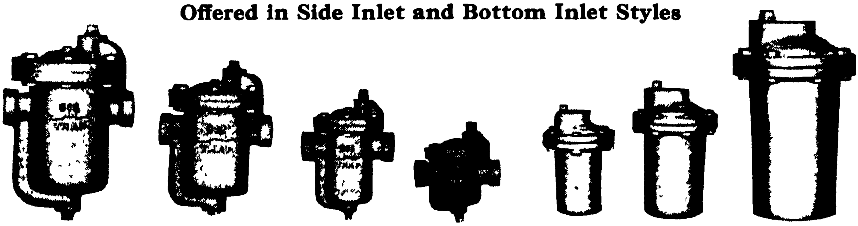
Side Inlet Traps

Trap Size		No. 800	No. 811	No. 812	No. 813	No. 801
Pipe Connections		1/2" or 3/4"	1/2" or 3/4"	1/2" or 3/4"	3/4" or 1"	1/2" or 3/4"
List Price (Regular)		$7.00	$10 00	$16.00	$22 00	$7 00
List Price (Blast Trap)		$8.50	$11.50	$18.00	$24.00	$8.50
Telegraph Code (Regular)		Aloe	Brown	Cherry	Dawn	Arrow
Telegraph Code (Blast Trap)		Aloette	Brownette	Cherette	Dawnette	Arrowette
Dimension A		3 3/4"	3 3/4"	5 5/8"	7"	3 3/4"
" B		5 1/8"	6 9/16"	8 11/16"	11 1/4"	6"
" C		5"	5"	6 1/2"	7 3/4"	
" D						2 1/16"
" E						11/16"
Number of Bolts		6	6	6	6	6
Diameter of Bolts		1/4"	1/4"	3/8"	1/2"	1/4"
Weight		4 1/2 lbs.	5 1/2 lbs.	13 1/2 lbs	25 lbs.	4 1/2 lbs
Maximum Pressure, lbs.		125	250	250	250	125
Continuous discharge capacity in lb of water per hour at pressure indicated. For more complete information see the Capacity Chart in Armstrong Steam Trap Book.	Lb Pressure 5	450	830	1600	2900	450
	10	560	950	1900	3500	560
	15	640	1060	2100	3900	640
	20	690	880	1800	3500	690
	30	500	1000	2050	4000	500
	50	580	840	1900	4100	580
	70	660	950	2200	3800	660
	100	640	860	1800	3600	640
	125	680	950	2000	3900	680
	150		810	1500	3500	
	200		720	1200	3200	
	250		760	1300	3500	

UNIT HUMIDIFIERS Steam Type

Armstrong Steam Type Unit Humidifiers offer a low-cost, practical method of preventing the evils of winter heating dry air Improve worker comfort and safety, prevent drying out of materials Write for new Bulletin No 158.

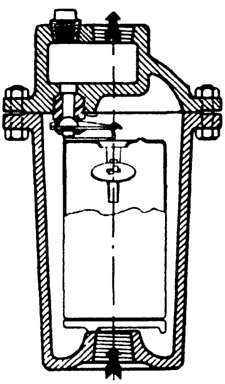
Offered in Side Inlet and Bottom Inlet Styles



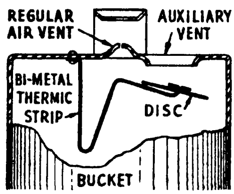
No 813 *No. 812* *No. 811* *No. 800* *No 211* *No. 212* *No. 213*

4. The wearing parts in all Armstrong Traps are identical in design, material, and precision workmanship with parts used in Armstrong Forged Steel Traps for pressures up to 1500 lb gage and total temperatures of 850 F.

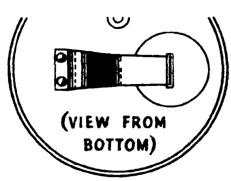
Armstrong Steam Trap Book. This 36 page book gives complete information on all sizes and types of Armstrong Traps. It also contains 17 pages of data on the subject of trap selection, installation, and maintenance. A free copy will be mailed on request.



No. 211-216, Blast Type



FOR BLAST TRAP JOBS



ALL Armstrong traps are readily convertible into "Blast" type traps merely by using buckets equipped with the patented auxiliary thermic air vent. As shown in the above sketches, the mechanism for this vent consists of a stainless steel disc slotted to receive the end of a bi-metal strip. Different coefficients of expansion in the bi-metal cause it to bend down when cold and up when hot. Normally, it is set to close at 212 deg, but it can be set to close at higher temperatures. Capacity, 50 to 100 times the air-venting capacity of a standard trap.

Bottom Inlet Traps

Trap Size		No 211	No. 212	No. 213	No. 214	No. 215	No. 216
Pipe Connections		1/2"	1/2" or 3/4"	1/2" or 3/4"	1"	1" or 1 1/4"	1 1/2" or 2"
List Price (Regular)		$ 9.25	$15.00	$20 75	$29.00	$38.00	$55.00
List Price (Blast Trap)		$10 75	$17.00	$22 75	$31.50	$40.50	$60.00
Telegraph Code (Regular)		Aspen	Birch	Walnut	Hemlock	Larch	Tamarack
Telegraph Code (Blast Trap)		Aspette	Birette	Walette	Hemlette	Larette	Tamrette
Height . Dimension B		6 3/8"	8"	10 1/4"	12 1/2"	14"	16 3/4"
Diameter " A		4 1/8"	5"	6 3/8"	7 1/2"	8 1/2"	10 9/16"
Wall Thickness " C		3/16"	1/4"	5/16"	3/8"	3/8"	7/16"
Diameter of Bolts		1/4"	1/4"	3/8"	3/8"	1/2"	1/2"
Number of Bolts ..		6	8	6	8	8	12
Weight		5 1/2 lb	10 1/2 lb	19 lb	32 lb	47 lb	80 lb
Maximum Pressure, lb		250	250	250	250	250	250
Continuous discharge capacity in lb of water per hour at pressure indicated. For more complete information, see the Capacity Chart in the Armstrong Steam Trap Book. — Lb Pressure	5	830	1600	2900	4800	7600	14500
	10	950	1900	3500	5800	9000	17300
	15	1060	2100	3900	6500	10000	19200
	20	880	1800	3500	6000	8500	18500
	30	1000	2050	4000	6800	9800	18000
	50	840	1900	4100	6300	9000	18200
	70	950	2200	3800	6000	9200	18300
	100	860	1800	3600	6200	10400	18000
	125	950	2000	3900	6700	10900	20000
	150	810	1500	3500	5700	9500	18500
	200	860	1600	3200	5300	9200	17580
	250	760	1300	3500	5700	7000	19000

ADSCO
PRODUCTS
for STEAM
SERVICE

AMERICAN DISTRICT STEAM COMPANY

NORTH TONAWANDA, N.Y

IN BUSINESS OVER SIXTY YEARS

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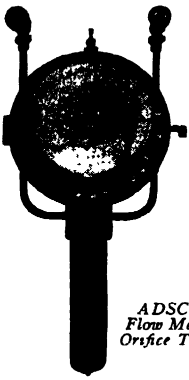
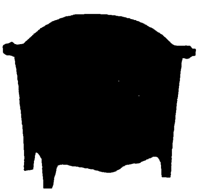
For Data on ADSCO Expansion Joints, refer to Insulation, Underground, page 1113.



ADSCO Flow Meter Orifice Type

ADSCO FLOW METER—ORIFICE TYPE

Exceptionally accurate at all rates of flow and will meter steam, water, gas or air. It is a compact unit for indicating, recording and integrating the flow and can be furnished in other combinations of these three devices. Easily installed and maintained by the purchaser. Frictionless meter mechanism, records on evenly-divided, direct-reading chart, giving a daily record from which to determine heating or processing costs Write for Bulletin No 35-83G



Rotary Condensation Meter

ROTARY CONDENSATION METER

Measures steam consumption by metering condensate from heating systems or industrial equipment. Accurate within 1 per cent and factory tested to 150 per cent of rated capacity. Compact, easily cleaned, tamper-proof and equipped with non-fogging counter mechanism. Counter reads directly in pounds Suitable for vacuum or gravity service Available in 7 sizes from 250-12,000 lb per hour capacity. Write for Bulletin No 35-80AG

ADSCO Vertical Steam Trap

ADSCO VERTICAL STEAM TRAP

A float type steam trap with or without thermostatic air by-pass for vacuum service to 15 lb pressure and gravity service to 125 lb pressure The cover with all working parts can be removed without disturbing the piping connections The trap is equipped with a reversible valve and reversible seat of stainless alloy steel Write for Bulletin No 35-86G

ADSCO Instantaneous Water Heater

ADSCO HEAT EXCHANGERS

Made in various sizes and capacities to heat or cool water, oils, other liquids or gases according to expert engineering specifications. Simple in design, sturdy in construction, dependable and economical in operation. Available in U-tube or straight tube types of heaters, economizers, condensate coolers or special units Write for Bulletin No 35-75BG, 35-76G.

Cochrane Corporation

3130 North 17th Street, **Philadelphia, Pa.**

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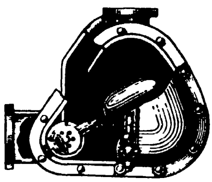
MULTIPORT DRAINERS

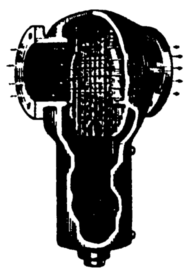
Of the multiport type, they afford unusual capacity for removing condensate or drips from purifiers, separators, jackets, radiators, pressure heating or drying coils, etc. Eliminating condensate delivers maximum heat from steam production at lower cost. Tremendous capacity assured by large port areas. Provides continuous discharge. Instantly responsive. Compact and light in weight. For pressures up to 150 lb. Publication 2925.



Multiport Drainer

ALL-SERVICE SEPARATORS

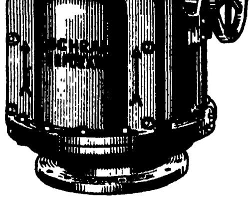
Cochrane Separators purify steam by separating out oil, slugs of water and condensate. Complete removal of entrainment is accomplished by vertical baffle ribs which guide it into a direct unrestricted fall, and a baffle area which extends far beyond the flow from the inlet pipe. Ports at the sides of the baffle prevent the purified steam from passing over the drip area and coming into contact with the entrainment. The steam flow is uninterrupted and pressure loss is minimized. Write for publication 2950.



All-Service Separator

COCHRANE MULTIPORT RELIEF VALVES

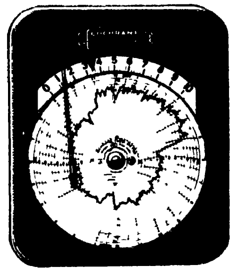
For back pressure, atmospheric relief, flow or check valve service on air, gas, steam or water lines. Positive protection against stuck, jammed or "frozen" valves as a number of small disks are used instead of one large disk. Write for publication No. 4150.



Multiport Back Pressure Valve

COCHRANE FLOW METERS

Flow meters of both mechanical and electrical types for measurement of steam, liquids and gases. Mechanical meter uses no working parts in the pressure chambers and no stuffing boxes. The electric meter measures flow by the extremely accurate galvanometer null principle. The new "Linameter" measures corrosive or viscous fluids. Publication 3010.



ROTAMETERS

Series 100 Rotameters are of the indicating area meter type, sizes 1/8 in. to 3 in., with both free and guided floats, which may be viscosity or density compensated. Tubes are Pyrex borosilicate glass, extra heavy wall, formed on mandrels, annealed, polariscope inspected and check-gaged by precision ball test. Features are:

1. Spring Float Stops
2. High Pressure Stuffing Boxes.
3. External Adjustment of Stuffing Boxes.
4. White-backed Tube
5. Interchangeable Calibration Scales.
6. Constant Tension Spring-loaded Guide Rod.

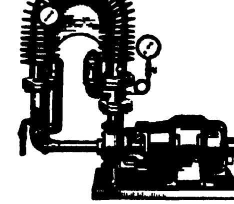
Write for publication 4135.

COCHRANE C-B HIGH PRESSURE CONDENSATE RETURN SYSTEM

In unit heaters, coil radiation, blast heaters, etc., this system will reduce fuel costs by returning condensate direct to boilers at high temperatures and pressures. Advantages are:

1. Faster Heating. 2. Higher Temperatures. 3. More Uniform Heating. 4. Lower Maintenance.

Plus 5 to 28 per cent fuel saving. Write for Publication 3250.



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Heating, Industrial and Power Plant Piping, Fittings, Hangers, Valves, Pipe Bending, Welding, Piping Supplies, Etc.

Executive Offices: **Providence 1, R. I.**

National Distributors of Thermoflex Traps and Heating Specialties

For data on other Grinnell Products, see pages 998-999

Thermoflex Specialties

The heart of all Thermoflex Traps is the Hydron Bellows.

The Hydron Bellows is formed under hydraulic pressure. This powerful internal pressure locates any weakness of any nature in the tubing. Such hydraulic pressure is many times more severe than any pressure the Trap will ever be called upon to control. Every Thermoflex Trap, therefore, is practically indestructible.

Thermoflex Traps have an exceptionally large orifice. This large orifice combined with high lift, insures fast action and freedom from clogging.

We supply Thermoflex Traps guaranteed for steam pressures of 25 lb, to 50 lb and to 125 lb. Complete information and details of typical installations will be gladly sent on your request.

Thermoflex Low Pressure Line

The entire Thermoflex line of low pressure specialties, designed for maximum steam pressure of 25 lbs, has been simplified to meet wartime needs with respect to critical materials. This simplification has been accomplished without sacrifice of quality or performance—only the appearance has been altered by the change from bronze to cast iron for the structural parts.

The new low pressure victory line includes thermostatic traps in angle pattern only, with cast iron bodies without unions, in the following sizes:

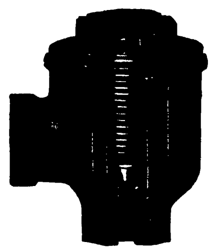
½ inch—200 sq ft rad. capacity.
¾ inch—400 sq ft rad. capacity.
¾ inch—700 sq ft rad. capacity.

A complete range of sizes of Combination Float and Thermostatic type traps continues to be available, as well as the Thermoflex Vapor Specialties for small and medium size installations.

Thermoflex Medium Pressure Traps

Thermostatic type traps, and Combination Float and Thermostatic type traps are furnished for working steam pressures in the range from 25 to 50 lbs.

Thermoflex High Pressure Traps



The No. 100A Thermoflex Trap is guaranteed for steam pressures from 50-125 lb. Must not be used where the steam temperature exceeds 400 F.

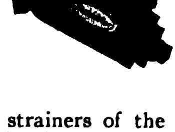
For use with all types of process work, Laundry Machinery, Kitchen Equipment, Hospital Sterilizers, Vulcanizers, Dry Kilns, Unit Heaters, Street Steam Service, etc., in fact any place that a trap is desired for service at the above pressures.

Small, compact and inexpensive.

Extra heavy body. Renewable nickel steel seat and disc. Bellows made from special bronze tubing and encased in brass sleeve to prevent distortion due to pressure.

Regularly furnished without unions.

Thermoflex Streamlined Strainers



Pipe line strainers of the self-cleaning Y-type are furnished for pressures up to 250 lbs, and in sizes ⅜ in. to 2 in. These are heavy duty strainers with semi-steel body and brass screen, which are suited to a wide field of use in removing harmful substances from pipe lines carrying steam, air and fluids.

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Mechanical Rubber Goods

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GIVES YOU EVERYTHING

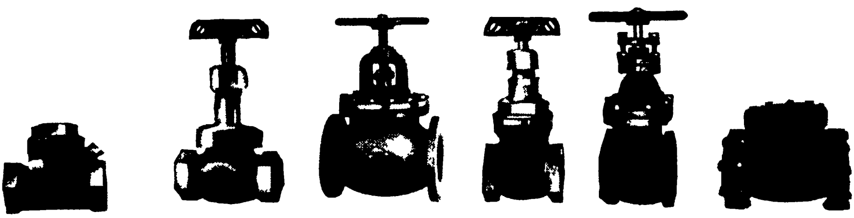

Fig 762 Bronze Regrinding Swing Check — *Fig 106A Bronze Globe, Renewable Comp Disc* — *Fig 613 Iron Body Regrinding Globe* — *Fig 370 Bronze Gate* — *Fig 325 Iron Body Gate* — *Fig 624 Iron Body Regrinding Swing Check*

OVER 500 DIFFERENT JENKINS VALVES
COVER EVERY HEATING AND AIR CONDITIONING NEED

To adequately describe the complete Jenkins line of valves requires a Catalog of more than 400 pages. There are over 500 different types and patterns of valves that bear the trusted "Diamond" trade mark. Practically speaking, Jenkins can furnish any valve that you may require for plumbing, heating, air conditioning, general industrial or engineering service.

General Classifications of Jenkins Valves Include—Bronze Valves fitted with Jenkins renewable composition disc. Bronze Regrind-Renew Valves with bevel and plug type seats. Bronze Gate Valves. Iron Body Valves fitted with Jenkins renewable composition disc Iron Body Regrinding Valves. Iron Body Gate Valves with solid wedge and double disc parallel seats. All-Iron Valves

Air Furnace Malleable Iron Valves. Cast Steel Gate, Globe and Swing Check Valves. Electrically and Hydraulically Operated Valves. Radiator Valves. Fire Line Valves. Quick-opening and Self-closing Valves, Needle Valves, Y Valves, Solder-End Valves, Stainless Steel Valves.

Other Jenkins Products Are—Composition Valves Discs exactly suited to service conditions. Sheet Packing. Gaskets, Friction Tape, Splicing Compound, Jenkins Capless Tire Valves, Mechanical Rubber Goods.

JENKINS VALVES ARE SOLD BY GOOD SUPPLY HOUSES EVERYWHERE

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LIQUID LEVEL CONTROLLERS. Direct and pilot-operated, for open or closed vessels

STRAINERS, for all steam, water, oil, gas or chemical services

REDUCING AND REGULATING VALVES. All types for all applications.

BACK PRESSURE AND RELIEF VALVES. Vacuum breakers

TRAPS. Steam, grease and air traps.

FLOAT AND BALANCED VALVES. Water feeders Altitude valves

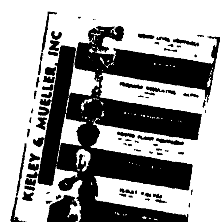
OTHER EQUIPMENT—Pump Governors — Stop - check valves Exhaust heads. Hi-low water alarms Damper regulators Steam separators Oil and grease extractors

CONSTRUCTION ***"Precision Craftsmanship is a K & M Heritage"***

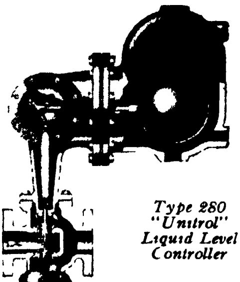
K & M controls are made in Government bronze, iron, cast steel, brass, standard or extra heavy semi-steel, special alloys, etc , with valve discs of leather, composition, bronze, rubber, etc.,—depending upon application, as indicated by our 66 years of experience and research Many controls are available with V-port, parabolic, beveled inner valve, as standard or to order This extremely wide variety assures exact suitability to service.

NATIONWIDE ENGINEERING SERVICE—We suggest you consult with our Engineering Department whenever selecting or installing equipment Also write for name of factory-trained sales engineer nearest you

"SHORT ORDER" REPAIR SERVICE—Send old equipment to us for re-building—you can count on repairs being made within 10 days to 2 weeks



SEND FOR CATALOG No. 66 -Describes and illustrates more than 200 K & M process and power controls and specialties, and includes Installation Pointers, Dimensions; Capacity Ratings, Pressure Tables; Flow Charts, etc. It is a handbook of informative data you will use frequently. Write for your copy



Type 280 "Unitrol" Liquid Level Controller

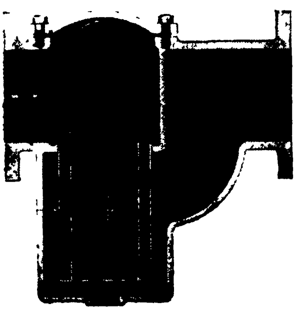
LIQUID LEVEL CONTROLLERS

DIRECT-CONNECTED TYPE for attachment directly to vessel or tank. Furnished with level to open or close valve. Sizes ½ to 1½ in with 6 in. float, and 2 to 6 in with 8 in. float. Rotary Stem Valve (Type 250) or Sliding Stem Valve (Type 260) with float cage or exposed float

KIELEY & MUELLER, Inc.—NORTH BERGEN, N. J.

REMOTROL PILOT-OPERATED TYPE for precision control where valve is at distance from float unit, or for auxiliary controlled main valve. For dead-end service, single-seated valves are furnished Sizes 1½ to 10 in. Float Cage Units (Type 270) and exposed float units (Type 271) with pressure to open or close.

UNITROL SELF-CONTAINED TYPE 280—Exclusive integral design, for service on recirculator systems of condenser hotwells, deaerators, etc. No packing glands, stuffing box or other restrictive elements Use as feed or drain regulator. Sizes ½ to 4 in with 8 in. float.



Type 330 Basket or Suction Strainer (Above)

STRAINERS

"Y" TYPE 340—Minimum pressure drop. Installation may be vertical or horizontal. Pressures to 600 lb. Screwed ends from ¼ to 3 in.; flanged ends from 1½ to 6 in Strainers in bronze, stainless steel, Monel, wire cloth or alloys; 1/64 in., 1/32 in. or larger perforations.

BASKET TYPE 330—Screwed ends ½ to 2 in.: flanged ends 2 to 18 in.

STRAIGHT FLOW TYPE 342—Where space limitations and minimum pressure loss are factors. Sizes 1½ to 6 in. with screwed ends, 2 to 12 in. with flanged ends

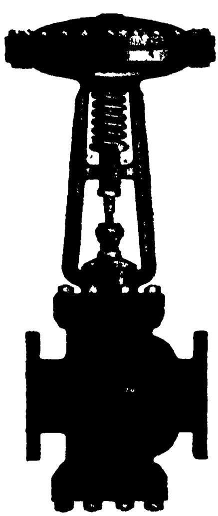
OFFSET TYPE 346 -Screwed ends only; ½ to 2 in with screwed covers. 2½ to 3 in. with bolted covers.

REDUCING AND REGULATING VALVES

SPRING-LOADED for steam, water, oil, gas, air or chemicals Pressures to 300 lb; reduced pressures 1 lb to 75 per cent of initial pressure, not exceeding 125 lb Pressure set by simple adjusting screw.

Type	Size In	Connections	Service
418	½ to 2	Screwed	Primary valve in double reduction service.
419	2 to 12	Flanged	Dead end, steam or hot oil
420	1 x 2 to 1½ x 2	Screwed	Primary valve in double reduction service.
	1¼ x 2½ to 2 x 4	Scr Inlet Fla Outlet	
	2 x 4 to 12 x 16	Fla Inlet Fla Outlet	
421	2 to 12	Flanged	Dead end, air, gas, cool oil or water.
422	½ to 3	Screwed	Ordinary reduction in process, heating or refining, steam, air or gas
	1 to 12	Flanged	
423	½ to 2	Screwed	Dead end, steam or hot oil
425	2 to 8	Flanged	Dead end, water, air, gas or cool oil

Type 422 Semi-Balanced Spring Loaded Reducing Valve (Below)



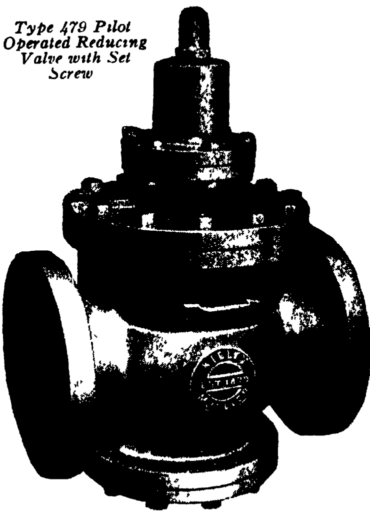
SPRING-LOADED "1900" VALVE—Famous for low cost Initial pressures to 150 lb.; reduced pressures 10 lb. to 75 per cent of inlet pressure. Also made for reduced pressures from 2 lb Sizes ⅜ to 2 in.

TYPE 449 for hot water tanks, sterilizers, etc , in dead end service

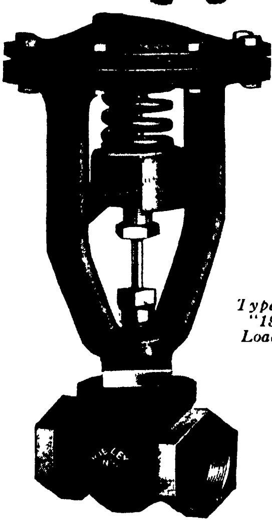
TYPE 451 for water, air or gas service with tight shut-off.

TYPE 450 where tight shut-off is not required; semi-balanced.

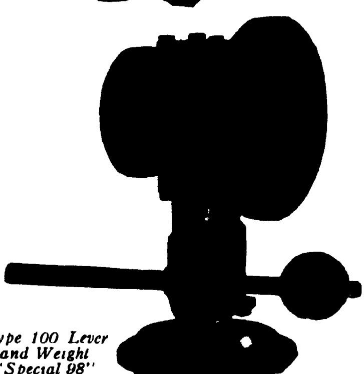
SPRING LOADED "SIMPLEX" TYPE 461 for simple pressure reduction of water, air, gas, cool oil, etc. Initial pressures to 150 lb; reduced pressures 10 to 75 lb; ½ to 2 in



Type 479 Pilot Operated Reducing Valve with Set Screw



Type 450 Special "1900" Spring Loaded Reducing Valve



Type 100 Lever and Weight "Special 98" Reducing Valve

KIELEY & MUELLER, Inc.

WEIGHT-LOADED VALVES for close regulation of inlet pressures to 250 lb; reduced pressures from 40 lb to medium vacuum.

TYPE 400, semi-balanced, ½ to 3 in., screwed ends; 1 to 12 in., flanged ends.

TYPE 401, single-seated, ½ to 2 in., screwed ends; 2 to 8 in., flanged ends.

TYPE 402, expanded outlet. 1 x 2 in. to 1¼ x 2 in., screwed inlet and outlet; 1¼ x 2½ in. to 2 x 4 in., screwed inlet and flanged outlet; 2 x 4 in. to 12 x 16 in., flanged inlet and outlet.

WEIGHT-LOADED SPECIAL "98"—Widely used regulating and reducing valve. Close regulation with utmost simplicity. Initial pressures to 125 lb; reduced pressures 0 to 25 lb Single or double-seated; screwed and flanged ends; ½ to 12 in.

PILOT-OPERATED HAND WHEEL TYPE 481 (self-contained) and 481A (with control line) for extremely close regulation. Initial pressures to 250 lb; reduced pressures 10 lb to 80 per cent of inlet pressure Special diaphragm for reduced pressures down to 2 lb; ½ to 2 in.

PILOT-OPERATED SET SCREW TYPE 479 (internal pilot-operated) and 479X (Outside operated) for precision regulation in steam service to 600 lb and reduced pressures of 2 lb to 80 per cent of inlet pressure. Shuts off tight for dead end service 2 to 12 in with flanged connections.

TYPE 476 for constant reduced water pressure regardless of inlet fluctuations Pressures to 250 lb, reduced pressures 10 lb to 80 per cent of inlet. Higher pressures on inquiry.

AUXILIARY-OPERATED TYPE—Regulation within 1 per cent of initial pressure, regardless of variation Steam, water, air, oil or gas service. Reduced pressures 0 to 90 per cent. Finned bonnet for extreme temperatures.

TYPE 492 (pressure to close) and 492 R (pressure to open) in ½ to 12 in.

TYPE 493—Single seated, ½ to 8 in.

DIAPHRAGM MOTOR VALVES for air-operated instruments or auxiliary pilot units controlling steam, liquids, gases, air—pressures to 900 lb; and temperatures to 850 F.

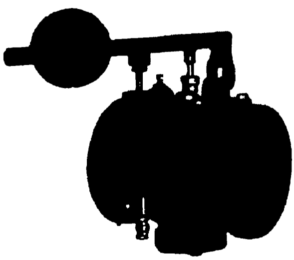
TYPE 484 (pressure to close) and 484R (pressure to open) in ¼ to 12 in.

TYPE 485, single-seated for tight shut-off. Pressure to open or close. ½ to 10 in.

TYPE 488 (semi-balanced) and 489 (single-seated) with cooling-fins for extreme temperatures.

KIELEY & MUELLER, Inc.—NORTH BERGEN, N. J.

BACK PRESSURE AND RELIEF VALVES

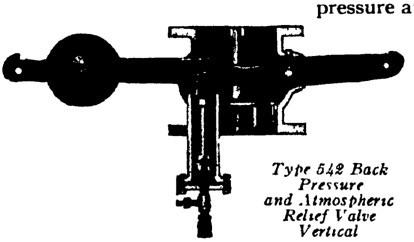


Type 520 Back Pressure Valve, Horizontal

BACK PRESSURE HORIZONTAL TYPE for non-condensing systems. Sensitive, noiseless. Lever and spring-loaded models, single or double seat. Easily adjustable.

Type	Size, In	Set Pressure Range, Lb	Type	Size, In.	Set Pressure Range, Lb
520	2 to 6	0 to 15	534	2 to 6	2 to 30
	8 to 12	0 to 10		8 to 12	2 to 15
	12 to 24	0 to 7		14 to 20	2 to 10
524	6 to 12	0 to 20	536	6 to 12	2 to 30
	14 to 18	0 to 15		14 to 18	2 to 15

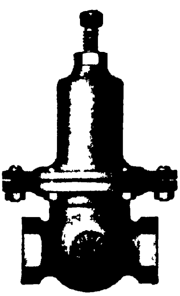
BACK PRESSURE AND ATMOSPHERIC RELIEF VALVES for any service requiring constant back pressure and positive shut-off.



Type 542 Back Pressure and Atmospheric Relief Valve Vertical

TYPE 526 (single-seated; horizontal) and 542 (single-seated; vertical) are lever-operated; for pressures to 15 lb, sizes 2 to 12 in.; pressures to 7 lb, 14 to 24 in. Higher pressures on application.

TYPE 538 (single-seated, horizontal) and 554 (single-seated; vertical) are spring-loaded; 2 to 6 in for pressures 2 to 30 lb; 8 to 12 in. for pressures 2 to 15 lb; 14 to 24 in. for pressures 2 to 10 lb.



Type 590 Diaphragm Relief Valve, with Set Screw

DIAPHRAGM RELIEF VALVES—Extremely compact and dependable for relieving pressures of 5 lb to 100 lb with wide range of applications in steam, gas, air, water, oil, etc. Sizes 3/8 to 2 in Globe or angle patterns; set screw, hand wheel or test lever

STEAM TRAPS

INVERTED BUCKET TYPE 720, discharges intermittantly. Will not air bind Pressures 5 to 250 lb Large capacity. Sizes 1/2 to 2 in

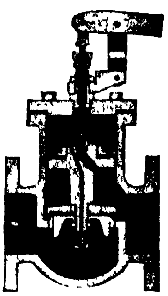
TYPE 716, similar but smaller in size. Low cost. 5 to 100 lb.

OPEN BUCKET STEAM TRAPS for pressures from 1 to 250 lb Sizes 1/2 to 3 in.

BALL FLOAT TYPES for continuous discharge to handle large quantities of condensate. 1/2 to 3 in.

TYPE 724—1 to 30 lb. **TYPE 726**—30 to 125 lb. **TYPE 727**—1 to 10 lb.

FLOAT AND BALANCED VALVES



Type 834 Globe Float Valve for Cold Water

BALANCED VALVES TYPE 813, single seated for dead end service on pressures to 250 lb. Screwed ends from 1/2 to 2 in. and flanged ends from 2 to 6 in.

TYPES 850 (rotary stem) and 860 (sliding stem) double seated. Operates by cable, float, thermostat, etc. 1/2 to 4 in. with screwed ends, 1 to 10 in. with flanged ends.

FLOAT VALVES FOR HOT WATER TYPES 844 (globe) and 846 (angle) Service to 150 lb and 350 F. 1/2 to 2 in. with screwed ends. Corrosion-resistant types also available.

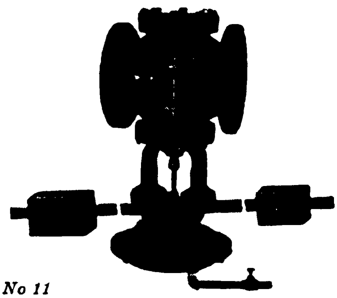
FOR COLD WATER TYPES 834 (globe) and 838 (angle). To 250 lb. Extremely sensitive. Free from pounding or pulsating. 2 to 6 in. with screwed ends. 2 to 12 in. with flanged ends.

Mueller Steam Specialty Co., Inc.

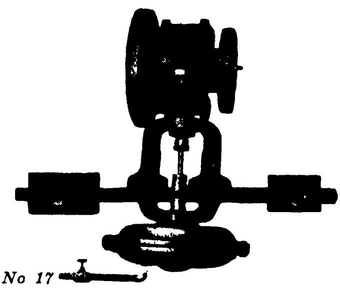
40-20 22nd Street, **Long Island City, N. Y.**

Steam, Water, Air, Oil and Gas Specialties for Heating and Power Plants

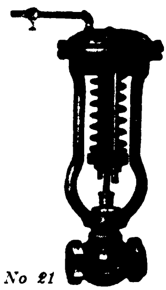
Pressure Reducing Valves—Straight Pattern and With Increased Outlet



No 11



No 17

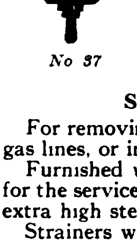


No 21

No. 11—For Vacuum, Vapor and Low Pressure Heating Systems Initial Pressures, up to 200 lb; Reduced Pressures, 0 to 10 lb

No. 17 and 21— For automatic control of reduced pressures on dead-end service, requiring a tight closing valve, such as tank heaters, kitchen utensils, sterilizing apparatus, laundry equipment, kettles, cookers, driers, etc Initial Pressures up to 200 lb. Reduced Pressures 0 to 150 lb.

Constructed with full globe bodies. Center guide eliminates the wings on discs, and increases efficiency, assures minimum noise and prolongs the life of the seats and discs. Lever and weight operates on a steel roller bolt, assuring a most sensitive valve. Spring type furnished with special long springs for sensitive operation and wide ranges of reduced pressures.



No 37

Water Pressure Reducing Valves

For controlling water pressures that require automatic and positive control of reduced pressures. Compact, and have full flow and renewable composition discs.

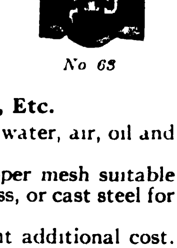
Standard and extra heavy weights suitable for all initial pressures and wide ranges of reduced pressures.

Diaphragm Operated Water Relief Valves

For relieving excessive pressure of hot or cold water, air, oil or gas

Very sensitive, respond quickly.

Their construction is durable and compact, and accessible without disturbing pipe connections



No 63

Sediment Strainers for Steam, Water, Oil, Gas, Air, Etc.

For removing scale, cuttings, and other foreign matter from steam, water, air, oil and gas lines, or in connection with valves, pumps or other apparatus

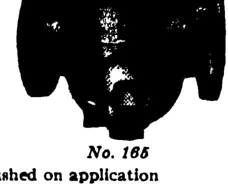
Furnished with iron bodies, plain or galvanized, with brass or copper mesh suitable for the service. Can be furnished all brass or nickel-plated iron or brass, or cast steel for extra high steam pressures, at additional cost.

Strainers with special mesh or of special metal furnished at a slight additional cost.

No. 125

Figures Nos. 145 and 165 Strainers have closed or open bottom baskets, so that the basket can be removed for cleaning, or all dirt and sediment can be blown out through bottom blow out connection.

No 145



No. 165

Catalogue and Bulletins covering our Complete Line gladly furnished on application

Wright-Austin Co.

309 West Woodbridge St., **Detroit 26, Mich.**

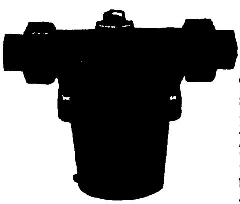
Established **50** Years Ago

PRODUCTS—Steam Traps, Strainers, Air Traps, Steam and Oil Separators, Compressed Air Purifiers, Exhaust Heads, Boiler Feeders and Controllers, Alarm Water Columns, Water Gauges, Trycocks.

"Airxpel" Bucket Type Steam Traps

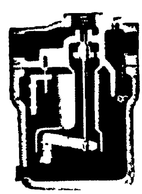
Are "double duty" traps, because they automatically discharge both air and condensate.

Union connections make them easy to connect up. Also, furnished with screw connections when desired. They save money for fittings and installation labor, by having straight through horizontal pipe connections



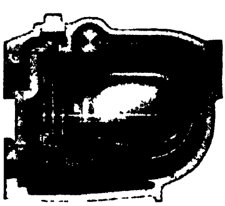
The Cub sizes are made in ½ in, ¾ in, 1 in Especially suitable for individual unit drainage on heating and process equipment

Also three "Master" sizes ½ in. to 2 in., for general service.

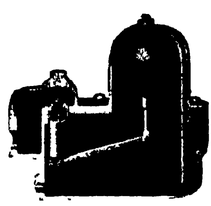


"Combination" Steam Traps

Float Type with internal thermostatic air bypass and strainer for pressures 0 to 40 lb. A modernly designed and very successful trap for vacuum and pressure heating.



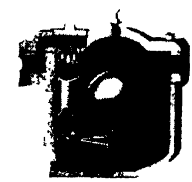
"Victor" Low Pressure Steam Traps

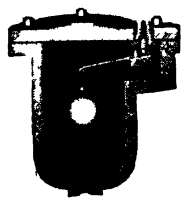


A heavy duty trap for large volumes of condensation at low pressures.

"Emergency" Float Type Steam Trap

Three valve trap with large capacity at high pressures. An exceptionally reliable trap for use in inaccessible places.





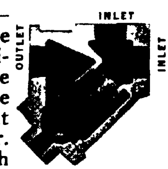
Air Relief Traps

For relieving air from forced circulation hot water heating systems, water supply lines, closed tanks, receivers, pumps, etc.

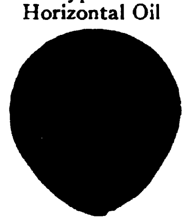
"Tuway" Strainer

May be used two ways—as a straight-way or angle strainer, in either horizontal or vertical pipe line, because it has the choice of two inlets at right angles to one another.

For cleaning, flush through blow-off connection, or remove screen by unscrewing bottom plug.



Separators—Steam and Oil

Type "A" Vertical Steam	Type "S" Horizontal Oil
	

We make separators of every type and all sizes for all pressures.

Exhaust Heads

Designed to eliminate noise and spray. Three types to select from—the "Cyclone" Heavy Duty, and Standard Galvanized Steel—also, the cast iron type, to remedy all conditions. Sizes 1 in. to 48 in.

Send for descriptive Bulletins on any of the items listed on this page

Yarnall-Waring Company

Manufacturers of YARWAY **Steam Specialties**

7600 Queen Street, **Philadelphia 18, Pa.**

YARWAY IMPULSE STEAM TRAPS

Construction—The Yarway Impulse Steam Trap is unique in that there is only one moving part, the simple valve F. This trap is made of bar stock throughout, no castings used. Body and bonnet of cold rolled steel, cadmium plated; cap of tobin bronze, valve and seat of heat treated stainless steel. For pressures 400 to 600 lb, trap is all stainless steel.

Operation—Movement of valve (F) is governed by changes in pressure in control chamber (K). At lower temperatures, condensate bypassing continuously through orifice in center of valve reduces chamber pressure below inlet pressure and valve opens, allowing free discharge of air and condensate through seat. As condensate approaches steam temperature, low chamber pressure causes vaporizing and the increased volume builds up pressure in control chamber, closing valve (F).

Advantages

Light Weight—Yarway traps need no support—½ in. trap weighs only 1⅜ lb. 2 in. trap weighs 8⅝ lb.

Small Size—They practically eliminate radiation losses—can be installed in cramped quarters—½ in. trap measures 2¼ in. long—2 in. trap, 4¾ in. long.

Will not air bind.

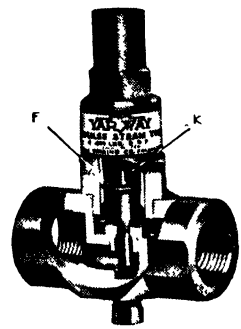
Require no priming.

Insure quick heating.

Operate on exclusive Impulse principle (*U. S. Patents No. 2,051,732 and 2,127,649.*)

Low Price—Often cheaper than repairing old traps.

Factory set to operate at all pressures up to 400 lb (or 600 lb) without change of valve seat.



List Prices, Weights and Dimensions

No. 60 Series—up to 400 lbs. and

No. 120 Series—up to 600 lbs.

Size	Trap Complete	Weight Pounds	Length Inches
½" Nos. 60 or 120	$15 00	1¼	2⅝
¾" Nos 61 or 121	22 00	2	3
1" Nos 63 or 123	31 00	2½	3⅜
1¼" Nos 64 or 124	48.00	4	3¾
1½" Nos 66 or 126	68 00	5¾	4¼
2" Nos 67 or 127	90 00	8½	4¾

Send for descriptive Bulletin T-1738.

Yarway "Fine-Screen" Strainers offer better protection against rust, scale and dirt for all steam equipment.

Made in six standard sizes from ½ in. to 2 in. Cadmium plated, inside and out. High grade Monel woven-wire screens Many thousands in use. Write for Bulletin S-200.

YARWAY EXPANSION JOINTS

All-steel welded construction; light but strong. Chromium covered sliding sleeves.

Cylinder guide and stuffing box integral, assuring perfect alignment. Internal limit stops. Gun-pakt and Gland-pakt types: Gun-pakt (illustrated) fitted with screw guns which permit addition of plastic packing while joint is under pressure. Sizes 2 in. to 24 in., single end or double end, flanged or welding ends; 150, 300 and 400 lb pressures. Also all-brass joints, ¾ in. to 2 in. For additional details send for Bulletin EJ-1909.

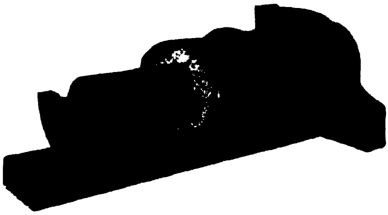
ADSCO
PRODUCTS for STEAM SERVICE

AMERICAN DISTRICT STEAM COMPANY

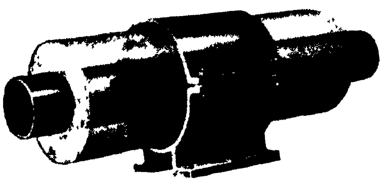
NORTH TONAWANDA, N.Y.

IN BUSINESS OVER SIXTY YEARS

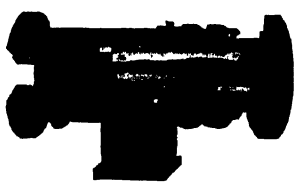
Branches and Agents in Principal Cities



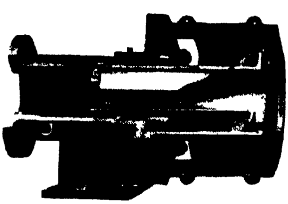
Tile Conduit with ADSCO Filler Insulation—a "Fiberglas" Product



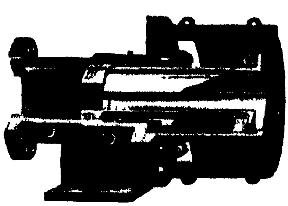
Alignment Guide



Internally Guided Joint



Internally-Externally Guided Joint



Piston-Ring Type Joint

OVER SIXTY YEARS EXPERIENCE IS BUILT INTO THE DESIGN AND MANUFACTURE OF DEPENDABLE ADSCO PRODUCTS FOR PIPE LINES

For over sixty years ADSCO engineers have specialized in the design and application of pipe fittings and accessory equipment for underground and surface steam, water, oil and other piping systems. An extensive, modern plant including foundry, machine shop, casing mill, shipping and storage facilities enable ADSCO to produce high grade products by skilled workmen under expert supervision.

LEADING MAKERS OF EXPANSION JOINTS

As pioneer manufacturers of expansion joints for pipe lines, ADSCO is the largest single producer of such equipment in the world. We offer the most extensive line of packless and slip type joints in various types to meet the requirements of any pipe line expansion and contraction problem. In addition, ADSCO produces all of the related equipment necessary to the permanent installation of efficient pipe lines, including tile conduit for underground lines, pipe supports, saddle plates, alignment guides, steam traps, condensation and flow meters, storage and instantaneous water heaters, strainers, manhole frames, and vapor heating specialties.

ENGINEERING ASSISTANCE

ADSCO engineers welcome the opportunity of working with industrial plants, utility companies, colleges, institutions, and government departments in the solution of their pipe line expansion and contraction problems and correspondence is invited giving the details of any proposed piping installation.

WRITE FOR ADSCO CATALOG No. 35

All ADSCO products are illustrated and described in the latest ADSCO Catalog No. 35 containing over 136 pages of informative data for the specification and purchase of dependable products for underground or surface pipe line distribution systems. Write for your copy today to the American District Steam Company, 65 Bryant St., North Tonawanda, New York.

H. W. Porter & Co.

INCORPORATED

817-G Frelinghuysen Ave., **Newark 5, New Jersey**

Permanent Protection and Insulation for Underground Pipe Lines

BALTIMORE, MD. CHARLOTTE, N C RICHMOND, VA.

Therm-O-Tile

REG US PAT OFF

STEAM CONDUIT SYSTEMS

"Permanent" Protection—Therm-O-Tile has been on the market over ten years. Its design is well known to all leading heating and ventilating engineers but we would again like to point to the importance of "Permanent protection."

In issues of this book previous to 1944 we stated that Therm-O-Tile is "A complete conduit system for the permanent support, protection, and insulation of underground mains for steam distribution"

Not Merely "Temporary"—We therefore want to emphasize the word "permanent." It is not difficult to provide temporary protection and insulation for underground pipe lines. Threads and joints don't fail immediately, and foundations don't sag immediately, but unless the job is *properly* done it won't be long before water seeps in and ruins the insulation. With wet and spoiled insulation, efficiency drops drastically. And, unless the conduit is built on a solid foundation there will be sagging and collection of water in pockets. Hence the importance of a "permanent" job

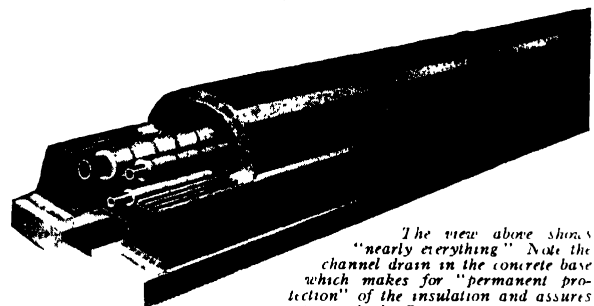
Therm-O-Tile Drainage System—In the Therm-O-Tile concrete base there is a drainage channel—clearly visible in the photo above—which carries off all water that may enter the conduit from any source, thereby keeping the insulation *permanently* dry. Drainage is entirely internal. The channel is accurately and *permanently* sloped so that condensate or other pockets cannot form. Open to thorough inspection at any time at manholes. Amply large to keep the pipe space dry at all times.

"Spread Footing" Foundation—So that there will be no settling or sagging the Therm-O-Tile "Spread Footing" foundation base is a thick concrete slab poured directly in the trench bottom as shown in the photograph. It is steel reinforced or placed on driven piles when installed over filled or boggy ground—constructions that are necessary for *permanent* protection

See Previous Issues—In issues of this GUIDE previous to 1944 we gave details regarding the Tile Envelope which produces 27 different conduit cross sections We told about the ideal accessibility of this conduit, its great strength, how it is water-proofed, and how the insulation is applied. For complete information ask for a copy of Bulletin 381

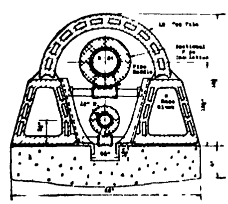
Help Win the War—Therm-O-Tile is made almost entirely of "Non-Critical Materials." Less than 2 per cent of the entire conduit, by weight, is made of metal—the pipe supports are the ONLY metal.

Competitive in Cost—Despite the high efficiency, strength, dependability, and other highly desirable features that are obtained in Therm-O-Tile, it is nevertheless competitive in total first cost. Thorough investigation is invited.



The view above shows "nearly everything" Note the channel drain in the concrete base which makes for "permanent protection" of the insulation and assures continous high efficiency

We Co-operate—Our engineers have had unusual and broad experience in the design and installation of steam conveying equipment for an exceptional range of purposes They will gladly cooperate in solving your own individual problems of this nature.



Single or Multiple Pipe Lines Using Sectional Pipe Insulation.

Showing a typical piping arrangement in Therm-O-Tile when there are two pipe lines Note the channel drain which "permanently protects" the insulation

Representatives— Therm-O-Tile is sold and installed by Johns-Manville Construction Units in all principal cities.

The Ric-wiL Company

INSULATED PIPE CONDUIT SYSTEMS

Union Commerce Bldg., Cleveland, Ohio • Agents in Principal Cities

There is a Ric-wiL insulated conduit system engineered to your specific needs—the transmission of steam, hot water, oil, hot or refrigerated process liquids—providing heat transfer with the lowest possible loss.

1. RIC-WIL INSULATED PIPE UNIT—SINGLE PIPE SYSTEM

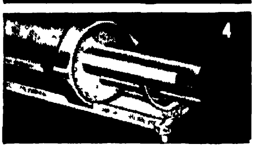
Prefabricated complete units—pipe as specified, thoroughly insulated, in helical corrugated conduit, coated and wrapped with asphalt saturated asbestos felt. 21-ft lengths for speedy installation. For underground or overhead systems.

2. RIC-WIL INSULATED PIPE UNIT—MULTIPLE PIPE SYSTEM

Any specified combination of pipes in prefabricated conduit —insulated and protected the same as the single pipe system. Any or all of the pipe lines may be specially insulated to meet job requirements.

3. RIC-WIL INSULATED PIPE UNIT—FOR PROCESS LIQUIDS

An adaptation of the multiple system used where a steam or hot water line heats fluids in other lines. Pipes are insulated from the exterior but not from each other. Sizes and specifications as required—conduit same as for other insulated pipe units.

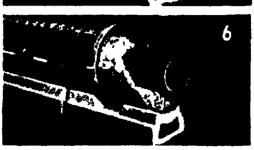


4. RIC-WIL STANDARD TILE CONDUIT—TYPE F

Vitrified glazed A.S.T.M. Standard Tile Housing—acid and waterproof—with foundation type base drain supporting weight of piping through correctly engineered pipe supports. Positive locked-in-place cement seals on sides and ends For single or multiple pipes.

5. RIC-WIL SUPER TILE CONDUIT—TYPE F

Same advantages as Standard Tile but with walls approximately double thick for strength under heavy traffic or where overhead load is above normal. Will support concentrated static load of 6 tons per wheel under actual installation conditions. Base drain of extra-heavy tile.



6. RIC-WIL CAST IRON CONDUIT—TYPE F

Heavy reinforced cast iron conduit for use where underground pipe lines run close to or under railroad tracks. Durable, water-tight and vibration-proof. Positive locked-in-place cement seals on sides and ends with metal clamps for extra tightness.

7. RIC-WIL TILE CONDUIT—UNIVERSAL TYPE

Where installation conditions dictate the use of a concrete pad Ric-wiL Universal Tile is recommended. Side walls are double-cell vitrified trapezoidal block design. Arch may be Standard Tile, Super-Tile, or Cast Iron.

8. RIC-WIL TILE CONDUIT—TYPE DA

For oil or process liquids where conduit must be insulated but individual lines are not insulated from one another. Insulation is a diatomaceous earth lining, moulded and keyed to inside of tile. May also be used (Type DF) with fibre insulation for steam heat, power and superheated steam. Applicable to Standard, Super-Tile and Cast Iron.

Ric-wiL accessories are available in all type systems; standard and special fittings, factory fabricated or field fabricated expansion devices, alignment guides, and anchors. Descriptive bulletins on request.

GET THE ORIGINAL—SPECIFY RIC-WIL

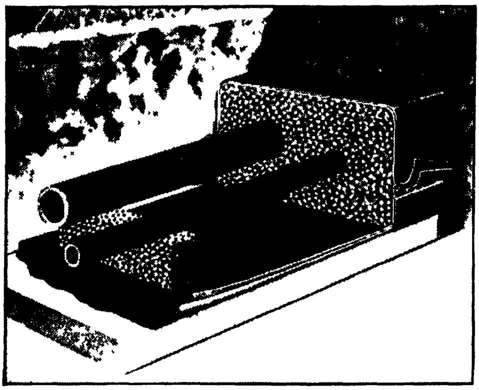
Universal Zonolite Insulation Co.

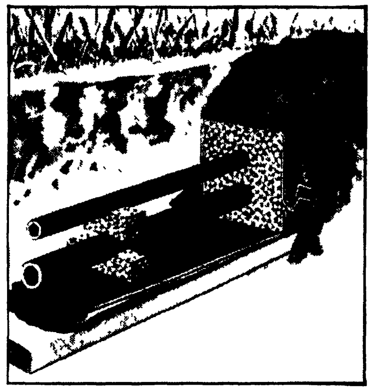
135 South LaSalle Street, **Chicago 3, Illinois**

ZONOLITE

b-t-u
BEST TYPE UNDERGROUND
INSULATION

The Simplified Method of Insulating Underground Piping





Zonolite b-t-u Insulation Technique

Zonolite b-t-u Insulation is the modern, superior insulation for underground pipes. The procedure is as follows:

1. A structural concrete base or pad is laid on the bottom of the trench to support pipe and insulation.
2. Waterproof covering is placed over base.
3. Precast Zonolite insulating concrete blocks are placed at intervals on this base and pipes are laid on them.
4. When pipes are placed, they are covered with a parting medium to prevent bond so that pipes are free to expand and slip through insulation.
5. Forms are set to receive Zonolite b-t-u Insulation which then is poured monolithically around and over pipes.
6. When insulation sets, forms are removed, and a waterproof covering is placed over top and sides of insulation.

COMPOSITION

Zonolite b-t-u Insulation is composed of Zonolite Insulation Aggregate, Portland cement, Zonolite Waterproofing Admix and integral waterproofing compounds.

ONLY Zonolite Offers These 7 New Basic Features

1. **High Insulating Efficiency** far in excess of ordinary practice. Zonolite b-t-u Insulation is applied from 4 in to 6 in thick around pipes, reducing heat loss to a minimum Its solid continuous embedment also eliminates heat losses due to voids, leaky joints and connections
2. **Permanence Unaffected** by rot, fire, wet, exposure or temperatures up to 1800°F.
3. **Waterproofed 3 Ways** outer envelope the insulation itself . . . the heat-formed integral barrier seal.
4. **No Voids, Cracks, Joints** because no hollow boxes, tunnels, conduits or drains
5. **Monolithic Structure** . . strong, amply protective, holds slope and alignment. Will carry heavy loads under highway traffic without damage.
6. **Foolproof Installation** supervised by competent, authorized engineers and contractors. Very rapid.
7. **Reasonable Cost** for any number or size of pipes. Cuts and patches easily for new connections

For Zonolite b-t-u Folder or for Engineering Service and Specification

***Write to* Dept. AHV-15.**

American 3 Way-Luxfer Prism Co.

24 N. Pulaski Rd., **Chicago-24, Ill.** | 3-WAY Daylighting Products | 103 Park Ave., **New York-17, N. Y.**

AMERICAN GLASS BLOCK SKYLIGHTS

Light-Up—the* AMERICAN *Way!

American 3-Way Rooflights make use of Glass Blocks of special design and strength, manufactured by the proven process, incorporating four way design lenses on inner surfaces of plates, also resulting in uniform even light distribution over wide areas, leaving top and bottom surfaces smooth for easy cleaning.

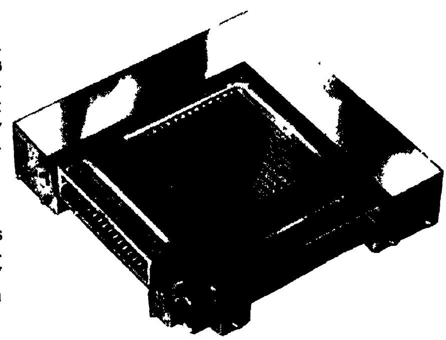
Glass Blocks are 9 in x 9 in. x 2½ in. and spaced approximately 10⅝ in on centers.

Low Heat Transfer

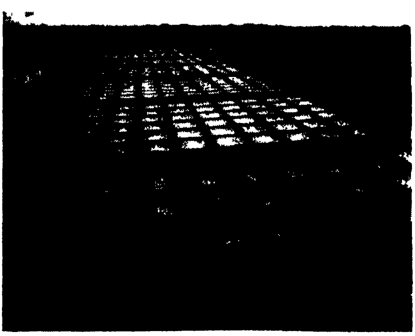
Tests conducted by methods suggested by the A.S H.V.E. Code show that Glass Block Rooflights have about two and one-half times the insulating value of sheet metal skylights with no heat losses by "escape," since the construction is air-tight.

Solar Heat Transmission

Reduction in total solar heat gain as compared with ordinary windows is indicated by relative values given in Table 7 on Page 145 and Table 8 on Page 146 in Chapter 7



Section of American Glass Block Skylight Showing Method of Block Application



American Glass Block Skylight

Triple Plates of Glass

Magnalite Diffusing Glass units may be attached to under side of glass block construction thus making for effective uniform light diffusion and even distribution; also very effective in condensation problems

Insulated Construction

Construction of rigid reinforced concrete grids can be arranged with insulation materials sufficient to approximately equal the performance of the glass blocks.

Condensation

Due to the nature of the grid construction where insulation materials are employed with semi-vacuum glass blocks assemblies there is little or no tendency for condensation to form on the under side. Should relative high humidities or abnormal conditions exist, further insulation treatment can be provided.

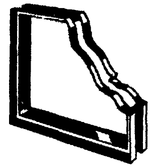
Glass Block Assemblies for all off-vertical arrangements are available. Details will be furnished on request.

3-Way Glass Block Skylights may also be furnished without special insulation treatment in reinforced concrete grid construction.

Write for Complete Information

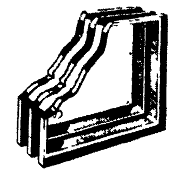
Libbey•Owens•Ford Glass Company

Nicholas Building, Toledo 3, Ohio



THERMOPANE
Insulating Glass

L-O-F Thermopane is a transparent factory-fabricated insulating glass unit composed of two or more panes of glass separated by ¼ in. or ½ in. of dehydrated captive air, hermetically sealed at the edges in the factory with a metal-to-glass bond.



Thermopane is Fabricated to ordered sizes at Libbey-Owens-Ford's factory. The glass, before being assembled into Thermopane units, is specially cleaned before the patented Bondermetic seal is applied. The layer of air inside the Thermopane units is scientifically cleaned, dried and hermetically sealed. The patented, metal-to-glass seal bonds the two or more panes of glass into one unit to prevent dirt and moisture from entering the air space

Thermopane Reduces the coefficient of heat transmission; increases the room-side surface temperature, thus promoting radiant comfort and lowering the dew point; furnishes a control of light quantity and quality through combinations of various types of glass, and deadens sound transmission to some degree.

Double-glass Thermopane: Glazing of wood or metal windows or doors for practically any purpose in structures requiring heating or air conditioning.

Triple-glass Thermopane: Large stationary units such as insulated glass walls in homes, apartments, public and commercial buildings and in show windows for refrigerated display where temperature differential must be considered.

Quadruple-glass Thermopane: Engineered to meet low temperature and high humidity conditions.

Thermopane Uses are many, but may be briefly summarized as below:

Thermopane Units provide a high resistance to heat flow, varying with the number of panes and the thickness of the air space used. In summer the low heat transmission coefficient reduces the load on air conditioning systems. In winter it saves heat. The greater efficiency of Thermopane makes it possible to incorporate larger windows in houses and keep the cost of fuel constant. For example, a house could have 107 sq ft of Thermopane instead of 35 sq ft of single glazing and not lose any more heat. Thermopane permits the influx of solar heat in exterior glazing of buildings without a prohibitive compensating loss from conduction

The over-all heat transmission coefficient U varies with the ranges of temperature at which the coefficient is determined. For most practical heat loss calculations coefficient U can be the value determined at 0 deg outside temperature, 70 deg inside temperature, 51 mph outside air velocity, 0.25 mph average inside air velocity. The following table gives such values:

Number of Panes	Thickness of Panes	U Values	
		¼ In Air Space	½ In Air Space
2	⅛″	0.62	0.57
	¼″	0.57	0.55
3	⅛″	0.42	0.37
	¼″	0.39	0.35

Note: U for single ⅛ in thick glass = 1.14, R = 0.88
U for single ¼ in thick glass = 1.07, R = 0.94

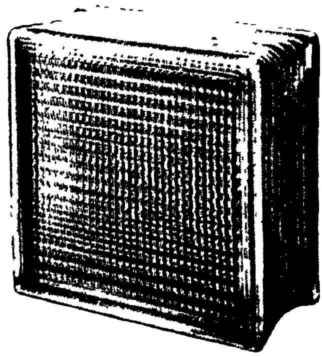
The Room Side Surface Temperature of Thermopane is considerably higher than that of single glazing Usually radiators or registers are near glass areas in buildings to offset conducted heat loss from a room and radiant loss from the bodies of persons near cold glass areas The higher glass surface temperature of Thermopane greatly reduces the amount of heat which must be supplied, permitting more flexibility in room design

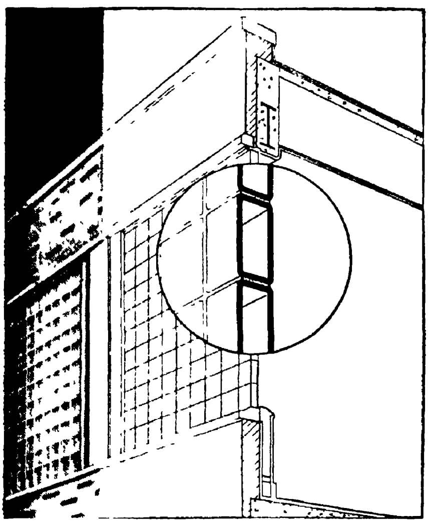
Another Important Benefit from Thermopane is the prevention of frost or condensation from forming on the room side surface of a single pane of glass in winter due to higher room humidity The absence of condensation on the room side surface of glass is of considerable importance where clear visibility is a factor as in residences, all types of commercial or industrial buildings and refrigerated display spaces.

More Complete Information on Thermopane is available by writing to Libbey-Owens-Ford, or its district office nearest to you, and requesting technical data sheets prepared by Don Graf, a general booklet about the product, and a brochure which discusses Solar Housing.

Insulux Glass Block Give Better Control of Interior Conditions

Insulux Glass Block are hollow, partially evacuated block, 3⅞ inches thick, with ribbed or smooth faces Laid up in mortar in solid panels, they form a light-transmitting area that also offers high insulation value Proper use of Insulux Glass Block results in better control of interior conditions, and therefore greater efficiency and lower initial and operating costs for cooling and heating plants





Better Insulation

The cross-section drawing above shows why Insulux Glass Block panels give higher insulation than ordinary light-transmitting areas The glass block, partially evacuated and with thick faces, lower the conductivity and solar heat transmission of the light-transmitting area. Air infiltration is eliminated. The better insulation provided by Insulux is a factor in planning air conditioning and heating equipment and operating costs.

Lower Heat Transmission

Tests on conductivity of Insulux Glass Block show that the heat transmission of Insulux is approximately the same as for a concrete wall 16 inches thick or a brick wall 8 inches thick The U factor for smooth face block is .49 Btu per sq ft per hour per degree difference in temperature. For ribbed block, the U factor is 46. This test data is available for inspection by engineers

Reduction of Solar Heat

In a comparative test of solar heat transmission, a single glazed steel sash transmitted 94 per cent more heat than an Insulux panel. As with sash, however, Insulux panels transmit less solar heat if properly oriented and well shaded. There is variation in the solar heat transmission of different designs of Insulux—data will be furnished on request.

Designs, Sizes, Erection

Insulux Glass Block are made in 11 attractive designs for both residential and industrial use They are available in three sizes: 5¾ in. square, 7¾ in. square and 11¾ in. square. Panels are easily and quickly erected by bricklayers. Additional technical data, installation information, etc , will be furnished on request.

Pittsburgh Corning Corporation

632 Duquesne Way, **Pittsburgh 22, Pa.**

Distribution through Pittsburgh Plate Glass Company warehouses in principal cities and by the W. P. Fuller Company on the West Coast.

PC **Glass Blocks** allow the economical use of large glass areas, reduce heat loss in cold weather and materially aid air-conditioning. This is because each PC Glass Block contains a sealed-in dead-air space that is an effective retardant to heat transfer.

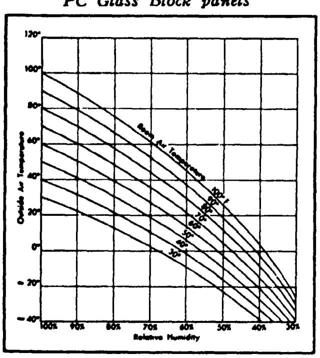
Thermal Insulation

Tests run by nationally recognized laboratories have established the value of glass blocks for insulation of light-transmitting areas. These tests have proved that with glass block panels, heat loss is slightly less than half that experienced with single-glazed windows. In computing heat losses through panels for most design purposes, it is recommended that a "U" value of 0.46 to 0.49 be used for all block sizes and face patterns. For complete data on heat transfer values see the section on heat transfer elsewhere in this Guide—page 108.

Surface Condensation

Due to high insulating value, condensation will not start forming on the room side of glass block panels until outside air has reached a temperature much lower than that necessary to produce condensation on single-glazed windows The accompanying chart shows at what temperatures condensation will form.

Outdoor temperature required to produce condensation on the room side surface of PC Glass Block panels



For example, with inside air at 70°F and relative humidity at 40 per cent, condensation will not begin to form on the interior surfaces of a glass block panel until an outdoor temperature of −14 deg is reached. Under similar conditions with single-glazed sash, moisture will begin to form when the outdoor temperature reaches +33°F.

Solar Heat Gain

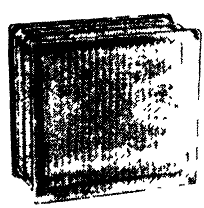
The use of glass blocks for light-transmitting areas results in a marked reduction in total solar heat gain as compared with ordinary windows. This factor is of considerable advantage in buildings that are properly air-conditioned, but does not eliminate the need for adequate ventilation or shading in non-air-conditioned rooms.

For data on solar heat gain through glass blocks see the table in the solar radiation section of this Guide—page 146 This table is for standard pattern glass blocks.

PC Glass Blocks Aid Air-Conditioning

The three chief aims of air-conditioning—temperature control, humidity control and cleansing of air are all aided by the use of PC Glass Blocks. Heat loss is less in winter—heat gain is less in summer. Ideal humidity conditions are much more easily maintained without undue condensation. Solar heat transmission and radiation are reduced. Dirt can't filter in, for each panel is a tightly sealed unit.

Sizes and Shapes Available



PC Glass Blocks are available in eight attractive patterns, some of the patterns being designed for special control and direction of transmitted daylight. For complete information on the sizes and shapes of PC Glass Blocks, and for illustrations of the many patterns available, write the Pittsburgh Corning Corporation, Pittsburgh, Pa., or call the nearest Pittsburgh Plate Glass Company warehouse.

• • •

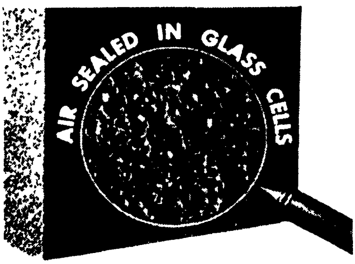
Additional technical data, including detailed figures on thermal insulation, solar heat gain, surface condensation, light transmission and construction data, will be furnished on request.

Pittsburgh Corning Corporation

Room 616, 632 Duquesne Way, **Pittsburgh 22, Pa.**

PC FOAMGLAS—waterproof, fireproof—Insulation

Also manufacturers of PC Glass Blocks



Magnifying Glass

PC FOAMGLAS is different from all other insulating materials, being composed of millions of minute cells of inert air sealed in glass. Thus PC Foamglas embodies not only remarkably efficient insulating qualities, it is also—*vaporproof, fireproof, waterproof and permanent.*

Protecting delicate high-speed machinery and goods in process, insuring continuous, efficient operation—demands rigid control of temperature and humidity in all sorts of mills and factories.

PC Foamglas, the new cellular glass insulation, has proved its ability to meet the most exacting requirements—efficiently and economically—in a wide range of industries

On roofs and in core walls, floors, ductwork and processing equipment, PC Foamglas retains its full insulating efficiency throughout the life of the building. It is an efficient vapor-seal and water-stop, requires no repairs, maintenance or replacement of the material during ordinary use.

Check these features of PC Foamglas against other insulating materials.

1. ***Permanent insulation.***
2. ***Vermin- and vaporproof.***
3. ***Fireproof.***
4. ***Waterproof.***
5. ***Light weight.***
6. ***Easy installation.***
7. ***Rigid structure.***
8. ***Economical.***

On many difficult insulating jobs, such as the roof pictured here, where high temperatures and a wide range of humidities are encountered, PC Foamglas retains its insulating efficiency Prevents condensation Protects the roof slabs permanently, largely because it is an effective vapor barrier, able to withstand 100 per cent relative humidity at high temperatures

When you install PC Foamglas Insulation in core walls, it becomes an integral part of the structure, insures efficient, trouble-free control of temperature and humidity levels in the enclosed area—permanently.

SEND FOR FREE BOOKLETS—To find out how and why PC Foamglas will save time and money on all sorts of heating, ventilating and air-conditioning jobs, send for free copies of our booklets which describe its advantages when used on roofs, in core walls and in floors. 8-07

Alfol Insulation Company

Incorporated

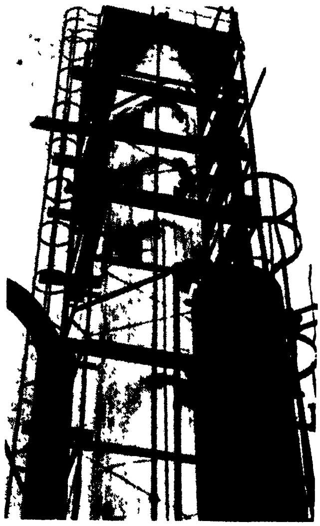
155 East 44th St., **New York 17, N. Y.**

Agents in Principal Cities

HEAT INSULATION for ALL PURPOSES

ALFOL PRE-FABRICATED INSULATION PANELS FOR TANKS, TOWERS, AND ALL TYPES OF HEATED EQUIPMENT

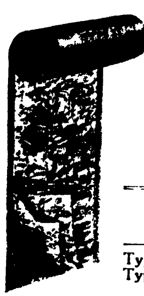
Prefabricated Panel At Right— Applied to Tower



- **Metal Jacketed Panels** containing insulation best suited for each particular condition.
- **Removable and Replaceable** by means of Lock-Joint construction.
- **Shop Fabricated** with Cut-outs for manholes and pipe connections.
- **Easily and Rapidly Applied** by any type of labor.
- **Trim Appearance** with minimum of up-keep.

FOR MORE DETAILED INFORMATION WRITE FOR ALFOL PANEL DATA BOOK

ALFOL BUILDING INSULATION BLANKET



Pure Aluminum Foil spaced on three-ply thick paper vapor barrier sheet. Single and Double Layers insure spaced sheet to reduce conduction and convection. Applied between structural members or furring, Alfol Blankets give high insulation value at low cost.

Specifications

Description	Widths	Net Area per Roll	Net Weight per Roll
Type I.—1 Layer ALFOL	16"-24"	250 sq. ft.	17 lbs
Type II.—2 Layers ALFOL	16"-20"-24"	250 sq ft	19 lbs

See technical data in Table 1, Section C, Page 86, this volume.

ALFOL RADIATOR REFLECTORS

ALFOL REFLECTORS behind radiators reduce heat loss through walls, save fuel. Temperature gradient to outside reduced 50 per cent.

THIRTEEN YEARS' SERVICE PROVES LASTING VALUE OF ALFOL

American Flange & Manufacturing Co., Inc.

30 Rockefeller Plaza, New York 20, N. Y.

Ferro-Therm

Registered Trade Mark Ferro Therm Fully Covered by Patents

STEEL INSULATION

Ferro-Therm Steel Insulation, made from rigid steel sheets with a special alloy coating, reflects 95 per cent of all radiated heat. This high reflectivity, combined with extremely low heat storage capacity, provides maximum insulating efficiency in a minimum overall thickness.

Saves Pay Space and Weight

In building construction, the general practice is to install one sheet of Ferro-Therm between roof rafters or over ceiling joists and between studs of exterior walls. In cold storage construction, the number of sheets of Ferro-Therm depends on the temperature to be maintained and the U value required. The k value of Ferro-Therm, based on tests, is listed in the Data Book of the *American Society of Refrigerating Engineers* as 0.226 Btu per (hr) (sq ft) (°F temperature difference). Laboratory tests and thousands of applications have demonstrated that a wall of Ferro-Therm will provide insulating efficiency equivalent to a wall of mass insulation approximately twice as thick.

Assures Rapid Pull Down of Temperature

The low heat storage capacity of Ferro-Therm is extremely important in achieving rapid pull down of temperature, and in saving refrigeration costs for the initial and each subsequent cooling of space. Specifically, the heat storage capacity of a single sheet of No. 38 gauge is 0.029 Btu per (hr) (sq ft) (°F temperature difference). This is approximately 1/16 of the heat storage capacity of 1 sq ft of 1 in. thickness corkboard.

Permanent, Fire-proof Insulation

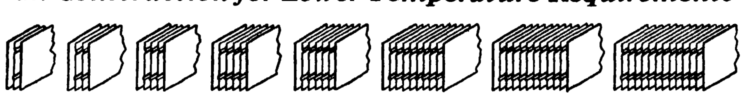
Ferro-Therm construction eliminates trapping of moisture condensate, with subsequent deterioration of the construction. As it is all-metal, Ferro-Therm cannot be penetrated by rodents, vermin or termites, and is absolutely non-combustible. The value of Ferro-Therm for fire protection is apparent.

125° Below Zero Maintained in Altitude Test Chambers

Ferro-Therm has proved its superiority in buildings, cold storage rooms, refrigerated cabinets, locker rooms, dry ice containers, refrigerated railway car construction, ovens, high-temperature storage tanks—in fact, practically every type of application where high insulating efficiency with economies in space and weight are a requisite. The most notable demonstration of Ferro-Therm performance has been its selection for the insulation of altitude chambers for the testing of Army and Navy aviation equipment and personnel. In these chambers, temperatures as low as −125° F were maintained, with a temperature drop of +70° F to −100° F in 10 to 12 min.

Our catalog, giving data and installation details on Ferro-Therm, will be sent upon request

Thinner Wall Construction for Lower Temperature Requirements



Thickness of Ferro-Therm Construction	3 sheets	4 sheets	5 sheets	6 sheets	7 sheets	10 sheets	12 sheets	14 sheets
	1 in.	1½ in	2 in.	2½ in	3 in.	4½ in	5½ in.	6½ in
U Factor.	13	.097	077	064	.055	04	.03	025

The chart above shows the wall thickness and the number of sheets of Ferro-Therm for various U factors. For any particular temperature requirement the installation with the correct number of Ferro-Therm sheets will require only about 50 per cent of the wall thickness required by equivalent mass insulation, resulting in a proportionate saving in storage space, and a great saving in weight.

Armstrong Cork Company

Building Materials Division

Lancaster **Pennsylvania**

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For detailed technical information, samples, and descriptive literature, ask any office or distributor. Specifications appear in Sweet's Catalogs for Architects and for Engineers and Contractors.

PRODUCTS—Armstrong's Corkboard, Cork Covering, Mineral Wool Board, Foamglas, Insulating Refractories, Heat Insulation, Vibracork, Corkoustic, Cushiontone, Temlok, Insulation Sundries.

Corkboard

The thermal conductivity of Armstrong's Corkboard is 0.27 Btu per hour, per degree temperature difference, per inch thickness at 60°F mean temperature. It is furnished in rigid boards 12 in. x 36 in., 18 in x 36 in., and 24 in x 36 in., in several thicknesses: 1 in., 1½ in , 2 in., 3 in , 4 in., and 6 in.

Armstrong's Corkboard conforms in all details to Federal Specification HH-C-561b, July 26, 1941

Cork Covering

Armstrong's Cork Covering is made of pure cork in sizes to fit all standard pipe sizes. The inside surfaces of each piece are machined to assure an accurate fit, free from moisture-catching air pockets. Cork covering is rigid and will not sag. Thicknesses are: Ice Water (1.20 in. to 1.93 in.); Brine (1.70 in. to 3.00 in); and Special Thick Brine (2.63 in. to 4.00 in.).

Armstrong's Fitting Covers are rigid and are designed to fit accurately all types of standard ammonia and extra heavy fittings, screwed, flanged, and welded.

Mineral Wool Board

Armstrong's Mineral Wool Board equals or exceeds Federal Specification HH-M-371 for board or block form insulation; has low thermal conductivity; is moisture resistant, odorless; is easily handled and erected; possesses structural strength. Standard size 12 in. x 36 in ; thicknesses 1 in., 1½ in , 2 in., 3 in , 4 in

Foamglas*

Armstrong's Foamglas has a closed cellular structure which will not permit passage of air or moisture. It is efficient, moistureproof, fireproof, and offers effective, lasting insulation. This new type of insulation is made in standard 12 in. x 18 in. blocks; thicknesses 2 in., 3 in , 4½ in., 6 in. It may be used to insulate refrigerated storage rooms and equipment.

Heat Insulation

The Armstrong Cork Co distributes and offers contract service on a complete line of high temperature insulations. Included are: 85 per cent magnesia blocks and pipe covering, hy-temp blocks and pipe covering; air cell blocks, sheets, and pipe covering, wool felt, hair felt; etc

Engineering and Contract Service

All Armstrong offices and distributors maintain skilled erection crews For aid in the solution of any technical problems involving insulation, isolation, or acoustical treatment, and for literature and prices, get in touch with an Armstrong district office or distributor or the Armstrong Cork Co., Building Materials Division, Lancaster, Pa.

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*Reg U S Patent Office Product Mfg. by Pittsburgh Corning Corp

The Philip Carey Mfg. Company

Manufacturers of Heat Insulation and Asbestos Products

Lockland — Carey EST. 1873 — **Cincinnati, Ohio**

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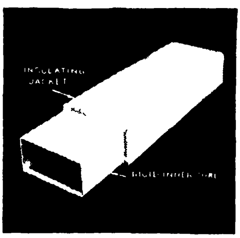
WHEELING, W. VA.

Careyduct AIR CONVEYING SYSTEMS

Before Pearl Harbor, Careyduct had won the widespread approval of the air conditioning industry because of its many advantages. Since then, the urgent need for conservation of metals and labor has added tremendously to the demand for this prefabricated duct.

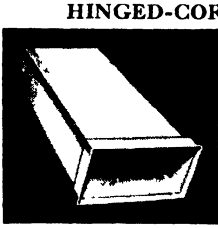
Careyduct is available in the following five different types:

INSULATED AND ACOUSTICAL TYPE



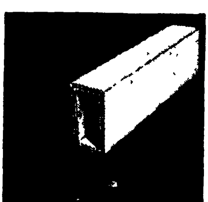
Made of Asbestos—combines both duct and insulation. Simple slip-joint construction and standardized parts provide easy and rapid fitting on job, "hushes" fan noises- reduces "speaking tube" effects.

HINGED-CORNER TYPE



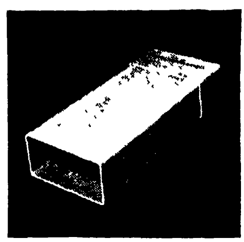
Fabricated of ⅛ in asbestos board and is for use in residential heating jobs. It is delivered in cartons ready for assembly.

KEY-LOCK TYPE



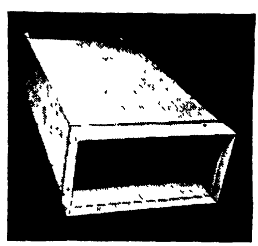
For high temperature applications Impervious to water, and effective for temperatures up to 500 F. This duct is supplementary to Careyduct—Insulation and Acoustical Type. It is made from Carey Firefoil panel and is delivered knocked-down, with asbestos key for locking corners.

SINGLE-WALL TYPE



For use in heating and ventilating systems It is an all asbestos product and is the same as inner core of the Careyduct—Insulated and Acoustical Type.

ASBESTOS-CEMENT TYPE



Made of different thicknesses of asbestos-cement wall board, in sizes from 23½ in. up. It is supplied with special metal corners and is shipped knocked-down, complete with duct turns, sheet metal screws, connector pieces and Careyduct adhesive.

FOR CAREYDUCT MANUAL
WRITE DEPT. 100

The Celotex Corporation

General Offices

120 South LaSalle Street, **Chicago 3**

CELOTEX

REG. U. S. PAT. OFF.

Celotex Cane Fibre Insulation products are made by felting the long, tough fibres of bagasse into strong, rigid boards. They are manufactured under the Ferox Process (patented) which effectively protects them from destruction by termites, fungus growth, and dry rot. They are integrally water-proofed which insures a non-hygroscopic insulation of low capillarity and enduring insulating efficiency.

Celotex Insulating Sheathing

An insulating, weather-resisting sheathing for use under any type of exterior. Surfaces and edges are moisture-proofed with a surface impregnation of asphalt.

Sizes: 25/32 in. thick: 4 ft wide: 8 ft, 9 ft, 10 ft and 12 ft long.

Center Matched—Available in the same thickness, in 2 ft x 8 ft T & G units for horizontal application.

Celotex Insulating Lath

Regular Insulating Lath—A cane fibre plaster base of high insulating efficiency. Surface provides a strong bond for plaster and the bevelled edges and shiplap joint provide additional reinforcement.

Size: 18 in. x 48 in.; thickness: ½ in.

Celotex Roof Insulation

Regular Roof Insulation—A cane fibre product possessing superior insulating properties. It prevents condensation; reduces roof heat transmission as shown by coefficients established in The Guide; reduces roof movement due to contraction and expansion.

Size: 23 in. x 47 in.; thicknesses: ½ in., 1 in., 1½ in. and 2 in.

Vapor-seal Roof Insulation—Same as above except coated on all edges and surfaces with waterproof asphalt and made with an offset on all bottom edges to provide a network of channels which equalize air pressure to reduce roof blisters and buckling.

Size: 23 in. x 47 in.; thicknesses: 1 in., 1½ in. and 2 in.

Cemesto

A completely fabricated fire and moisture resistant structural insulating wall unit. Consists of a Celotex cane fibre core surfaced on both sides with a ⅛ in. layer of asbestos-cement. The established low thermal conductivity of the Celotex core is maintained in the manufacture of Cemesto.

Sizes: 4 ft x 4 ft, 4 ft x 6 ft, 4 ft x 8 ft, 4 ft x 10 ft, 4 ft x 12 ft; thicknesses: 1⅛ in., 1 9/16 in. and 2 in.

Celo-Siding

A weather-resistant, insulating, structural siding. Replaces wood or other sheathing materials and provides the exterior finish as well. Made of a Celotex cane fibre core that has been asphalt coated on all sides and edges. The weather side is additionally coated with a high grade asphalt into which mineral granules are firmly embedded.

Sizes: Thickness, ⅝ in. and ⅞ in.; ⅝ in.—4 ft x 8 ft (square edges only); ⅞ in.—2 ft x 8 ft tongue and groove (long edges only), and 4 ft x 8 ft, 4 ft x 10 ft (square edges only).

Celo-Block

A cane fibre cold storage insulation. Made from ½ in. low density Celotex Cane Fibre Boards laminated with waterproof asphalt mastic. The finished block is surfaced front and back with vapor proofing asphalt. Its efficient low thermal conductivity and protection against moisture makes Celo-Block the ideal answer for all low temperature requirements.

Size: 18 in. x 36 in.; thickness: 2 in.

Celotex Rock Wool Products

Available in the following forms—Loose, Granulated, Plain Batts, and Paper-backed Batts. Celotex Rock Wool is made from the clean fibres of molten rock. It is incombustible and integrally waterproofed.

Q-T Ductliner

An acoustical material designed especially for duct lining in air conditioning systems. Absorbs duct noises. Made of rock wool and a special binder. Designed to withstand air duct humidity conditions. Is fire resistant and will not smoulder or support combustion. Thermal conductivity of 0.30.

The Eagle-Picher Lead Company

General Offices: American Building, **Cincinnati 1, Ohio**

Offices in Principal Cities

EAGLE
PICHER

A Remarkable Insulating Wool Made From Minerals

Years ago Eagle-Picher pioneered a method of fusing and fiberizing carefully selected minerals into a dark gray insulating wool. This mineral wool is chemically inert. Fibers are mechanically strong, extremely resilient and flexible. They withstand expansion and contraction without loss of efficiency even at elevated temperatures.

From this mineral wool, Eagle-Picher has fabricated a long list of insulating products to meet a wide range of temperatures and operating requirements.

Eagle H-2 Loose Wool

A clean fill insulation that is highly efficient for temperatures to 1200° F. Averages considerably lighter in weight than many rock and slag wools—goes farther. Fibers are soft and flexible. Approved by Underwriters Laboratories as fireproof and a non-conductor of electricity. Retains physical and chemical stability in presence of water. Packed in 40-lb. bags.

Eagle 7-B Granulated Wool

Another grade of fill insulation that has all the advantageous properties of Eagle H-2 Loose Wool. It consists of small pellets averaging ⅛ to ½ in. in size. For all fill jobs in irregular spaces. May be poured. Packed in 40-lb. bags.

Eagle Low Temperature Felt

A highly efficient insulating material for subzero and low temperatures (to 400° F). Available in densities 6-lb to 8-lb per cu ft. Recommended for refrigerator rooms, trucks, refrigerators, stoves, etc. Sheds water. Extensively used in marine field.

Paper Encased Batts and Blankets*

These light-weight, sturdily constructed batts and blankets are easy to apply. Enclosed on four sides with paper, one side of which is an approved vapor barrier. Strong tacking flanges. Quickly cut with knife or shears. Three thicknesses—Ful-Thik, Semi-Thik and 1-in. For home use.

*Available as described after war is over.

Eagle Super "66" Cement

A high-temperature plastic insulation. Easy to apply and trowels to a smooth finish. Actively inhibits rust. Will stick on any clean, heated surface. Dry coverage 50-55 sq ft per 100 lb. 100 per cent reclaimable up to 1200° F. Packed in 50-lb bags.

Eagle Supertemp Blocks

An all-purpose high-temperature block insulation which will withstand elevated temperatures up to 1700° F without loss of efficiency or structural strength. Fibers are water-repellent. Light weight. Easily cut to fit irregularly shaped surfaces. Blocks withstand all normal vibration and abrasion encountered in use for which they are recommended. All standard sizes.

Eagle Insulseal

A protective coating for Industrial Insulation Blankets, Supertemp, "66" Cement and other kinds of heat insulation. Provides a permanent seal that safeguards insulation against air infiltration, moisture, water, fumes; also against vibration and abrasion. Does not support combustion.

For more complete specifications and technical data on these and other Eagle Insulating Products, see Sweet's Engineering or Power Plant catalogs.

INSULITE

General Offices
500 Baker Arcade Bldg., **Minneapolis 2, Minnesota**
THIRTY-ONE YEARS PROVEN DURABILITY

For 31 years engineers and architects have specified Insulite materials for structural uses, interior finish, low temperature duct lining, and for other thermal insulation and sound control work. Insulite materials have proved themselves practical through their performance on the job.

STRUCTURAL MATERIALS

Lok-Joint Lath—An insulating plaster base, fabricated from Ins-Lite or from Graylite. Patented "Lok" firmly locks the sheets between supporting members. Thickness: ½ in. Size: 18 x 48 in.

Sealed Graylite Lok-Joint Lath—An insulating plaster base of Graylite, sealed on stud space side with an effective vapor barrier. Has patented "Lok" on long edges. Furnished in same thickness and size as Ins-Lite and Graylite Lok-Joint Lath.

Bildrite Sheathing is an asphalt-containing wood fiber insulating board manufactured under an exclusive process which provides increased strength and moisture resistance. It is 25/32 in. thick and has a gray-brown color. Thermal conductivity: Maximum 0.36 Btu per inch thickness. Each sheet is marked to indicate proper nail spacing. Available in sizes 4 x 8 ft up to 4 x 12 ft with all edges square Also available in 2 x 8 ft size with interlocking joint on long edges. Used as a structural sheathing board and as roof boarding.

Condensation Control—Where low outside temperatures and high inside humidities may occur, authorities recommend "sealing the warm side and venting the cold side" of the wall to prevent condensation. An adequate vapor barrier, Sealed Graylite Lok-Joint Lath, should be used on the warm (room) side of the wall thereby effectively reducing vapor transmission into the stud space. Bildrite Sheathing is designed to allow any surplus vapor in the stud space to "breathe" or be vented to the exterior air. If vapor is trapped within the stud space and cannot escape through the sheathing, destructive condensation may occur.

THE APPROVED INSULITE WALL OF PROTECTION

This construction consists of Bildrite Sheathing on the exterior of the frame work and Sealed Graylite Lok-Joint Lath on the interior Transmission coefficient (U) is shown below

Exterior Finish and Sheathing	Interior Finish
	No Insulation Between Studding
	Plaster (½ in.) on Sealed Graylite Lok-Joint Lath (½ in.)
Wood Siding, 25/32 in. Bildrite Sheathing	0.15

The above value is typical of results which can be obtained by utilizing Insulite materials in frame construction For further (U) values refer to Chapter 4 pages 94 and 95

Applying Bildrite Sheathing

Applying Lok-Joint Lath

INTERIOR FINISH MATERIALS

Ins-Lite Building Board—A wood fiber board with the light color of natural wood—burlap and linen textured surfaces. Thermal conductivity: Nominal 0.33 Btu/hr/sq ft/ in./F; density: 16 lb/cu ft. Furnished in thicknesses of ½ and ¾ inch and sizes of 4 x 7 ft to 4 x 12 ft. Also available in 6 x 8 ft, 6 x 12 ft and 8 x 12 ft sizes.

Graylite Building Board—An integrally treated asphalt containing wood fiber board of grayish brown color—burlap and linen textured surfaces. Thermal conductivity nominal 0.35 Btu per inch thickness. Furnished in same thicknesses and sizes as Ins-Lite Building Board

Smoothcote Interior Board—Factory Coated Insulating Board with smooth, finished surface one side, having 68 per cent light reflection. Furnished in ½ inch thickness only and in sizes of 4 x 7 ft to 4 x 12 ft.

Satincote Interior Board—Factory finished Insulating Board in colors light ivory, oyster white, coral and green. Light reflection from 64 per cent for green to 80 per cent for the buff color Requires no further decoration Resistant to abrasion and washable In ½ inch thickness and in sizes of 4 x 7 ft to 4 x 12 ft.

TileBoard—Available in Smoothcote and Satincote. TileBoard is furnished with the Lok-Grip Joint that permits concealed nailing and which together with the Lok-Pin (a flat diamond shaped metal dowel) definitely and mechanically safeguards against any falling units even though no face nailing is used.

Smoothcote and Satincote TileBoard available in ½ inch thickness and sizes of 12 x 12 inches to 16 x 32 inches.

Plank—Available in Smoothcote and Satincote. Plank has the Lok-Grip joint which permits concealed nailing and is beveled and beaded both long edges. Smoothcote and Satincote Plank furnished in ½ inch thickness, widths of 8 to 16 inches and lengths of 8 to 12 ft.

Acoustilite or Fiberlite effectively quiet and control sound

Acoustilite—A high efficiency acoustical material for sound control. Coefficient of sound absorption, at 512 cycles, is 0.79 when mounted on solid background and 0.80 when on furring strips. Noise reduction coefficient is 0.65 when mounted on solid background and 0.75 when on furring strips. Factory painted in buff, (light reflection 77 per cent) and in white (light reflection 80 per cent). Units have a butt joint and are beveled on four edges. Thickness, ¾ in.; sizes, 12x12 in. to 16x32 in.

Fiberlite—An efficient sound absorptive and decorative material. Coefficient of sound absorption, at 512 cycles, is 0.53 when mounted on a solid background and 0.72 when on furring strips. Noise reduction coefficient is 0.55 when mounted on solid background and 0.65 when on furring strips. Factory painted in buff (light reflection 77 per cent) and in white (light reflection 80 per cent). Units have a butt joint and are beveled on four edges. Thickness, ½ in.; sizes, 12x12 in. to 16x32 in

HardBoard Products

HardBoard materials are tough, durable, grainless, pressed wood fiber boards with a hard, smooth surface. Available in a range of densities from 55 to 68 lb/cu ft. Thicknesses are from ⅒ to 5/16 in. and sizes of 4 x 2 ft to 4 x 12 ft.

Industrial Insulation

Industrial Insulation is a wood fiber board for use in all types of manufacturing industries producing items such as refrigerators, coolers, showcases, brooders, partitions and cabinets.

It can be cut-to-size and fabricated to customer's specifications. Three types of industrial board are available.

Lowdensite Industrial Board—A 10 to 14 lb density board with an average tensile strength of 100 lb/sq in. and an average conductivity of 0.30 Btu/hour /sq ft/F/inch thickness.

Ins-Lite Industrial Board—A 14 to 18 lb density board with an average tensile strength of 250 lb/sq in. and an average conductivity of 0.33 Btu/hour/sq ft/F /inch thickness.

Graylite Industrial Board—Differs from two above products in that it has an integral asphalt treatment which provides increased strength and moisture resistance as well as minimum thickness and linear expansion. A 16 to 20 lb density board with an average tensile strength of 350 lb/sq in. and an average conductivity of 0.35 Btu/hour/sq ft/F/inch thickness

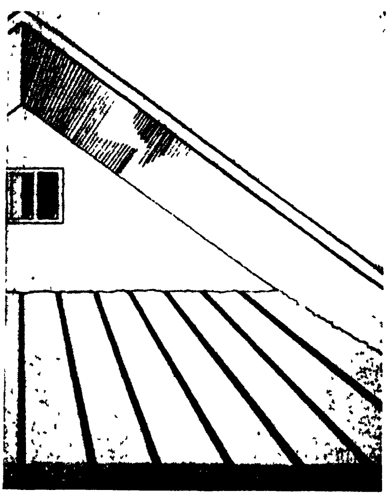
Insul-Wool Insulation Corp.

General Offices, **Wichita, Kansas**

Branches in Principal Cities

Manufacturers and Distributors of Insul-Wool

Insul-Wool is a fibre insulation of the "fill" type—made of wood pulp, a natural insulating material. By the exclusive "Insul-Wool" method the wood pulp is converted into a loose fluffy substance which, when installed in a building, forms a soft heat-resisting blanket having millions of tiny air cells capable of resisting passage of either heat or cold.



Insul-Wool Applied over Ceiling

UNIFORMITY OF PRODUCT

Only one grade of Insul-Wool is made and every "run" is tested at the factory to insure uniformity of product and unvarying high quality. It is free from grit, silicon particles, or "shot."

FIRE PROOF AND VERMIN PROOF

A special "Insul-Wool" method of chemical treatment makes Insul-Wool thoroughly vermin proof and fire proof.

"INSUL-WOOL" SERVICE

Insul-Wool is distributed and installed only by specially trained men—direct factory representatives or men in the organizations of the largest insulation material dealers throughout the United States.

ADVANTAGES OF INSUL-WOOL

1. It is made from wood pulp, a natural insulating material.
2. Chemical treatment makes Insul-Wool safe under all conditions and hazards.
3. Its light weight of 2.5 lb to the cubic foot adds very little load to the ceiling rafters.
4. Does not pack or settle and outlasts the building in which it is installed.
5. Does not draw moisture.
6. Cuts fuel costs and reduces Summer temperatures, indoors.
7. Meets U. S. Government requirements of Federal Construction with a thermal conductivity of 0.24 Btu per hour, per square foot, per degree Fahrenheit, per inch thickness.

Analysis of Insul-Wool in Terms of Commercial Thickness

Material	Commercial Form	Comm'l. Thickness Inches	D. Wt Per Cu. Ft.	C. Conductivity
INSUL-WOOL	Wood Fiber-Loose Type, Fire proofed and Germ proofed.	1 4	2.5	0.24* 0.067**

*Kansas City Testing Laboratory, Inc., February 25, 1938.
**J. C. Peebles, Armour Institute of Technology, April 8, 1937.

Complete data on Insul-Wool Insulating Product will be sent upon request.

Johns-Manville

Executive Offices: 22 East 40th Street, New York 16, N. Y.

Offices in All Large Cities

Johns-Manville Home Insulation

Johns-Manville Rock Wool Home Insulation is a light, fluffy mineral wool, highly efficient in heat-proofing practically any building, old or new. It is durable, rot-proof, fire-proof and odorless, and will not corrode or settle. Full stud thickness of this material will cut fuel costs up to 30 per cent in winter and help keep rooms up to 15 deg cooler in hottest weather. J-M Rock Wool Home Insulation is furnished in two forms: for new construction,

Applying J-M Super-Felt Type B batts in new home

in easily handled batts; for existing buildings, in nodulated form to be installed pneumatically.

For New Construction J-M Super-Felt Type B Batts

Super-Felt Type B Home Insulation is furnished in pre-fabricated batts of uniform thickness and density, in both full stud thickness and semi-thick, in sizes 15 x 23 in. and 15 x 48 in., designed to fill completely the space between studs, joists and rafters on the usual 16 in. centers. The sturdy felted "wool" is strong enough to be handled rapidly without damage. The batts are backed with waterproof, vapor-resistant paper, extending on both the long sides in 1½ in. wide flanges, by which the batt is fastened in place and which also aid in sealing the joints This backing protects against penetration of moisture from wet plaster and also resists infiltration of moisture vapor from the house into the wall.

As a further protection against moisture, the felted wool is also waterproofed

Super-Felt may also be obtained in blanket form, in Thick, Medium and 1 in thicknesses. The blankets have a waterproof vapor barrier paper on one side and a permeable kraft paper on the opposite side, cemented together along the long edges to form a strong nailing flange.

For Existing Homes and Buildings Type A "Blown" Rock Wool

Type A Rock Wool is blown pneumatically into the spaces between studs in outer walls and between rafters or joists in roofs or attic floors Insulation thickness in walls corresponds to stud depth, approximately 3⅝ in., the density, approximately 5 to 8 lb per cu ft, assures maximum thermal efficiency This type of insulation is installed only by Johns-Manville or by Approved J-M Home Insulation Contractors, who are equipped with the necessary apparatus and trained crews.

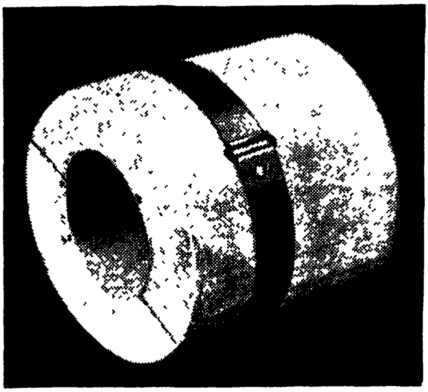
Write for Details

Complete information on all types of J-M Rock Wool Home Insulation will be furnished on request.

J-M Airacoustic Sheets for lining Air-Conditioning Ducts

J-M Airacoustic Sheets, for duct linings of air conditioning systems, are flameproof, highly sound-absorbent and moisture-resistant, with a surface which will not materially increase friction losses in the duct system. Write for Bulletin AC-23A.

Johns-Manville Pipe and Boiler Insulation



J-M 85% Magnesia Pipe Insulation

J-M Pre-Shrunk Asbestocel Pipe Insulation

Cellular type of insulation for pipes carrying low pressure steam or hot water

Made up of alternate layers of plain and corrugated, specially-treated, moisture-resistant, asbestos paper Sections surfaced with asbestos paper for high speed work or with a canvas cover

Furnished in 3-foot sections, in thicknesses of 2 to 8 plies, each ply approximately ¼ inch thick *

J-M 85% Magnesia

Recommended as the most widely used insulation of the molded type for temperatures up to 600 F. Pipe insulation is furnished in sectional or segmental form for all commercial pipe sizes,* in thicknesses up to 3 in. Blocks are 3 in. by 18 in. and 6 in. by 36 in., flat or curved, from 1 in. to 4 in. thick. Minimum thickness for curved blocks, 1¼ in.

J-M Pre-Shrunk Wool Felt Pipe Insulation

Due to its Dual-Service Liner—an asphalt-saturated felt—J-M Pre-Shrunk Wool Felt is equally effective and durable on either hot or cold water service piping. By the use of moisture-resistant felts, shrinkage troubles have been minimized.

Supplied in the regular canvas finish, it is furnished in 3-ft sections in thicknesses of ½ in., ¾ in., 1 in., Double ½ in., and Double ¾ in., for all commercial pipe sizes.*

J-M Asbesto-Sponge Felted Pipe Insulation

Recommended on all high pressure steam piping at temperatures up to 700 F where insulation may be subjected to rough usage or where maximum efficiency and durability are desired. Furnished in 3-ft sections up to 3 in. thick, for all commercial pipe sizes.*

J-M Superex Combination

Superex Combination Insulation (an inner layer of high temperature Superex and an outer layer of 85% Magnesia) is recommended where temperatures exceed 600 F. Superex and Magnesia are both furnished in sectional and segmental pipe covering, and in block forms.

J-M Asbestocel Sheets and Blocks

Asbestocel Sheets and Blocks are used for insulating warm-air ducts, flues, heater casings and fan housings in the ventilating systems. Temperature limit 300 F. Furnished 18 and 36 in. wide by 36 in. long, from ¼ in. to 4 in. thick.

J-M Rock Cork Sheets and Pipe Insulation

J-M Rock Cork is made of mineral wool and a moisture-proof binding ingredient molded into sheets for insulating refrigerated rooms, air conditioning equipment and other low temperature requirements; and into sectional pipe insulation with an integral waterproof jacket, for all low temperature service. It is strong, durable, and will not support vermin. Because of its unusual moisture resistance, its high insulating efficiency is maintained in service.

Furnished in sheets 18 in. by 36 in., in 1, 1½, 2, 3 and 4 in. thicknesses; also 18 in. by 18 in. by 1 in. thick. In lagging form, for curved surfaces, supplied 18 in. long by 1½, 2, 3 and 4 in. thick, 2 to 6 in. wide, depending on diameter. In pipe covering form, in ice water, brine and heavy brine thicknesses, for all commercial pipe sizes.*

Details on Request

Write for complete information on any Johns-Manville insulating material.

*Can also be supplied in sections to fit straight runs of copper pipe or tubing with outside diameter ⅝ in. and larger.

Kimberly-Clark Corporation

Neenah, Wisconsin

KIMSUL
REG. U. S. & CAN. PAT. OFF.
INSULATION

. . . to increase efficiency in Heating, Ventilating, Air Conditioning

A PRODUCT OF
Kimberly Clark
RESEARCH

THERMALLY EFFICIENT!

KIMSUL* blanket insulation in an attic will usually make a house up to 15 degrees cooler in summer, comfortably warm in winter. Fuel cost is often cut 30 per cent by KIMSUL. Thermal efficiency: 0.27 Btu/hr/sq ft/deg F/in. (J. C. Peebles, Armour Institute).

RESISTS FIRE, MOISTURE, VERMIN, INSECTS, FUNGUS!

PERMANENT!

Strong stitching prevents shifting, sagging, settling—Hence, no **transoms** through which heat can leak. Made of wood-fiber, impregnated with asphalt, KIMSUL lasts indefinitely.

ECONOMICAL!

Mass production lowers cost. Because it is compact (stretches to 5 times packaged length), KIMSUL is less expensive to store, ship, handle. Installation cost is small also, because it is exceptionally easy to apply . . can be shaped swiftly around obstacles, as it is flexible as a towel.

3 THICKNESSES—4 STANDARD WIDTHS!

KIMSUL blanket comes in Commercial Thick (nominally ½ in.), Standard Thick (nominally 1 in.) and Double Thick (nominally 2 in.) . . . giving you choice of thicknesses to fit specific needs. Each thickness is available in four standard widths: 16 in , 20 in., 24 in., and 48 in. Also furnished on special orders in wider sizes to fit unusual requirements.

Consult Our Engineers

Architects, engineers, contractors and builders are invited to discuss their insulation problems with our engineers. These insulation experts are equipped to render a valuable advisory service . . and to give accurate information regarding the application of KIMSUL to all types of construction

Send for free KIMSUL book giving full technical data

*Kimsul (trade mark) means Kimberly-Clark insulation

To cut the compressed KIMSUL blanket to length required, simply count off trade marks—placed approximately 24 in. apart when blanket is expanded—allow about 6 in for end fastening, and cut.

To install KIMSUL in walls, partially expand blanket and attach to under-side of top plate with fiber cleat provided. Expand blanket to base of wall and attach similarly. Then staple KIMSUL to sides of studs

Ordinarily a difficult job, insulation of sloping roofs is usually a one man job with KIMSUL. Simply fasten blanket to sides of rafters and collar beams

HOW KIMSUL INCREASES WINTER COMFORT BY RAISING WALL AND CEILING SURFACE TEMPERATURES

		Wall Surface Temperature, Deg	Ceiling Surface Temperature, Deg
No Insulation		60.0	52.0
KIMSUL in Walls and Ceilings	Commercial Thick	64.4	60.0
	Standard Thick	65.3	63.0
	Double Thick	66.5	65.4
	Commercial Thick in Walls, Standard Thick in Ceilings	64.4	63.0
	Commercial Thick in Walls, Double Thick in Ceilings	64.4	65.4
	Standard Thick in Walls, Double Thick in Ceilings	65.3	65.4
KIMSUL in Ceilings only	Commercial Thick	60.0	60.0
	Standard Thick	60.0	63.0
	Double Thick	60.0	65.4

HOW KIMSUL LOWERS FUEL BILLS

		Tons of Coal per Season	Tons Saved per Season	Per Cent of Fuel Saved
No Insulation		12.1		
KIMSUL in Walls and Ceilings	Commercial Thick	8.0	4.1	33.9
	Standard Thick	7.1	5.0	41.3
	Double Thick	6.4	5.7	47.1
	Commercial Thick in Walls, Standard Thick in Ceilings	7.5	4.6	38.0
	Commercial Thick in Walls, Double Thick in Ceilings	7.0	5.1	42.1
	Standard Thick in Walls, Double Thick in Ceilings	6.7	5.4	44.6
KIMSUL in Ceilings only	Commercial Thick	9.1	3.0	24.8
	Standard Thick	8.5	3.6	29.7
	Double Thick	8.1	4.0	33.1

Calculations apply to a one-story home in the Chicago area.

As shown in the table, almost any desired performance can be produced by insulating with KIMSUL. The *economic thickness* provides the best balance between insulation costs and resultant fuel savings.

KIMSUL installs easily in flat roofs and ceilings. After cutting blanket to proper length, expand it and fasten one end to top wall plate Pull across, and fasten other end to opposite wall plate Fasten edges if there is no supporting finishing material

To install KIMSUL in attic floor, place blanket directly on plaster base, with paper side down. Fasten one end, expand blanket, and fasten other end. That's all!

Left-overs and odd pieces of KIMSUL are ideal for caulking spaces around window frames and doorways. Stops drafts and wasteful heat leaks

Mundet Cork Corporation

65 S. Eleventh St. INSULATION DIVISION **Brooklyn 11, N. Y.**

Manufacturers of Corkboard, Cork Pipe Covering, Compressed Machinery Isolation Cork, Natural Cork Isolation Mats, and all kinds and varieties of Cork Specialties.

Authorized contractors for high temperature insulation.

Mundet Branches

Albany 5, N Y	Chicago 21, Ill	Houston 1, Tex	New Orleans 16, La.
Atlanta, Ga	Cincinnati 2, Ohio	Jacksonville 7, Fla	Philadelphia 39, Pa
Brooklyn, N Y	Dallas 1, Tex	Kansas City 7, Mo	St Louis 4, Mo
Boston (No Cambridge) 40, Mass	Detroit 26, Mich	Los Angeles 31, Calif.	San Francisco 7, Calif.

Mundet Distributors are Located in the Following Cities—Names and Addresses on Request

Amana, Ia	Hartford, Conn	Norfolk, Va	Salt Lake City, Utah
Baltimore, Md	Johnson City, Tenn	Oklahoma City, Okla	Seattle, Wash
Buffalo, N Y	Memphis, Tenn	Portland, Oregon	Tucson, Ariz
Denver, Colo	Minneapolis, Minn	Richmond, Va	Tulsa, Okla
El Paso, Tex.	Nashville, Tenn	Rochester, N Y	Utica, N Y

Mundet "Jointite" Corkboard

—for all low temperature insulation and for acoustical correction 100 per cent pure cork, fabricated in accordance with U. S. Government Master Specifications and unsurpassed in its field. Sold in standard 12 in. x 36 in. sheet. Standard thicknesses, ½ in., 1 in , 1½ in , 2 in., 3 in , 4 in., 6 in.

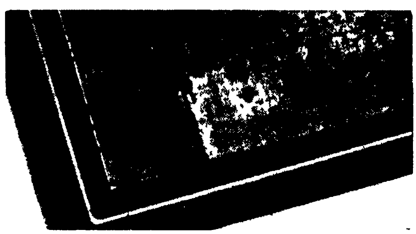
Mundet "Jointite" Cork Pipe Covering

Shown below, with fitting cover. Protects all types of low temperature lines. Made in 3 thicknesses, with complete line of standard covers, suitable for pipes carrying sub-zero to 50 F temperature

Section of Mundet Moulded Cork Pipe Covering, with Fitting The pipe covering is made in sections 36 in long, to fit all sizes of pipes

Mundet Cork Vibration Isolation

Machinery vibration encountered in heating and ventilating work is effectively controlled by the use of Mundet Natural Cork Isolation Mats. These consist of blocks of pure cork, held together within a rigid steel frame or bound with asphalt paper applied with hot asphalt top and bottom. Mundet steel bound mats are usually used under exposed mounts; asphalt paper bound mats under concrete foundations of the envelope type. Mats are made to fit under any type of machine foundation. For loads exceeding 2000 lb per square foot, we manufacture Mundet Machinery Isolation Cork, which is a board form of compressed granulated cork, available in 3 densities All types of isolation are furnished in 1 in., 1½ in., 2 in., 3 in., 4 in., and 6 in. thicknesses, depending on class of service.



Above close-up of Mundet Natural Cork Isolation Mat shows how the blocks of cork are held together within a rigid steel frame

Engineering and Specification Service

Our engineering department is at the service of Architects and Engineers, to assist and advise in the preparation of specifications pertaining to cork. This service is available without obligation to any one who has a low temperature insulation or a vibration isolation problem. Our latest catalogue will be sent on request It is replete with information and data of value to every specification writer whose field touches our products

Mundet Contract Service

Covers the complete installation of our products, in accordance with best established practice. Divided responsibility is avoided. Materials and workmanship are guaranteed.

The Pacific Lumber Company

PALCO WOOL INSULATION

100 Bush Street SAN FRANCISCO	35 E. Wacker Drive CHICAGO	5225 Wilshire Blvd. LOS ANGELES	122 East 42nd St. NEW YORK

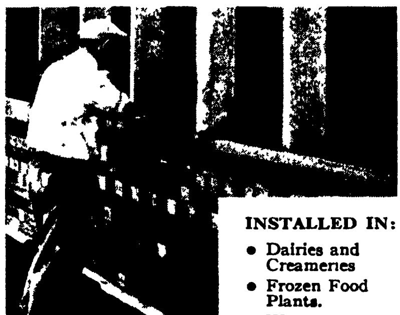
HOUSE INSULATION

INSTALLED IN CEILINGS AND WALLS

PALCO INSULATION WOOL

TRADE MARK REG U S PAT OFFICE

COLD STORAGE INSULATION



INSTALLED IN:

- Dairies and Creameries
- Frozen Food Plants.
- Warehouses
- Refrigerators.
- Pre-cooling Plants
- Ice Plants.
- Fur Vaults.
- Fruit and Produce Storage, etc.

PALCO WOOL INSULATION

Provides maximum protection against the intrusion of heat or cold Readily installed in upper ceilings of uninsulated homes

Can be installed by blower or handpack methods

FLAME PROOF SAFERIZED

EIGHT POINTS OF PALCO WOOL SUPERIORITY

1. **Thermal Efficiency:** The established conductivity of PALCO WOOL is .26 Btu per hour per sq ft per inch of thickness per degree F difference in temperature by the Flat Plate Method.

2. **Non-Settling:** The fibres of PALCO WOOL possess such resilience that no settlement in a wall can occur under the most severe conditions of vibration.

3. **Moisture Resistant:** The fibres of PALCO WOOL are entirely lacking in capillarity and have little attraction for moisture, enabling it to remain dry and efficient when in use.

4. **Permanent:** The inherent antiseptic qualities of PALCO WOOL make the existence of fungus impossible. The fibres retain their resilience indefinitely.

5. **Vermin Repellent:** PALCO WOOL is distasteful and repellent to rodents and insects.

6. **Fire Resistant:** PALCO WOOL, like the Redwood bark it comes from, is inherently fire resistant. As an additional protection it is *Saferized* to make it flameproof.

7. **Odor Proof:** PALCO WOOL is odorless itself and does not absorb or give off odors.

8. **Economical:** PALCO WOOL is light in weight and low in density, offering exceptional thermal efficiency per dollar invested.

WRITE FOR INSULATION MANUALS

- House Insulation Manual.
- Cold Storage Manual.
- Frozen Food Locker Plant Manual.
- How to Build a Plant Manual.

Get your copies today.

Reynolds Metals Company

Reynolds Metals Building

Richmond, Virginia

NEW YORK CHICAGO BOSTON MINNEAPOLIS SAN FRANCISCO

REYNOLDS PROCESS COTTON INSULATION

The Fire Test: Blowtorch Flame on Reynolds Cotton Insulation Will Not Cause Combustion

Reynolds Process Cotton Insulation stops up to 73 per cent of heat flow and permits complete air circulation around framing. Thermal conductivity is 0 24 Btu per hour, per square foot, per 1°F, per inch thickness. (Authority—Prof J C Peebles, Armour Institute of Technology)

Reynolds Process Cotton Insulation is one of the most efficient barriers to the passage of heat that is commercially available today. It consists of heat retarding dead air cells. There are myriads of minute and hollow cellulose fibres, entwined and interlocked into a flexible, clean, resilient and light-weight mass

Reynolds Process Cotton Insulation is not affected by heat or vibration. It will not settle, sag or pack. Age does not impair its lifetime efficiency.

AIR CIRCULATES FREELY

Reynolds Process Cotton Insulation permits free circulation of air on both sides of the insulation, thus allowing rapid evaporation of any moisture which may occur. Possible damp-rot, decay or other damage to structural materials is thereby minimized.

INSECT AND VERMIN REPELLENT

Reynolds Process Cotton Insulation insures utmost cleanliness, not only during installation, but during the lifetime of the structure. This blanket-type insulation does not harbor vermin or other insects. It is odorless, and will not decay.

MOISTURE RESISTANT AND FLAME RETARDENT

Reynolds Process Cotton Insulation will not absorb water or moisture It has successfully withstood flame tests up to 1800°F.

Reynolds Process Cotton Insulation is one of the most effective sound absorption materials.

APPROVED AND ACCEPTED

Reynolds Process Cotton Insulation is manufactured under constant United States Government inspection and in strict accordance with Department of Agriculture specifications. It is approved for home and industrial insulating purposes by Federal, State and Municipal bureaus, builders, architects, and heating engineers throughout the United States.

Reynolds Process Cotton Insulation is ideally suited for equipment insulation. It can be furnished cut to size and in special widths up to 60 in.

COSTS LITTLE TO INSTALL

Furnished in convenient blankets, or rolls, Reynolds Process Cotton Insulation is adaptable to all constructions without expensive cutting or waste. For existing homes, as well as for walls, ceilings or roofs of new structures, it provides maximum insulating efficiency. Labor costs for installation are exceptionally low.

Reynolds Process Cotton Insulation is Easily Installed

Send for literature describing its many advantages in existing houses as well as in new construction.

Reynolds Metals Company

Reynolds Metals Building

Richmond, Virginia

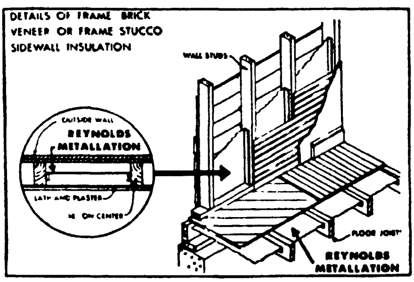
NEW YORK CHICAGO BOSTON MINNEAPOLIS SAN FRANCISCO

REYNOLDS METALLATION

REFLECTIVE ALUMINUM FOIL

Reynolds Metallation is produced by mounting bright aluminum foil to one or both sides of 80-lb tough Kraft paper—or on heavy Sisalkraft when greater tensile strength is desired.

Reynolds Metallation serves as a permanently efficient vapor barrier and reflector of radiant heat. It appreciably reduces heat loss in air-filled areas it faces or divides. It has extremely low heat-storage capacity



HOW IT INSULATES

Heat is transferred by radiation, convection and conduction. Up to 85 per cent of all heat transmitted through air spaces is by radiation. Reynolds Metallation reflects 95 per cent of the radiant heat imposed upon it without regard to direction of heat flow This high reflectivity and resultant low emissivity permits the passage of not more than 5 per cent of radiant heat into adjoining air spaces The above is accomplished through use of Reynolds Metallation, installed to divide the normal space in a hollow wall into two separate reflective air spaces.

CHARACTERISTICS

Reynolds Metallation is so light in weight it places no strain on structural members to which it is affixed. It is flexible vet sufficiently rigid to conform to angles or curves in a structure when not under tension.

Reynolds Metallation is soundproof, vermin and water proof; dust will not impair its insulating value.

The aluminum surface of Reynolds Metallation retains its reflectivity indefinitely under all normal conditions. Recent tests by Prof. Gordon B. Wilkes, Massachusetts Institute of Technology, proved that Reynolds Metallation suffers only negligible loss of its reflective insulating quality after years of exposure.

Table below gives values for the rate of moisture transmission through the more common building materials encountered.

The values given are expressed in terms of Btu/Hr 100 sq ft/Grains per lb difference

Moisture Transmission Values	Latent Heat Btu/Hr/100 Sq Ft/Gr/Lb Difference
Double aluminum foil on paper	0.1
50-lb asphalt sheathing paper	0 45
Coated surface aluminum foil (single)	1 0
2 coats asphalt paint	1.1
2 coats aluminum paint	1.2
Duplex building papers	1.3
Wood per inch thickness	4 0
Concrete—per inch thickness	5.5
Common brick—per inch thickness	6 0
Cinder concrete—per inch thickness	9 0
Plaster—(3/8 in.)	23.0
Fiber Board—(1/2 in)	38.0
Mineral Wool—per inch thickness	40.0

Table prepared by Carrier Corporation, Syracuse, N Y

EASY TO INSTALL

Reynolds Metallation is recommended for all types of building construction involving hollow walls; for use between floors and in roof sections. It is available in rolls of convenient lengths and correct widths for standard stud and rafter spacings.

Reynolds Metallation is easily and quickly installed, thereby adding a low installation cost to a low purchase price and assuring users of maximum efficiency with a minimum investment.

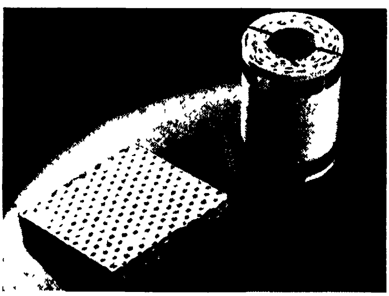
Send for literature describing REYNOLDS *REFLECTIVE ALUMINUM* METALLATION

Ruberoid Calsilite

Calsilite is a molded insulation that provides the physical characteristics required by engineers designing modern high pressure steam equipment. It is a combination of raw materials, ingeniously controlled, to form a chemically inert product. It is not only unique in its extremely light weight for such material, but it will stand soaking heat of 1250 degrees Fahrenheit indefinitely without changing its value to any appreciable degree. This permits its use on pipe lines and equipment operating at high temperature without the usual protective inner layer of less efficient insulation. The use of single layer insulation also speeds up its application.

Calsilite is highly efficient in heat saving; it has a high modulus of rupture; it resists abrasion excellently. There is little breakage of Calsilite in shipment or when handled on the job. It has low moisture absorption when subject to high humidity atmospheres.

Calsilite is furnished in both pipe covering and block form. Pipe Covering is made in sections 36 in. long, canvas covered and in such thicknesses that combinations of layers can provide whatever thickness may be required to cope with the conditions. Blocks are made in standard sizes of 6 in. x 36 in. in thickness up to 3½ in. All are packed in cartons suitable for easy handling on the job.



Imperial Pipe Covering

This is a laminated asbestos paper insulation that has been indented to use 22 laminations of asbestos paper per inch thickness. Its efficiency makes it satisfactory for most medium pressure steam work in industrial plants Its construction makes it ideal for vibrating conditions. It is recommended for temperatures to 600 F. Being an asbestos felt laminated material it is used on vibrating pipes or where hard service is expected. Will withstand water conditions for underground piping Excellent as an industrial, oil refinery and synthetic rubber plant insulation.

Ruberoid Insulating Cements

For the finishing of sheet and block insulation and the insulating of irregular surfaces, such as valves, unions, flanges, etc., the Ruberoid line of insulating cements is complete. This group of cements not only uses asbestos as its base, but also takes advantage of such excellent natural products as magnesia, mineral wool and Vermiculite.

Asbestos Cements—Factory Prepared—Grades AA, A, HF.

Asbestos Cements—Mine Run—Grades 115, 214.

Magnesia Cement—85 per cent Magnesia.

High Temperature Cement—Grade H.T.

Mineral Wool Cement—Good Insulation in plastic form for temperatures to 1500 F.

Vermiculite Cement—Grade A-11.

Ruberoid Asbestos Insulating Papers and Millboard

Asbestos Paper

Made of pure asbestos, fire-resisting. May be obtained in 6, 8, 10, 12, 14, 16 and 32 lb weights. Also thicknesses ⅛ in. and 3/32 in. known as asbestos rollboard.

Asbestos Corrugated Paper

Efficient for insulating warm air pipes and ducts. 36 in. wide. Rolls contain 250 sq ft.

Asbestos Millboard

A rigid board of exceptional strength and whiteness. Cuts and drills easily. Withstands temperatures to 1000 F Sheets 42 x 48 in.

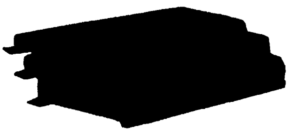
United States Gypsum Company

General Offices: 300 W. Adams Street, **Chicago, Ill.**

INSULATION PRODUCTS

Blanket **Decorative** **Structural**

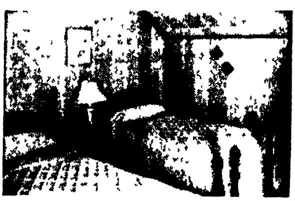
Blanket



Red Top Insulating Blanket
A Fiberglas Product

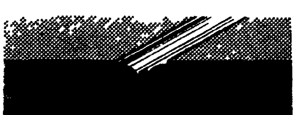
RED TOP INSULATING BLANKETS—Made for standard framing in three thicknesses: one inch, medium and thick, in rolls of 125, 75 and 50 sq ft respectively. Also available in bats 3 ft long in same thicknesses. Light-weight *RED TOP INSULATING WOOL blanket is enclosed with an asphalt-type vapor barrier front side and a vapor permeable paper on the back side. This is to resist accumulation of moisture within blanket.

Decorative



Decorative

WEATHERWOOD* PLANK—Manufactured in widths of 8, 10, 12 and 16 inches and in lengths 6, 8, 10 and 12 feet—½ inch thick. The "Ogee" tongue and groove on the long edges (see cut) conceals nails and seals against dust and air infiltration. Weatherwood Plank is made in Blendtex (gray and tan blends) and Hilite (ivory) colors. When combined in variations of shades and width, Weatherwood Plank produces maximum values in both insulation and decoration.



Ogee Edge

*Red Top Registered Trademark.

WEATHERWOOD TILE—Available in 12 x 12, 12 x 24, 16 x 16 and 16 x 32 inches in ½ and 1 inch thicknesses. Colors are Blendtex (gray and tan blends) and Hilite (ivory).

WEATHERWOOD PANELTILE—Hilite color available in 12 x 24, 16 x 32, 12 x 48, 24 x 48, 48 x 48, 48 x 96 and 24 x 96 inches in ¾ inch thickness. Blendtex colors available in 12 x 24, 16 x 32, 12 x 48 and 24 x 48 inches. Tile sizes 12 x 24, 16 x 32 and 24 x 48 inches can be mill cross scored to represent ½ inch Tile dimensions. Dimensions (see above). All tile have "Ogee" tongue and groove edges.

Structural

WEATHERWOOD SHEATHING—Asphalt coated 2 feet x 8 feet x 25/32 inches thick, with tongue and grooved long edges for horizontal application Also available in 4 x 8, 4 x 9, 4 x 10 and 4 x 12 feet in either ½ inch or 25/32 inches thickness with square edges for vertical application.

WEATHERWOOD BUILDING BOARD—4 feet wide, made in lengths 6, 7, 8, 9, 10 and 12 feet, ½, ¾ and 1 inch thick in either ivory or gray tan colors. Applied by nailing; effectively insulates, strengthens and decorates.

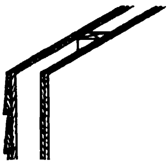
WEATHERWOOD INSULATING LATH—18 x 48 inches x ½ thick with V joint on long edges. Gives an excellent plaster bond and also acts as a cushion for plaster with sound deadening qualities.

ROOF INSULATION—In sheets 22 x 47 inches—½, 1, 1½ and 2 inches thick. All but the ½ inch size are supplied laminated with either square or "ship-lapped" edges.

Heat Loss Factors

The heat loss factors shown on the opposite page indicate the comparative insulation value of various insulating treatments included in common construction systems.

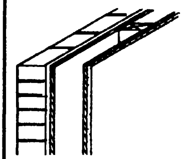
NOTE: These figures apply to 1 story buildings. To get figures for 2 story homes add 20 per cent to the values below for the wall constructions and divide by one-half for floor and ceiling constructions It is important to use correct factor due to variations in the ratio of wall and window areas.



WALLS

Basic Construction—Frame
Wood Siding
Wood Sheathing
2 x 4's
Rocklath & Plaster

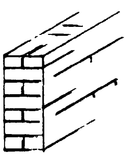
	No Wool Between Studs	Adding Red Top Wool 1"	2"	3"
Basic Construction	248	.120	083	064
Substituting in Above Basic Const.				
a. 25/32" WW Sheathing	186	105	.076	.059
b. 1/2" WW Plaster Base	183	.103	.076	.059
c. 3/4" WW Plaster Base	157	.094	069	055
d 1" WW Plaster Base	143	088	.067	.054
e. 25/32" WW Sheathing and 1/2" WW Plaster Base	147	090	068	.054
f. 1/2" WW Bldg Bd., Tile or Plank	187	104	077	.059
g. 3/4" Bldg Bd., or Tile	160	.095	070	056
h. Gyplap Sheathing	310	135	090	068
i. 1/2" Sheetrock	260	123	085	.064
j Gyplap Sheathing and 1/2" Sheetrock	330	136	091	.069



WALLS

Basic Construction—Brick Veneer
4" Brick
Wood Sheathing
2 x 4's
Rocklath & Plaster

	No Wool Between Studs	Adding Red Top Wool 1"	2"	3"
Basic Construction	.270	.125	.085	.065
Substituting in Above Basic Const.				
a. 25/32" WW Sheathing	.202	111	078	.061
b. 1/2" WW Plaster Base	.200	107	.077	.060
c 3/4" WW Plaster Base	178	.102	.073	.058
d 1" WW Plaster Base	.157	.094	.070	.055
e. 25/32" WW Sheathing and 1/2" WW Plaster Base	.162	.095	.070	.055
f. 1/2" WW Bldg Bd., Tile or Plank	.215	.112	.079	.061
g. 3/4" WW Bldg Bd. or Tile	.187	.104	.075	059
h. Gyplap Sheathing	.350	.128	087	.066
i. 1/2" Sheetrock	288	130	.088	.066
j. Gyplap Sheathing and 1/2" Sheetrock	.368	142	093	.069

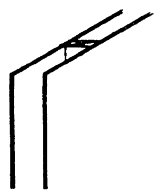


WALLS

Basic Construction
8" Brick Wall—4" Face Brick and 4" Common Brick—No interior finish

	No Wool Between Studs*	Adding Red Top Wool** 1"	2"	3"
Basic Construction	500			
Adding to Above Basic Const.				
a. 1/2" Plaster	.480			
b. Rocklath and Plaster (Furred)	300	143	093	069
c. 1/2" WW Plaster Base and Pl (Furred)	220	121	.084	.064
d. 3/4" WW Plaster Base and Pl. (Furred)	190	112	.079	.061
e. 1" WW Plaster Base and Pl. (Furred)	.160	101	.073	.058
f. 1/2" WW Bldg. Bd., Tile or Plank	230	.124	.085	.065
g. 3/4" WW Bldg. Bd., Tile or Plank	200	.115	.081	.062
h. 1" WW Bd., Plank or Tile	.170	.104	.075	.059
i. 1/2" Sheetrock Furred	.320	.146	.095	.070

***Based on 5/8" Furring Strip **Based on Full Dimension**



WALLS

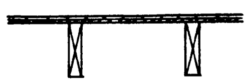
Basic Construction—Plywood
Plywood on Wood Studs
3/8" Outside—1/4" Inside with one Air Space Over 3/4"

	No Wool Between Studs	Adding Red Top Wool 1"	2"	3"
Basic Construction	.431	151	.095	.074
Substituting in Above Basic Const				
a. 1/2" WW Bldg. Bd., Tile or Plank	.275	.126	.087	.065
b. 3/4" WW Bldg. Bd. or Tile	.230	.115	.081	.062
c 1" WW Bldg. Bd. or Tile	.196	.106	.076	.060
d. 3/8" Sheetrock	.430	.151	.095	.074
e. 1/2" Sheetrock	.413	.148	.095	.074
f. Adding to basic construction 25/32" WW Sheathing	.218	.113	.080	.061

CEILINGS

Basic Construction
3/8" Rocklath and 1/2" Plaster

	No Wool Between Joists	Adding Red Top Wool 1"	2"	3"
Basic Construction	.610	169	.116	.080
Substituting in Above Basic Const.				
a. 1/2" WW Plaster Base & Plaster	.329	136	091	.068
b. 3/4" WW Plaster Base & Plaster	.290	.128	.087	.066
c. 1" WW Plaster Base & Plaster	213	.110	.079	.061
d. 1/2" WW Bldg. Bd., Tile or Plank. No Plaster	.356	.139	.092	.067
e. 3/4" WW Bldg. Bd. or Tile	.268	.124	.086	.065
f. 1/2" WW Bldg. Bd. or Tile	.220	.113	.080	.062
g. 3/8" Sheetrock	.670	.174	.118	.089
h. 1/2" Sheetrock	.635	.170	.115	.079



FLOORS

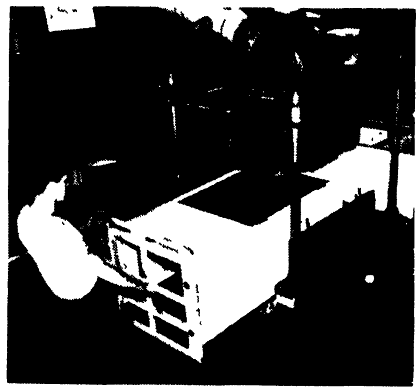
Basic Construction
Maple or Oak Flooring on Yellow Pine Sub-Flooring

	No Wool Between Joists	Adding Red Top Wool 1"	2"	3"
Basic Construction	.340	.138	.091	.068
Adding to Above Basic Const.				
a. 1/2" WW Bd. on bottom of joists	.180	.102	.075	.059
b. 3/4" WW Bd. on bottom of joists	.158	.094	.070	.055
c. 1" WW Bd. on bottom of joists	.141	.088	.066	.053

Above calculations based on data from A.S.H.V.E. GUIDE—1942.

BALSAM-WOOL PATENTED PNEU-PRO FIBRE INSULATES CABINETS AND DOORS IN SECONDS

In one, fast, streamlined operation requiring small space, refrigerator doors and cabinets can now be insulated in less than 60 sec Such speed and simplicity are made possible by the Balsam-Wool pneumatic system which uses patented *Pneu-Pro Fibre.* Especially designed for blowing in under pressure, this new fibre is asphalt treated and has an amazingly low "K" factor. The pressure method of application fills every void—no uninsulated areas, no paper liners, no joints or fillers are required. *Pneu-Pro Fibre* is shipped in units easily handled by one man. One type of fibre will insulate any size unit, eliminating multiple inventories Complete information will be sent on request.



BALSAM-WOOL K-25 LOCKER PLANT AND COLD STORAGE INSULATING FIBRE

Balsam-Wool K-25 Fibre is scientifically designed to fill the need for a highly efficient, fill-type insulation for use in locker and cold storage plants. This fibre has remarkably low conductivity—cuts down operating and maintenance costs. Controlled fibre length assures high resilience to eliminate packing or settling—offers uniform density and better felting. The fibre is asphalt coated and is especially treated to resist fungi, rot and moisture Easy to install, clean to handle, *K-25 Insulating Fibre* is remarkably low in cost per cubic foot in the wall Assures efficient insulation throughout the life of the locker or cold storage plant For complete information, write us.

PLASTIC FILLER FIBRE

A group of special fibres are now available as plastic fillers. Designed to fill special needs of the plastics industry, these fibres not only act as fillers, but add tensile strength as well.

American Society of Refrigerating Engineers

50 West 40th Street, **New York 18, N. Y.**

REFRIGERATING ENGINEERING

The most rapidly growing magazine in the refrigeration field

REFRIGERATING ENGINEERING

REFRIGERATION and air conditioning have been found even more necessary during wartime than in times of peace—every man engaged in these industries knows of the rapid developments made during the past year.

Long acknowledged the most authoritative periodical in the field, *Refrigerating Engineering* has added steadily to the practical value of its contents, and its number of readers has grown in proportion. A wide variety of material is presented, all from the viewpoint of its usefulness to the reader in his own business. This magazine is a must for men who keep in touch with all that is new and important in refrigeration and air conditioning.

THE REFRIGERATING DATA BOOK

THE REFRIGERATING DATA BOOK is now an essential tool in the refrigeration and air conditioning industries. Editions have been published in 1932, 1934, 1936 and 1938. The 1940 Edition (Volume II) is *entirely different* from any preceding volume. It consists wholly of practical, how-it-is-done chapters on all the known applications of air conditioning and refrigeration. This ***Applications Edition*** carries information of a scientific and popular nature to the scores of industries using refrigeration processes.

The 1942 (Fifth Edition) of the Data Book, replacing Volume 1, has been rewritten in line with 1942 practice, and many valuable new sections have been added to make the Data Book the most comprehensive authority available on refrigeration problems.

APPLICATION DATA BULLETINS

SOME 34 bulletins are available separately at reasonable prices for single copies or quantity orders and can also be had bound with a paper cover, the complete set for $5.25.

The APPLICATION DATA Bulletins tell precisely how refrigeration is used in various fields, giving examples and specific information on the best practice up to date. Some of the subjects covered to date are: refrigeration of locker plants, of fur storage, of restaurants, of liquids, of apples and pears, humidity in refrigeration, refrigeration service charts, refrigeration for skating rinks, butter and cheese making, milk plants, citrus fruits, beer dispensing, retail stores, wine making, load calculations, operation of ammonia machines, how to figure air conditioning, refrigeration of ships' stores, etc.

CODES AND STANDARDS

THE A.S.R.E. further contributes to refrigeration progress by its participation in establishing codes and standards in the industry. Among the recent codes made available are: No. 21—Testing and Rating Milk Coolers; No. 22—Rating and Testing Water-cooled Refrigerant Condensers (Tentative—1942); No. 23—Rating and Testing Refrigerant Compressors (Tentative—1942); No. 24—Rating and Testing Water and Brine Coolers (Tentative—1942); No. 25—Rating and Testing Forced-circulation Air Coolers for Commercial and Industrial Refrign. (Tentative—1942), (Supplement to Cir. No. 13, not sold separately.)

MEMBERSHIP ACTIVITIES

IT is the policy of the A.S.R.E. to treat in its meetings current subjects touching upon all phases of the art of refrigeration. Membership is in two grades with dues from $10.00 to $15.00. Sections hold meetings in the following cities: Baltimore-Washington, Boston, Central N. Y., Chicago, Cincinnati, Cleveland, Detroit, Kansas City, Los Angeles, Milwaukee, New Orleans, New York, Northern New Jersey, Pacific Northwest, Philadelphia, Pittsburgh, Richmond, San Francisco, St. Louis, Twin Cities.

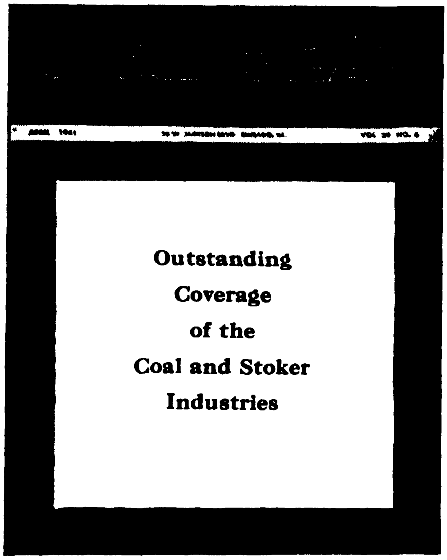
To keep apace with progress in refrigeration and air conditioning, read the publications and follow the activities of THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS, 50 West 40th St., New York 18, N. Y.

Coal-Heat

Published at

20 W. Jackson Blvd., **Chicago 4, Illinois**

Phone Wabash 9464



FOR information on the use and sale of stokers, coal and coal heating equipment, you can turn to COAL-HEAT with complete confidence.

Here is a magazine that appeals to every man concerned with the market, use and sale of solid fuel and modern heating equipment. Having long since recognized the importance of properly designed efficiently operated, properly maintained equipment to the successful use of coal, and therefore to the welfare of the coal industry, COAL-HEAT constantly emphasizes the significance of the "equipment factor" in heating merchandising.

It is only natural that COAL-HEAT was the first trade magazine to recognize and promote the small stoker; to introduce many new developments in coal-burning equipment to further fuel conservation; to support the use of dustless treatment; and to urge the sale of equipment by coal men.

COAL-HEAT has at its disposal an almost unlimited number of sources of authentic information on the topics it covers; its articles are written by the best informed men in the coal, stoker and heating industries. It enjoys quite a following, not only among the most progressive merchants in these industries, but among the industry's leading combustion and heating engineers. For years it has championed the importance of the fuel engineer to the coal and stoker industries, and each year prints many articles *for* and *by* fuel engineers.

COAL-HEAT's fundamental editorial policy is "to further the more satisfactory use and sale of coal and modern coal-burning equipment." It actively supports the application of scientific and engineering knowledge to the use of coal and coal-burning equipment. It covers both the merchandising and utilization of the coal, stokers and modern heating equipment.

With over a million stokers in use today, the importance of COAL-HEAT's field is clearly evident. It has been and is COAL-HEAT's job to supply coal and stoker men with the information they need to insure satisfaction for stoker users. The same is true with hand-fired heating plants and all kinds of household and commercial coal heating equipment.

In addition to providing its readers with a basic and diversified editorial program, COAL-HEAT also publishes a number of books and booklets, manuals and reprints covering a wide range of subjects of interest to coal, stoker and heating men. Its series of heating guides for the consumer have proved particularly popular. These are available at small cost.

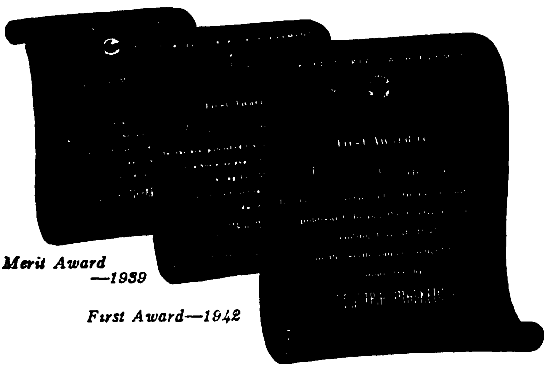
Subscription rates—$2 00 a year; $3.00 for two years. Rates apply for both United States and Canada. Foreign rates —$2.00 a year; $4.00 for three years.

Advertising rates and other information will be furnished upon request.

Domestic Engineering Publications

1900 Prairie Avenue, **Chicago 16, Illinois**

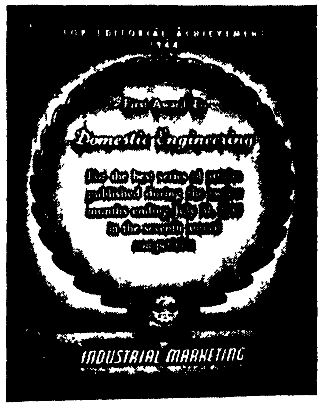
4 *times a winner*



Merit Award —1939

First Award—1942

First Award—1943

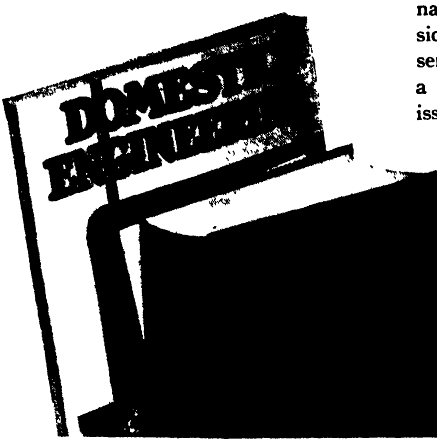


First Award—1944

DOMESTIC ENGINEERING

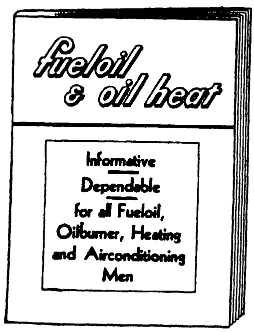
FOR the fourth time, DOMESTIC ENGINEERING has achieved outstanding success and recognition for its service to the nation and to the industry it represents. The high honors conferred upon it are offered as a sign of an alert organization, keyed to the requirements of the times and composed of determined men and women engaged in the task of serving great industries.

The large plaque shown above and received by DOMESTIC ENGINEERING is for excellence in the Class 1 Division of a recent nation-wide competition. This class one division was for entries competing for the best series of articles or editorials on one theme on a definite objective appearing in a series of issues.



DOMESTIC ENGINEERING CATALOG DIRECTORY . . . is a complete, centralized source of product information in the Plumbing, Heating and Air Conditioning industry. Basic circulation covers the buyers and specifiers in this field. Additional coverage for the duration includes Navy Yards, Ordnance Plants, Ship Yards, Industrial Plants, etc.

Fueloil & Oil Heat



A. E. COBURN, *Editor*

ROBERT GRAY, *Business Manager*

ARTHUR G. WINKLER, *Advertising Manager*

OILHEATING is a vertical, integrated industry, selling through the same prime outlets; oilburning equipment (including boiler-burner and furnace-burner units), heating and air conditioning, and fueloil.

The integration of oilheating equipment and fuel has grown until in 1944, exactly 82.3 per cent of all oilheating equipment was sold by companies also selling fueloil, while 54.2 per cent of all fueloil was sold by companies also selling burners.

This publication's organization dates back to 1922. It is alone in the field and is read by all elements—manufacturers and dealers.

MANUFACTURERS

Oilheating equipment manufacturers, with few exceptions, are presently devoting most of their facilities to the production of war products. In some instances these are their peacetime products, being sold for war uses . . . others are concentrating on unrelated lines. All of these companies, however, are maintaining dealer and product relationships through the sale of repair and replacement oilheating necessities.

The industry produced 49,915 oilheating jobs in 1944, under wartime restrictions—38,280 replacements and 11,635 to new users.

DEALERS

Mortality among oilheating dealers has been severe under the handicaps of war, though the larger dealers, with strong service department and fueloil departments, have not been seriously hurt. In January, 1945, the dealers total 10,000 compared with 16,000 before Pearl Harbor. Of these about 7200 have income from fueloil.

OILHEATING TOMORROW

According to our projections of post-war activities, there will be a 100 per cent expansion of oilheating markets during the first 5 post-war years, with sales and installations of 2½ million oilheating jobs in all types and sizes of heating plants. They will be divided: 1,544,000 conversions from coal; 561,000 in new homes, and 388,000 replacements of existing jobs

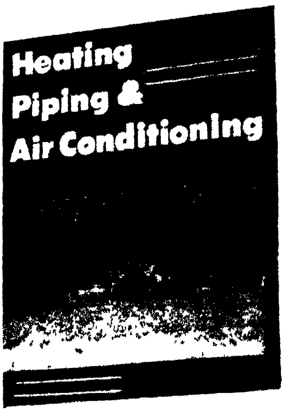
The equipment will be divided into: 1,443,000 conversion jobs, 406,000 boiler-burner units, 644,000 furnace-burner units.

The factors aiding this expansion include. (1) large amounts of fueloil on the civilian market when the military forces stop their present wartime 40 per cent of all petroleum products, (2) there will be 25 per cent to 40 per cent fewer cars on the road for some years after the war, to consume petroleum products; (3) possible lower prices for fueloil, (4) large home-building program; (5) public preference for oilheat—varying from 35 per cent to 55 per cent, depending on area.

For a statistical picture of all phases of oilheating—wartime and postwar, send $1.00 for the January, 1945, Yearbook Issue and for a free copy of "Oilheating Tomorrow"—an 8-page statistical study.

KEENEY PUBLISHING COMPANY
6 North Michigan Avenue, **Chicago 2, Ill.**

Heating, Piping and Air Conditioning



This is the publication which carries the Journal of the A.S.H.V.E. in addition to its own regular editorial section. Its field is that of industry and large buildings. It is devoted to the design, installation, operation, and maintenance of heating, piping and air conditioning systems in such plants and buildings.

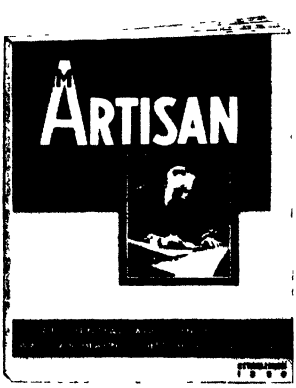
Each January issue includes a complete directory of commercial and industrial heating, piping, and air conditioning equipment, which lists all products, their trade names, and the manufacturers' addresses. It is the established buying and specifying guide of the industry.

H. P. & A. C. is read by consulting engineers and architects . . . contractors . . . and engineers in charge of heating, piping and air conditioning in industrial plants, and other large buildings, federal, state, and city governments, school boards, and public utilities. All A.S.H.V.E. members are subscribers.

Such coverage means, for the advertiser, consideration at all points in the selling of a heating, piping, or air conditioning product . . . consideration in its selection during the preparation of plans and specifications; in its actual purchase for installation; in its year-'round buying for operating and maintenance requirements. Without waste, the manufacturer of air conditioning products and equipment can reach through H. P. & A. C. those from whom he is seeking the necessary engineering acceptance. Write for our new booklet "A Quick Picture."

Subscription Price—No. and So. America—$2 per year. Elsewhere $4 per year. Member—A.B.P.—A.B.C.

American Artisan



AMERICAN ARTISAN covers the field of warm air heating, residential air conditioning, and sheet metal contracting. Its readers are warm air heating and sheet metal contractors, dealers, jobbers, manufacturers, and public utility companies.

Each January issue includes a complete directory of warm air heating, air conditioning, and sheet metal products and equipment, which lists all products, their trade names, and the manufacturers' addresses.

A special section of each issue has been devoted to air conditioning since 1932, when it first became apparent that air conditioning for homes was to be along the lines of the central forced warm air heating system. As a result of the ready adaptability of this type of heating system to all air conditioning factors, tens of thousands of homes today have winter air conditioning —supplied through forced warm air heating with air cleaning and humidification. Cooling apparatus can be attached to these systems readily whenever year-'round air conditioning is desired.

The key man in the residential air conditioning picture is the warm air heating and sheet metal contractor—the one man experienced in "treating air" at a central place and getting it properly distributed. And the key publication—because it reaches these key men with an information service that has made it the recognized authority on residential air conditioning practice—is AMERICAN ARTISAN. For full information on this field, write for a copy of "The Residential Heating Market."

Subscription rate—$2.00 per year. Member A.B.P.—A.B.C.

HEATING AND VENTILATING is edited for engineers, contractors, and equipment manufacturers who have the final word in the specification, installation, and operation of mechanical equipment for heating, air conditioning, ventilating, piping and refrigeration for industrial, commercial, large residential, and institutional buildings.

The editorial content is designed to be of practical use to engineers engaged in this work, and is prepared under the direction of field-experienced professional engineers. A maximum amount of space is given each month to articles showing how specific problems have been met, authoritative discussion of timely subjects, compilations of useful data, and descriptions of the latest practice, techniques and equipment.

Generally speaking, the emphasis is on practical rather than on theoretical considerations.

Each month an original Reference Data sheet is included for permanent use in a standard binder (back copies are available).

Special sections are published from time to time. These sections are devoted to subjects of timely interest, such as Radiant Heating, Dehydration, Fuel Conservation, Air Conditioning, etc. A comprehensive Buyers Directory is published early each fall

CIRCULATION

HEATING AND VENTILATING'S total distribution (May, 1944)—11,102, classified as follows:

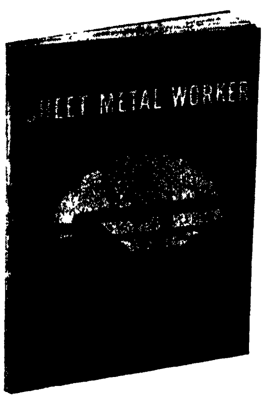
Consulting Engineers (228) and Architects (123) Engineers Employed by them (295)	646
Contractors (1,267) and Engineers Employed by Contractors (235)	1,502
Governments and School Boards, and their Engineers	1,177
Public Utility Group	560
Industrial Firms, their Executives, Engineers and other Employees	2,561
Buildings, Real Estate Management Companies, Their Engineers	782
Manufacturers of Air Conditioning, Heating, Piping and Ventilating Equipment, Their Officials and Employees (766) and Designing Engineers (178)	944
Manufacturers' Agents and Sales-Engineering Firms (162) Sales Engineers and Salesmen (731)	893
Wholesalers (85) and Dealers (364)	449
Educational Institutions, Libraries, Technical Associations	464
Miscellaneous and Unclassified	708
	10,686
Field Staff, Correspondents, Exchanges and Advertising Agencies	416
TOTAL	11,102

Subscriptions to HEATING AND VENTILATING, 148 Lafayette St., New York 13, are $2.00 a year.

Sheet Metal Worker

Published by Edwin A. Scott Publishing Company

45 West 45th Street, **New York 19, N. Y.**



Subscription rates—$2.00 per year, U. S., Canada and Pan Amer. Foreign $3.00. Advertising rates on request.

THE January 1945 issue of SHEET METAL WORKER was its 71st Anniversary and Directory Number. It is the oldest publication in its field and is of vital importance to men interested in sheet metal work—air conditioning—warm-air heating and ventilation. Founded in 1874 and published to 1909 by David Williams Company; 1909 to 1920 by United Publishers Corp.; since 1920 by, the Edwin A. Scott Publishing Co.

Subscribers are mainly merchandising contractors purchasing practically all products and equipment which they fabricate, erect or install. Manufacturers, jobbers and distributors also subscribe.

The market has three main divisions:

(1) Equipment for resale in connection with erection or installation work
(2) Materials for fabrication
(3) Shop equipment and supplies

Circulation: SHEET METAL WORKER is a member of the Audit Bureau of Circulations and the Associated Business Papers.

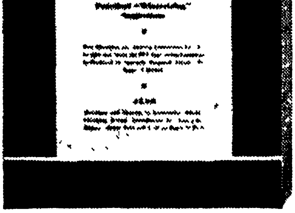
SHEET METAL WORKER also publishes books on heating, ventilating, sheet metal work, air conditioning, etc.

The Annual Issue published in January, contains a comprehensive and valuable Directory Section.

Plumbing and Heating Journal

Edwin A. Scott, Publisher

45 West 45th Street, **New York 19, N. Y.**



Subscription rates—$2.00 per year, U. S., Canada and Pan Amer. Foreign $3.00. Advertising rates on request.

PLUMBING and Heating Journal is edited to furnish a well-rounded, efficient service to the men engaged in the plumbing, heating, ventilating and air conditioning fields. It covers both the technical and business phases of their work.

It gives free technical service through a staff of practical engineers; expert merchandising assistance, and its technical and business articles are by men of recognized competence.

THE JOURNAL editorial department draws its news from scores of trained correspondents located at strategic points throughout the country.

This combination of the technical, business, news and other aspects of the industry enables THE JOURNAL to achieve a finely balanced magazine that gives the reader the type of information he wants and needs, in brief, compact form.

A department "With the Water Systems," informs the trade of the latest developments in the rural plumbing field and its increasing potentialities for the plumbing—heating contractor, especially with the recent extensions of rural electric lines throughout the country.

International Heating & Ventilating Exposition

THE AIR CONDITIONING EXPOSITION

Permanent Address—Grand Central Palace, New York 17, N. Y.

EXPOSITIONS HELD

The first in Philadelphia, 1930.
The second in Cleveland, 1932.
The third in New York, 1934.
The fourth in Chicago, 1936.
The fifth in New York, 1938.
The sixth in Cleveland, 1940.

POST-WAR EXPOSITION PLANNED

While the Seventh Exposition, planned for Philadelphia, in 1942, was postponed due to war activities, plans are now being developed for the first post-war exposition. Although no definite date has been established, manufacturers have indicated a keen interest in exhibiting their newest products of research and development at the earliest practical time Resumption of this Exposition will undoubtedly be a substantial factor in the re-establishment of the heating, ventilating and air conditioning industries in the civilian market

There has developed a tremendous pent-up demand for new equipment that will be a challenge to the industry The manner in which you meet this opportunity will be a guide to your future progress. As in the past, the Exposition will be ready to help you by offering a timely, quick and effective means of renewing old contacts, developing new ones, and actually demonstrating your products

These expositions have been and will be held coincident with the Annual meetings of the American Society of Heating and Ventilating Engineers and are directed by the International Exposition Company, under the auspices of the A S H V E.

EXHIBITORS

Comprise leading firms in each phase of the industry; number has varied from 150 to 327 exhibitors.

EXHIBITS

These range from and comprise all the types of articles discussed or advertised in this copy of The A.S.H.V.E. Guide.

1. *The* Combustion *Group:*
   Furnaces, burners (coal, oil and gas), grates, stokers, boilers, radiators (various types), refractories and auxiliaries.
2. *The* Oil Burner *Group.*
3. *The* Hydraulic *Group:*
   Water feeders, water heaters, pumps, traps, valves, piping, fittings, expansion joints, pipe hangers, etc.
4. *The* Steam Heating *Group:*
   Vapor heating and steam specialties.
5. *The* Hot Water Heating *Group.*
6. *The* Air *Group:*
   Warm Air furnaces and stoves, registers and grilles, cooling towers, air filters, motors, fans, blowers, conditioning equipment, ventilators (room and industrial types), unit heaters, etc.
7. *The* Air Conditioning *Group:*
   Equipment which circulates and filters the air, in summer dehumidifies and cools; in winter heats and humidifies, and does all these in proper season for complete, all year-round air conditioning.
8. *The* Control *Group:*
   Instruments of precision for indicating, controlling or recording temperature, pressure, volume, time, flow, draft or any other function to be measured.
9. *The* Refrigerating *Group:*
   Compressors, condensers, cooling apparatus, contingent apparatus and refrigerants.
10. *The* Central Heating *Group:*
    Apparatus and materials especially designed or adapted to the uses of central heating and central heating station supplies
11. *The* Insulating *Group:*
    Structural insulators (refractory and cellulose materials), asbestos, magnesia clays and combinations thereof, pipe and conduit covering, etc., weather-stripping, etc.
12. *The* Miscellaneous *Group·*
    Electric Heaters, boiler and pipe repair alloys, liquids and compounds, tools of all kinds, and equipment not specifically included in the above groups, but related thereto
13. *The* Machinery and General Equipment *Group.*
14. Books and Publications

VISITOR ATTENDANCE

Attendance is by invitation and registration only, thereby presenting a selected audience Included are contractors, dealers, jobbers, supply houses, home owners, industrial users, professional and service organizations, public utilities, real estate management concerns, etc A detailed analysis or registered attendance is available on request. In 1940 at Cleveland, there were 20,652 registered visitors.

Industrial Expositions in America lead the expositions of the world in style, business effectiveness, industrial influence and educational value. This Exposition stands among the leaders in Industrial Expositions in America. It is an educational institution which brings together the research developments and improvements in equipment and materials for use in heating, ventilating and air conditioning all types of buildings.

1945
Roll of Membership

Corrected to January 15, 1945

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

Published at the Headquarters of the Society
51 Madison Avenue, New York 10, N. Y.

Contains List of Members Arranged Alphabetically with Mail Addresses Only

• • •

WARTIME EDITION

With the membership at an all-time high, and the need for paper conservation vital, under W.P.B. Order L 245, it was the decision of the Council that members' names and mail addresses *only* would be included in the Roll of Membership for 1945, and the geographical listings would be omitted.

It was realized that this might cause minor inconvenience to some members who used the geographical listings, but it is sincerely hoped that it will be possible to continue full listings in future issues.

SUMMARY OF MEMBERSHIP

Honorary Members	2	Associate Members	1374
Charter Members	2	Junior Members	246
Life Members	102	Student Members	25
Members	2178	Total	3929

Officers and Council

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

51 Madison Ave., New York 10, N. Y.

1944-45

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Second Vice-President	ALFRED J. OFFNER
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Guide Publication: J. F. Collins, Jr., *Chairman;* W. C. Bevington, C. S. Leopold, T. F. Rockwell, G. H. Tuttle.

Publication: J. H. Walker, *Chairman* (*one year*); L. E. Seeley (*two years*); A. P. Kratz (*three years*).

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Western New York	S M Quackenbush	S W. Strouse
Wisconsin	A S. Krenz	M. W. Bishop

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A C. Fieldner, *Ex-Officio*

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1945

(Corrected to January 15, 1945)

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BODINGER, J. H. (*M* 1931) Pres, J H Bodinger Co, Inc, 530 Tenth Ave., New York 18, N Y
BODMER, Emmanuel (*M* 1937) Lt., Air Force (Present address unknown).
BODWELL, George B. (*M* 1944) Contract Mgr, The Philip Carey Manufacturing Co, 5906 Euclid Ave, Cleveland, Ohio
BOEHLER, Carl L. (*A* 1944) Sales Mgr, Htg and Refrig Depts, J. A Walsh & Co, Inc, 3215 McKinney, P O Box 1773, Houston, Texas
BOELTER, L. M. K.* (*M* 1944) Assoc Dean, Dept. of Engineering, University of California, Los Angeles 24, Calif
BOESTER, Carl F.* (*M* 1939; *A* 1936) Housing Research Executive, Purdue Research Foundation, Purdue University, Lafayette, Ind
BOGATY, Hermann S. (*M* 1921) 735 E. Phil-Ellena St., Philadelphia, Pa.
BOGGS, Dennis (*M* 1943) 1354 West Boulevard, Cleveland, Ohio.
BOLAND, L. C., Jr. (*M* 1941) 1140 Rosedale Drive N.E., Atlanta, Ga.
BOLAND, Roy O. (*A* 1938) Industrial Steel & Fibre Products, Ltd, Room 1206, 507 Place D'Armes, Montreal, Que, Canada.
BOLES, Harry S. (*A* 1944) 903 E. Maple Rd, Indianapolis, Ind
BOND, Harry H. (*M* 1938) Partner, Edward E Ashley, Cons Engrs., 10 East 40th St, New York 16, N Y.
BOND, Horace A. (*M* 1930) 12 Ramsey Pl, Albany, N. Y.
BONNER, John (*M* 1943) Htg Engr, The John Bonner Co, 121 Hanover St, Boston 13, Mass.
BOOKER, Jack W. (*M* 1943) 505 Nancy St, Charleston, W Va.
BOONE, Fred S. (*A* 1943) Secy.-Treas, Hall-Neal Furnace Co., 1324 N. Capitol Ave., Indianapolis 7, Ind.
BOOT, Arthur (*M* 1943) Owner, Arthur Boot Co, 1444 Lake Drive S E., Grand Rapids 6, Mich.
BOOTH, Charles A. (*M* 1917) Vice-Pres., Buffalo Forge Co., 490 Broadway, Buffalo, N. Y.
BOOTH, Clifford A. (*A* 1942) Sales Engr., Fiberglas Canada, Ltd., 1025 Confederation Bldg., Montreal, Que., Canada.
BORAK, Eugene (*M* 1937) 2237 Pennsylvania, Detroit, Mich.
BORG, Elmer H. (*M* 1938) Partner, Brooks & Borg, 820 Hubbell Bldg., Des Moines 9, Iowa.
BORKAT, Philip (*A* 1943; *J* 1936) 869 East 128th St., Cleveland 8, Ohio.
BORNEMANN, Walter A. (*M* 1924; *J* 1923) Carrier Corp, 12 South 12th St., Philadelphia 7, Pa.
BORNQUIST, George W. (*A* 1943) Pres., Bornquist, Inc., 629 W. Washington Blvd., Chicago 6, Ill.
BORNSTEIN, Alfred B. (*A* 1942) 7417 Piney Branch Rd., Takoma Park 12, Md.
BORNSTEIN, William (*M* 1943; *A* 1937) Partner, Wm. Bornstein & Son, 2209 Channing St. N.E, Washington, D. C
BORTON, A. Robert (*A* 1943; *J* 1939) Engrg. Dept, John J Nesbitt, Inc, State Road and Rhawn St, Philadelphia, Pa
BOTELHO, Nanto J. (*A* 1937) Chief Engr and Mgr, Cetbrasil Representacoes Ltda., Rua Da Quitanda 7—1 Andar, Rio De Janeiro, Brazil, S A.
BOTTUM, Edward W. (*A* 1942; *J* 1938) Chief Engr., Skuttle Manufacturing Co., 517 E. Larned St., Detroit, Mich.
BOUCHER, Kent D. (*M* 1944) Mgr., Carrier Div, United Clay Products Co., 931 Investment Bldg, Washington, D. C
BOUEY, Angus J. (*A* 1937; *J* 1930) Sales Engr, The B. F Sturtevant Co, 681 Market St, San Francisco, Calif
BOUILLON, Lincoln (*M* 1933) Lt, U S N R, 10th N. C. B, F P O, San Francisco, Calif
BOWEN, Leroy F. (*A* 1942) 723 East 71st Terr, Kansas City, Mo
BOWERMAN, E. L. (*A* 1937) Flight Lt., R.C A.F, Officers' Mess, R C A F. Station, Rockcliffe, Ottawa, Canada
BOWERS, A. F. (*A* 1919) Pres, Industrial Heating & Engineering Co, 828 N Broadway, Milwaukee 2, Wis
BOWERS, Charles L. (*M* 1944) Mgr, E G. Sheet Metal Works, 258 Pryor St S W, Atlanta 3, Ga
BOWLER, R. W. (*S* 1943) Ensign, U.S.N.R., 2733 France Ave S, Minneapolis, Minn.
BOWLES, Potter (*A* 1928) Pres, Hoffman Specialty Co., Inc, 1001 York St, Indianapolis, Ind
BOXALL, Frederick (*M* 1937) 36 Kenwood Ave, Verona, N J.
BOYAR, Sidney L. (*A* 1944; *J* 1938) 9514 S Hamilton Ave., Beverly Hills, Chicago, Ill
BOYD, Lyle E. (*M* 1944; *A* 1941) Air Cond Supvsr., Caterpillar Military Engine Co, Decatur, Ill
BOYD, Robert L., Jr. (*J* 1941) 2nd Lt., 465th A. A. F., B U., Paine Field, Everett, Wash
BOYD, Spencer W. (*M* 1937; *J* 1931) Lt, U S N R, Newcomb & Boyd, 615 Trust Co of Georgia Bldg., Atlanta, Ga
BOYD, Thomas D. (*M* 1937) Sales Engr., Johnson Service Co., 1905 Dunlap St, Cincinnati, Ohio
BOYDEN, Davis S.* (*Life Member, M* 1909), (*Presidential Member*), (Pres., 1937; 1st Vice-Pres, 1936; Treas., 1933-34; Council, 1917; 1930-38) 1496 Commowealth Ave, Brighton 35, Mass.
BOYER, Clair L. (*M* 1944) 1003 S Washington St, Dillon, Mont.
BOYLE, John R. (*M* 1936) 6858 Osceola Ave, Chicago 31, Ill
BRAATZ, Chester J.* (*M* 1930) 1819 Clinton St, Rockford, Ill
BRACKEN, John H. (*M* 1927) Mgr, Industrial Uses Dept, The Celotex Corp., 120 S. LaSalle St., Chicago, Ill
BRACKMANN, Walter H. (*A* 1944) Solid Fuel Engr., Kansas City Coal Service Institute, 540 Dwight Bldg., Kansas City, Mo.
BRADBURY, George L. (*M* 1944) Co-Partner and Mgr., Bradbury Brothers Heating Co., 1219 Stout St., Denver, Colo.
BRADFIELD, Joseph L. (*M* 1944) 0-3540 N. Pennsylvania St., Indianapolis, Ind.

BRADFIELD, William W. (*Life Member*; *M* 1926) Mech. Engr., 341 Michigan Trust Bldg., Grand Rapids 2, Mich.
BRADLEY, Eugene P. (*M* 1906) 4 Yale Ave., St. Louis, Mo.
BRADY, George A. (*M* 1944) 548 Douglas St., Salt Lake City, Utah.
BRAGG, Robert E. (*A* 1944) Mfrs. Repr., 5859 N. New Jersey St., Indianapolis 5, Ind.
BRAKHA, M. V. (*M* 1943) Gen Mgr., Carrier Egypt, S. A. E. 37, Kasr El Nil, Cairo, Egypt
BRANDT, Allen D.* (*M* 1940) Safety and Security Div. Ord. Dept., 333 N. Michigan Ave, Chicago 1, Ill.
BRANDT, E. H., Jr. (*M* 1928) Pres., Reliance Engineering Co., Inc., P. O. Box 1292, Charlotte 1, N. C.
BRANDT, Fred C. (*M* 1943) Engr, Minneapolis-Honeywell Regulator Co., 1305 Capitol, Houston 2, Texas.
BRANIFF, Paul R. (*A* 1939) Secy-Treas., Braniff Engineering Co., 817 N. Broadway, Oklahoma City, Okla. (Present address unknown).
BRATT, Hero D. (*M* 1937) 2259 Stafford Ave. S W, Grand Rapids 7, Mich.
BRAUER, Roy (*M* 1926) Dist Mgr., The Trane Co., 512 Magee Bldg, Pittsburgh 22, Pa.
BRAUN, Charles R., Jr. (*J* 1943, *S* 1939) Lt. (jg) U.S.N R., U S. S., YMS 125, c/o F P O, San Francisco, Calif.
BRAUN, Louis T. (*M* 1921) Exec. Secy., Heating, Piping & Air Conditioning Contractors' Chicago Association, 228 N. LaSalle St, Chicago 1, Ill
BRAYMAN, Albert I. (*J* 1937) 340 Boulevard, Revere, Mass
BREN, Leo W. (*A* 1944) 4612 Drew Ave. S, Minneapolis 10, Minn
BRENEMAN, Robert B. (*A* 1931; *J* 1927) Branch Mgr., Armstrong Cork Co, 50 E. Broad St, Columbus, Ohio
BREWER, F. M. (*J* 1941) 1762 Preston Rd, Park Fairfax, Alexandria, Va
BREX, Irving E. (*A* 1939) 1223 Hillside Rd, P O Box 985, Lake Mohawk, Sparta, N. J
BREYER, Frederick (*S* 1940) Ensign, U.S.N R (Address unknown).
BRICKHAM, Ben A. (*M* 1944) Cons. Engr, 1311 S. Vine, Denver 10, Colo.
BRICKHAM, Nelson H. (*A* 1943) 3605 Reed, Cheyenne, Wyo
BRIDE, William T. (*M* 1928; *J* 1925) Bride, Grimes & Co, 9 Franklin St, Lawrence, Mass
BRIGHAM, Clare M. (*M* 1935) Vice-Pres in charge of Sales, C A. Dunham Co, 450 E. Ohio St, Chicago, Ill
BRIGHAM, Frederick H. (*M* 1944) Engr., C. R. Swaney Co, 28 St Botolph St., Boston, Mass.
BRINDELL, Carl E. (*A* 1943) Partner, Brindell & Cooper, 206 Balter Bldg., New Orleans 12, La.
BRINKER, Harry A. (*M* 1934) 2521 University Ave., Kalamazoo, Mich
BRINTON, Joseph W. (*M* 1920) Mgr., Boston Dist., American Blower Corp, 1003 Statler Bldg., Boston 16, Mass.
BRISSENDEN, Carrol W. (*J* 1939) Lt, U S N R., Naval Air Station, Armament Test, Patuxent River, Md.
BRISSETTE, Leo A. (*M* 1930) Treas., Trask Heating Company, 217 Park St, Medford, Mass
BRITTAIN, Alfred, Jr. (*M* 1938) 138 Wheeler Ave., Toronto, Ont., Canada.
BROCHA, John F. (*M* 1936) 1804 Northeast 45th Ave., Portland, Ore.
BROD, Bernard M. (*J* 1941) 63 Sycamore Ave., Mt Vernon, N. Y.
BRODERICK, Edwin L.* (*M* 1933) 9696 Northlawn Ave., Detroit 4, Mich.
BRODIE, Aaron H. (*M* 1942) Pres., J. Brodie & Son, Inc., 1329 E. Fort St, Detroit, Mich.
BRODNAX, George H., Jr. (*M* 1938) Sales Engr., Georgia Power Co., 75 Marietta St. N W., Atlanta 1, Ga.
BROKAW, George K. (*A* 1942; *J* 1939; *S* 1938) 5931 Whitney St., Oakland, Calif.
BRONSON, Carlos E.* (*M* 1919) Chief Mech. Engr., Kewanee Boiler Corp, Kewanee, Ill.
BROOKE, Bernard B. (*M* 1945) Kynance, Beatty Ave., Roath Park, Cardiff, Glamorgan, South Wales.
BROOKE, Irving E. (*M* 1938) Cons. Engr., 189 W. Madison St, Chicago, Ill.
BROOM, Benjamin A. (*Life Member; M* 1914) 2230 East 70th Pl., Chicago 49, Ill.
BROOME, Joseph H. (*A* 1936) 137 McCosh Rd., Upper Montclair, N. J.
BROWN, Alfred P. (*M* 1927) Vice-Pres., Reynolds Corp., 4224 S. Lowe Ave., Chicago 9, Ill.
BROWN, Aubrey I.* (*M* 1923) Prof. of Htg. and Vtg., Ohio State University, Columbus, Ohio.
BROWN, David (*M* 1936) Owner, 67 Cooper Square, New York 3, N. Y.
BROWN, Edward A. (*A* 1944) Owner, E. A. Brown Manufacturer's Agency, 513 Redick Tower, Omaha 2, Nebr.
BROWN, Foskett* (*M* 1926) Pres., Gray & Dudley Co, 2300 Clifton Rd., Nashville 3, Tenn.
BROWN, Guy M. (*M* 1943) 4845 Harriet Ave, Minneapolis, Minn.
BROWN, Harper J. (*J* 1940) 1st Lt., Ordnance Dept., Instructor, Armored Force School, Ft. Knox, Ky.
BROWN, John S., Jr. (*A* 1943; *J* 1937) 428 Hadley Ave, Dayton, Ohio.
BROWN, Mack D. (*M* 1938; *J* 1936) Engr., The Bahnson Co, 1001 S. Marshall St, Winston-Salem, N. C.
BROWN, Marvin L. (*M* 1939) 4328 Stanhope St., Dallas, Texas.
BROWN, Maurice W. (*M* 1943; *J* 1938) 619 Texas Bank Bldg., Dallas 1, Texas
BROWN, Norman (*A* 1944) Owner, Norman Brown Co, 765 Minna St, San Francisco 3, Calif.
BROWN, Richard C. (*M* 1944) Partner and Mgr., Campbell Elsey & Co., 326 Ness Bldg., Salt Lake City 1, Utah.
BROWN, Robert M. (*A* 1944) Owner, Brown & Co, 305 Techwood Drive N W., Atlanta, Ga.
BROWN, Sterling D. (*A* 1944; *J* 1939) Assoc. Mech Engr, U. S. Navy Yard, Bldg No 47, Public Works Design Sect., Mare Island, Calif.
BROWN, Tom (*M* 1930) 22151 Gratiot Ave., East Detroit, Mich.
BROWN, Ward A. (*A* 1944) Assoc. Engr., George Wagschal Associates, 1131 Majestic Bldg., Detroit 26, Mich.
BROWN, William L., Jr. (*A* 1942) 1418 Woodland Drive, Durham, N C.
BROWN, W. Maynard (*A* 1930) Secy. and Treas., Warren Webster & Co., 17th and Federal Sts, Camden, N. J.
BROWNE, Alfred L. (*M* 1923) Repr., Illinois Engineering Co., 253 Highland Rd., South Orange, N J
BROWNE, Leland W. (*M* 1944) Exec. Vice-Pres, The Darby Corp., First and Walker, Kansas City 15, Kans
BRUCE, Marshall (*A* 1942) Asst. Secy., George W. Akers Co., 16525 Woodward, Detroit 3, Mich.
BRUNDAGE, F. Ward (*J* 1940) 1st Lt., U. S. Army, Fort Bliss, Texas.
BRUNET, Adrian L. (*M* 1923) P. O. Box 36, Rockville, Md.
BRYAN, William L., Jr. (*J* 1942) Lt, Hygiene Unit, Astoria Health Center, 12-26–31 Ave., Astoria 2, N. Y.
BRYCE, John H. (*M* 1944) Supervising Engr, Kaighin & Hughes, 125 S. Huron St, Toledo, Ohio
BRYNER, John J. (*M* 1942) 5701 Canal Blvd, New Orleans 19, La
BUCHANAN, Walter P. (*M* 1944) N E. Mgr., B. F. Sturtevant Co, 89 Broad St., Boston 10, Mass.
BUCK, David T. (*M* 1940; *A* 1936) Pres., Buck Engineering Co, Inc, 37-41 Marcy St., Freehold, N. J.
BUCK, Lucien (*M* 1928) 105 Jericho Manor, Jenkintown, Pa.
BUCKINGHAM, Robert B. (*A* 1944) Partner, Associated Southern Industries, 1161 Union Ave, Memphis, Tenn.
BUENGER, Albert* (*M* 1920; *J* 1917), (Council, 1934-37) Bldg. Supt., Hotel Gibson, Cincinnati 1, Ohio.
BUENSOD, Alfred C. (*M* 1918) 33 Fifth Ave., New York 3, N. Y.
BULL, Frederick W. (*M* 1942) Lt., U.S.N R., 148 State St., Portland, Maine.

BULL, Harold A. (*M* 1944) Sales Engr., Kruse Engineering Co., 24 Commerce St., Newark 2, N. J.

BULLER, C. R. (*M* 1943) 2629 S. Shore Drive, Milwaukee, Wis.

BUNNELL, E. W. (*M* 1943) Cons. Engr., 426 ▶ Hutton Bldg., Spokane 8, Wash.

BURBO, W. G. (*M* 1944) 13 Fordham Rd., West Newton 65, Mass.

BURCH, Laurence A. (*M* 1934) 78 Amherst Rd., Pleasant Ridge, Mich.

BURGES, Joseph H. M. (*J* 1939) Apprentice Seaman, U. S. Navy, 1949 McGraw Ave., Bronx 62, N Y.

BURKE, J. J. (*M* 1939) 709 N. Broom St., Wilmington 34, Del

BURKE, J. S. (*M* 1942) Dealer Coordinator, New Orleans Public Service, Inc., 317 Baronne St., New Orleans 9, La.

BURKES, Lloyd C. (*A* 1943) Mgr, Branch Office, Barber-Colman Co., 625 Southeastern Bldg, Greensboro, N. C.

BURNAM, C. M., Jr. (*M* 1938; *A* 1937) Editor, Heating, Piping & Air Conditioning, 6 N. Michigan Ave., Chicago 2, Ill.

BURNAP, C. H. (*M* 1941) Sales Engr., A K. Howell Co., 1635 Syndicate Trust Bldg, St. Louis 1, Mo.

BURNETT, Earle S. (*M* 1920) 4223 West 11th Ave, Amarillo, Texas.

BURNEY, L. Byron (*M* 1944) 22 Frederick Rd., Portsmouth, Va.

BURNS, Edward J. (*M* 1923) 3900 Holmes St., Kansas City, Mo.

BURNS, F. G. (*M* 1940) Major, U. S Army, P O. Box 2601, Washington 13, D. C.

BURNS, Fred C. (*M* 1942) Branch Mgr., Kewanee Boiler Corp, 2020 Wyandotte St, Kansas City, Mo.

BURNS, Harold J. (*A* 1941; *J* 1939) 105 Allen Rd, Friendship Station, D. C.

BURNS, Robert (*M* 1944) 54 Elliot Ave, Yonkers, N Y.

BURNS, Robert O. (*A* 1944) Lt, U S Army, 1704 S. Travis St, Sherman, Texas

BURR, Griffith C. (*M* 1937) Box 239, Wrightsville Beach, N C.

BURRITT, Charles G. (*A* 1916) Mgr, Dallas Office, Johnson Service Co, 922 Second Ave. S, Minneapolis 2, Minn.

BURRITT, Edward E., Jr. (*A* 1941) 202 Kathmer Rd., Brookline, Upper Darby, Pa

BURROWS, Austin J. (*A* 1944) 237 Bridge St, Northampton, Mass

BURTCHAELL, James T. (*A* 1941) Pres, Rushlight's, Inc, 407 S.E Morrison St., Portland, Ore.

BURTON, W. Russell (*A* 1939) Warrant Officer, U.S.N.R., 2816 Northeast 19th St., Portland 12, Ore

BUSCHMANN, Alfred W. (*A* 1944) Mgr., Buschmann Co., 208 E St. Clair St., Indianapolis, Ind.

BUSENLENER, Louis V. (*M* 1944; *A* 1943) Mgr, Air Cond Div., Higgins Industries, Inc, 1755 St. Charles Ave., New Orleans, La

BUSHNELL, Carl D. (*A* 1921) Pres., The Bushnell Machinery Co, 311 Ross St., Pittsburgh, Pa

BUSSE, Herbert (*M* 1938) 16760 Greenview Rd., Detroit, Mich

BUTLER, Peter D. (*M* 1922) 127 Edgewater Rd, Cliffside Park, N J

BUTLER, Robert P. (*A* 1944) Partner, Kerr Machinery Co., 608 Kerr Bldg., Detroit 26, Mich.

BUTT, Roderick E. W. (*A* 1936; *J* 1930), (Present address unknown).

BUZZARD, Francis H. (*M* 1939) 624 Wood Lane, Haddonfield, N. J.

BYERLEY, Francis A., 3rd (*A* 1944) 730 W. Washington Ave, South Bend, Ind.

BYERS, Robert L. (*M* 1942) Chief Engr., John Paul Jones, Cary & Millar, 448 Terminal Tower, Cleveland, Ohio

BYRD, T. I. (*A* 1936) 2311 S. Sutphin St., Middletown, Ohio

BYRNE, Joseph J. (*A* 1939) Htg. Engr., Kleenair Furnace Co., 5329 N. E. Sandy Blvd., Portland, Ore.

BYSOM, Leslie L. (*M* 1938) 1214 Eighth St., Bremerton, Wash.

C

CADY, Edward F. (*A* 1943; *J* 1937) 2240 Rexwood Rd., Cleveland Heights, Ohio.

CAIN, William J. (*J* 1943; *S* 1940) 9111 Delphine Ave., Overland, Mo.

CALCATERRA, Louis A. (*A* 1944) 3111 Reed's Lake Blvd., Grand Rapids, Mich.

CALDWELL, Arthur C. (*M* 1930) 550 South 48th St., Philadelphia 43, Pa.

CALDWELL, Robert J. S. (*M* 1941) Resident Engr., c/o Air Conditioning & Engineering Co., Ltd, P. O. Box 7821, Johannesburg, S. Africa

CALDWELL, William J. (*A* 1945) Gen. Mgr., William J. Caldwell Co, 1129 Vermont Ave. N W., Washington 5, D. C.

CALEB, David (*M* 1923) Engr., Kansas City Power & Light Co., 1330 Baltimore Ave., Kansas City, Mo

CALHOON, Floyd N. (*M* 1942) Asst. Prof of Mech Engrg, University of Michigan, 237 W. Engineering Bldg., Ann Arbor, Mich

CALL, Joseph (*M* 1938; *J* 1936) 50 Fairfield Rd, Brookline Park, Del Co, Pa.

CALLAHAN, Peter J. (*M* 1934) 4057 Amboy Rd, Great Kills, S.I., N. Y.

CALLAHAN, Thomas H. (*M* 1943) Asst. Treas, Callahan Equipment Co., Inc., 2 William St, White Plains, N. Y.

CALLAN, John J., Jr. (*A* 1943) Mgr, Htg Equipment Dept., Claflin-Sumner Coal Co, 10 Franklin St, Worcester 8, Mass.

CALLAN, William D. (*A* 1944) 5041 Coad Drive, Cincinnati 29, Ohio.

CALLAWAY, James H. (*A* 1944) 209-14th St. N E., Atlanta, Ga.

CALNAN, Daniel J. (*A* 1942) 6 Homewood Ave, Yonkers 2, N Y.

CALMAN, Edward J. (*M* 1941) Control Supt, Ontario Paper Co, Ltd, Thorold, Ont, Canada

CALONGNE, Henry J., Jr. (*A* 1944) Mech Engr., Emile M. Babst Co, 1050 Camp St, New Orleans 13, La

CAMBRIDGE, W. A. (*A* 1944) Pres, Reid & Cambridge, Ltd., 1223 Greene Ave, Westmount, Que, Canada

CAMERON, Joseph A., Jr. (*A* 1944) Mgr, New York Office, The Marley Co, Inc, 2 Rector St., New York 6, N. Y.

CAMERON, Robert T. (*J* 1941; *S* 1938) 208 East 16th St, New York, N. Y

CAMPAU, W. R. (*M* 1940) Secy and Gen Mgr, Kendall Heating Co, 1636 N.W. Lovejoy St, Portland 9, Ore

CAMPBELL, Alfred Q., Jr. (*A* 1940; *J* 1933) Capt, Field Artillery, U. S Army, 1678 York St, Memphis, Tenn

CAMPBELL, E. K., Jr. (*M* 1944) Vice-Pres, E K Campbell Heating Co, 2445 Charlotte St., Kansas City, Mo

CAMPBELL, Everett K.* (*Life Member*; *M* 1920), (Treas, 1942-43; Council, 1931-33; 1939-43) Vice-Pres., E K. Campbell Heating Co, 2441-3-5 Charlotte St., Kansas City 8, Mo

CAMPBELL, George S. (*A* 1941; *J* 1937) Lt (jg) Tech Asst. Naval Inspector of Ordnance, 306 Sunnyside Drive, Chattanooga 4, Tenn.

CAMPBELL, George W. (*J* 1939) Major, Air Corps, 253 Northwest 34th St., Miami, Fla

CAMPBELL, Roger P. (*J* 1939) Ensign, U.S.N R., E. K. Campbell Heating Co., 2445 Charlotte St., Kansas City, Mo

CAMPBELL, Thomas F. (*M* 1928) T. F. Campbell Co., 1013 Penn Ave., Pittsburgh 21, Pa.

CAMPION, Charles R. (*M* 1944) Partner, Wrightson & Campion, 55 West 42nd St., New York 18, N. Y.

CANDEE, Bertram C. (*M* 1933) 19 Tremont Ave., Kenmore 17, N. Y.

CANNON, Vernon K. (*A* 1944) Capt. QMC, Quarters 113, Ft Francis E. Warren, Wyo.

CARBONE, James H. (*M* 1937) 121-13-198th St., St. Albans, L. I., N. Y.

CARDER, William W. (*A* 1944) Branch Mgr., Johnson Service Co., 313 Bona Allen Bldg., Atlanta 3, Ga.

CAREY, Paul C. (*M* 1930) Cons. Engr., Runyon & Carey, 33 Fulton St., Newark 2, N. J.

CARLE, William E. (*Life Member*; *M* 1926) Pres.-Treas., Carle-Boehling Co., Ltd., 1641 W. Broad St., Richmond 20, Va.

CARLETON, Herbert G. (*A* 1944) Box 143, College Park, Md.
CARLOCK, Marion F. (*M* 1936) Capt., Officer in Charge, Army Map Service, 650 S. Clark St, Chicago, Ill.
CARLSON, Clarence J. (*A* 1944) 2701 Roosevelt Ave., Indianapolis, Ind.
CARLSON, C. O. (*A* 1937) Owner, C. O. Carlson Heating Co., 1627 Washington Ave., N., Minneapolis, Minn.
CARLSON, Everett E. (*M* 1932; *A* 1929) Branch Mgr., The Powers Regulator Co., 2726 Locust St., St. Louis, Mo.
CARNAHAN, John H. (*A* 1940; *J* 1937) 1808 West 26th St., Pine Bluff, Ark
CARPENTER, R. H. (*M* 1921), (Council, 1930-35) Mgr., New York Office, Nash Engineering Co., Graybar Bldg., 420 Lexington Ave, New York, N. Y.
CARRIER, Earl G. (*M* 1936; *J* 1929) 326 Highland Ave., Winchester, Mass
CARRIER, Willis H.* (*Honorary Member*; *Life Member*; *M* 1913), (*Presidential Member*), (Pres, 1931; 1st Vice-Pres., 1930; 2nd Vice-Pres, 1929, Council, 1923-32) Chairman of the Board, Carrier Corp, 300 S. Geddes St, Syracuse 1, N. Y.
CARROLL, Daniel E. (*A* 1941) 216-09-40th Ave, Bayside, N. Y.
CARROLL, Edgar E. (*A* 1939) Owner, Kleenair Furnace Co, 5329 N.E Sandy Blvd., Portland, Ore.
CARROLL, William M. (*A* 1943; *J* 1938) Gen Mgr, Macklanburg Supply Co., 23rd at Robinson, Oklahoma City, Okla
CARSEY, E. A. (*M* 1944) 359 Howell, Cincinnati 20, Ohio.
CARSON, C. C. (*M* 1943) 6751 Potomac Drive, N W, Washington 16, D C
CARTER, A. Walter (*M* 1940, *J* 1936) Engr, Chatco Steel Products, Ltd, Chatham, Ont, Canada.
CARTER, Chester A. (*M* 1944) 3025 South 33rd St, Omaha, Nebr
CARTER, Doctor (*M* 1934) 50 Nevill Rd, Hove, Sussex, England
CARTER, Henry G. (*A* 1944) Mfrs Agent, 5505 Branch Ave, Tampa 4, Fla.
CARTER, John H.* (*M* 1936) Lt U.S.N.R, 160 Navy Walk, Apt 6-B, Ft Greene Houses, Brooklyn 1, N.Y.
CARTIER, Marcel E. (*M* 1944) Research Engr, American Radiator & Standard Sanitary Corp, 675 Bronx River Rd., Yonkers 4, N. Y
CARY, Edward B. (*M* 1935) Capt., U S N R., 448 Terminal Tower, Cleveland, Ohio
CASE, Delbert V. (*M* 1937) Engrg Cons, War Food Administration, 2200 Fidelity Bldg, Kansas City 6, Mo
CASE, Donald P. (*A* 1944) 8631 Hiawatha Rd, Kansas City 5, Mo
CASE, Walter G. (*A* 1930) 66 The Ridgeway, Kenton, Harrow, Middlesex, England.
CASE, Byron L. (*M* 1921) Mgr, Northern District, Ilg Electric Ventilating Co, 222 N LaSalle St., Chicago, Ill.
CASKEY, Luther H., Jr. (*J* 1941, *S* 1938) Capt, U.S C E., Newcastle Army Air Base, Wilmington 99, Del.
CASKEY, Thomas C. (*M* 1943) Owner, Caskey Engineering Co, 1736 First Ave., S., Seattle 4, Wash.
CASSELL, William L. (*M* 1936) Mech. Engr., 912 Baltimore Ave., Kansas City, Mo.
CATLETT, W. A. (*A* 1943) Owner, Catlett Engrs., 1350 Buckner Blvd, Dallas 18, Texas.
CAUHORN, A. V. (*A* 1943) Owner, A V. Cauhorn Co., 1001 Broadstreet, Detroit 4, Mich.
CAVANAGH, P. A. (*M* 1944) Mgr, The Herman Nelson Corp., 1015 Chestnut St, Philadelphia 7, Pa.
CHADWICK, John B. (*M* 1944) 11 Orville Drive, Levenshulme, Manchester, 19, England.
CHALMERS, C. H. (*Life Member*; *M* 1925) Gen. Mgr., Chalmers Oil Burner Co, 318 First Ave. N., Minneapolis 1, Minn.
CHAMBERS, Fred W. (*M* 1936) Pres, F. W. Chambers & Co., Ltd., 96 Bloor St. W., Toronto 5, Ont.
CHAMBERS, S. A. (*M* 1944) 81 Ross Ave., Hackensack, N. J.
CHAMPLIN, Robert C. (*M* 1945; *A* 1938) 13640 Mendota Ave., Detroit 4, Mich.
CHANDLER, Roy (*A* 1944) W. H. Wheeler, Inc., 234 East 46th St., New York, N Y.
CHAPIN, C. Graham (*M* 1933) Treas., Hopson & Chapin Manufacturing Co., 231 State St, New London, Conn.
CHAPIN, Harvey G. (*M* 1935) Capt 0-917608, 883rd Bomb Squad., 500th Bomb. Group, APO No. 17159, c/o Postmaster, San Francisco, Calif.
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MAYNARD, J. Earle (*M* 1931) 11711 Ilane St, Detroit 4, Mich
MAYNE, Walter L. (*M* 1938) Production-Sales Engr., Tanner Mfg, Co, 1014 West 12th St., Erie, Pa.
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McBRIDE, J. Nevins (*A* 1941) Vice-Pres, Frank A McBride Co., 160 Ward St., Paterson 1, N J.
McCABE, John H. (*M* 1944) Rocky Mountain Sales Mgr., American Blower Corp., 1730 Glenarm St., Denver, Colo.
McCAFFRAY, Charles E. (*M* 1938) Ruxton, Baltimore 4, Md
McCAIN, H. King (*M* 1939; *A* 1938, *J* 1937) Major, Q M. C, c/o Mrs. H. K McCain, 515 Forrer Blvd., Dayton, Ohio
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McCARL, Harry E. (*M* 1944) 2206 37th St, Des Moines 10, Iowa.
McCAUL, Lynn K. (*M* 1942) 622 W. Marshall, Ferndale 20, Mich.
McCAULEY, James H. (*M* 1921) Pres, J. H. McCauley & Son, 5620 West 65th St, Chicago, Ill.
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McCLINTOCK, William (*M* 1935) 4116 Carpenter Ave., New York, N. Y.
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McCORMACK, Denis (*M* 1933) Pres., Fox Munitions Corp., 21st and Arch Sts., Philadelphia 3, Pa.
McCORMICK, George W., Jr. (*A* 1941) Capt., U.S M.C., 1320 Sheridan Rd., Menominee, Mich.
McCOY, C. E. (*M* 1936) Partner, Turner-McCoy, 315 W. Second St., Little Rock, Ark.
McCOY, Thomas F. (*M* 1924) Mgr., The Powers Regulator Co., 125 St. Botolph St., Boston 15, Mass.
McCREA, Joseph B. (*M* 1937) 2919 Drexel Ave, Detroit, Mich.
McCREA, Lester W. (*M* 1940) Prop., McCrea Sales Co., 15 East 21st St., Baltimore 18, Md.
McCULLEY, D. E. (*A* 1941) Sales Engr., 814 South 14th St., Omaha 2, Nebr.
McCULLOUGH, Henry G. (*M* 1939) 7042 Lincoln Drive, Mt. Airy, Philadelphia, Pa.
McCULLOUGH, John L. (*M* 1939) 731 Country Club Drive, Pittsburgh 16, Pa.
McCUNE, Laban J. (*M* 1943) 118 E. Sixth St, Tulsa 3, Okla.
McCUSKER, James P. (*J* 1944; *S* 1940) 1445 Evarts St. N.E., Washington, D. C.
McDANIELS, James A. (*A* 1944) 4950 Ford St., Speedway City, Indianapolis, Ind.
McDERMOTT, John P. (*A* 1942; *J* 1939) Major C.E., Asst Division Engr., Alaskan Div, Air Transport Command, A. P. O. 462, Minneapolis, Minn.
McDERMOTT, Paul F. (*M* 1943) Section Head in Charge of Asbestos Thermal Insulations Development, Johns-Manville Corp, Manville, N J
McDONALD, Frank L. (*M* 1944) Branch Mgr, Barber-Colman Co., 816 West Fifth St, Los Angeles 13, Calif.
McDONALD, Ivan (*A* 1938) Capt. (overseas), Minneapolis-Honeywell Regulator Co, 117 Peter St, Toronto, Ont., Canada.
McDONALD, John J. (*M* 1943) 401 Beacon St, Chestnut Hill, Mass
McDONALD, Thomas (*A* 1931) Vice-Pres, Minneapolis-Honeywell Regulator Co, 1135 N Cicero Ave, Chicago 51, Ill
McDONNELL, Everett N. (*M* 1923), (Council, 1940-44) Pres., McDonnell & Miller, 400 N. Michigan Ave, Chicago 13, Ill
McDONNELL, John E. (*A* 1936) Sales Engr, McDonnell & Miller, 400 N. Michigan Ave., Chicago 13, Ill.
McDOWELL, Harry L. (*J* 1939) Lt. (jg), U.S.N R. P O. Box 366, Scotland Neck, N. C.
McEAHERN, Edward J. (*A* 1945; *J* 1944) 4754 Raleigh St, Denver 12, Colo.
McELGIN, John W.* (*A* 1937; *J* 1931) Tennis Ave., Ambler, Pa
McFADDEN, H. J. (*A* 1944) Sales Engr., Servel, Inc., 706 Lone Star Gas Bldg, Dallas 1, Texas
McFARLAN, Alden I. (*M* 1942) 691 Dorian Rd., Westfield, N. J.
McGEARY, Paul W. (*M* 1944) Mgr., Indiana Dist. Office, The Trane Co., 635 N. Penn, Indianapolis, Ind.
McGEORGE, Richard H. (*M* 1937) 14565 Glastonbury Rd., Detroit, Mich.
McGINNIS, Frank L. (*M* 1940) 332 N. Henry St., Williamsburg, Va.
McGONAGLE, Arthur (*M* 1932) Cons. Engr, 6815 Prospect Ave., Ben Avon, Pittsburgh 2, Pa.
McGOVERN, Arthur F. (*A* 1945) Dist. Mgr., Modine Manufacturing Co., 83 S. High St., Columbus, Ohio.
McGOWAN, T. Edward (*M* 1942) 1545 Bradbury Rd, San Marino 9, Calif.
McGOWN, Frederick H., Jr. (*J* 1941; *S* 1939) Cpl., U. S. Army, c/o The McGown Co, Cooperstown, N. Y.
McGRAIL, Thomas E. (*M* 1926) Local Repr., Canadian Sirocco Co., Ltd., 63 Sparks St., Ottawa, Ont., Canada.
McGREW, Joseph A. (*A* 1944) Owner, J. A. McGrew Supply Co., 2163 Market St, Denver 2, Colo.
McGRUER, Alfred E. (*A* 1945) Supvsr of Stationary Boiler Plants, Canadian Pacific Railway Co., Room 238, Union Station, Toronto, Ont., Canada.
McGUINESS, Cyrus H. (*A* 1943) 218 Franklin, Des Moines, Iowa.
McILVAINE, John H. (*M* 1929) Pres., McIlvaine Products, Inc., 1516 Callowhill St., Philadelphia 30, Pa.
McINDOE, James F. (*M* 1939) 1900 46th Ave, San Francisco 16, Calif.
McINTIRE, James F.* (*M* 1915; *A* 1914), (*Presidential Member*), (Pres., 1939; 1st Vice-Pres, 1938; 2nd Vice-Pres., 1937; Council, 1926-28; 1932-40) 1st Vice-Pres., U. S. Radiator Corp., and Pres., Pacific Steel Boiler Corp, 1500 United Artists Bldg., Detroit 31, Mich.
McINTOSH, Fabian C. (*M* 1921; *J* 1917), (Council, 1929-31; 1933-35, 1942) Mgr., Pittsburgh Office, Johnson Service Co, 1238 Brighton Rd., Pittsburgh 12, Pa.
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McKEEMAN, Clyde A.* (*M* 1936) Director of War Training, Case School of Applied Science, 10900 Euclid Ave, Cleveland 6, Ohio.
McKENZIE, Murdock C., Jr. (*M* 1938) 3806 Boyce Ave, Los Angeles, Calif.
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McKINNEY, Carl A. (*A* 1939, *J* 1937) 2136 Addison Rd, Houston 5, Texas.
McKINNEY, William J. (*M* 1938; *A* 1934) Southern Dist. Mgr, American Blower Corp, 101 Marietta St. Bldg, Atlanta 3, Ga.
McKITRICK, Walter D. (*M* 1936) 3511 Upton Ave, Toledo, Ohio
McKITTRICK, Percy A. (*A* 1934) Treas-Gen. Mgr, Parks-Cramer Co. P O Box 444, Fitchburg, Mass.
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McLAREN, T. H. (*A* 1938) Gen Sales Mgr, The James Morrison Brass Manufacturing Co, Ltd, 276-278 King St W, Toronto 1, Ont., Canada.
McLARNEY, Harry W. (*M* 1933) Indus Engr, Union Electric Co. of Missouri, 315 North 12th Blvd, St. Louis 1, Mo
McLEAN, Dermid (*M* 1917) Mech. Engr., Snyder & McLean, 2214 Penobscot Bldg., Detroit, Mich.
McLEISH, William S. (*A* 1932; *J* 1928) Chief Engr., The Ric-wiL Co, Barberton, Ohio.
McLENEGAN, David W.* (*M* 1933) Engr in Charge, General Electric Co, 5 Lawrence St., Bloomfield, N. J.
McLENNAN, Millard C. (*M* 1944) 1714 Fourth Ave N, Seattle, Wash
McLEOD, Clarence H. (*M* 1943) Application Engr, York Corp, 1239 Liberty Bank Bldg, Dallas 1, Texas.
McLOUTH, Bruce F. (*M* 1936; *J* 1934) 3445 Humboldt Ave. S., Minneapolis 8, Minn.
McMAHON, Louis A. (*A* 1944) Htg Engr, American Furnace Co., 2719 Delmar Blvd, St. Louis 3, Mo
McMAHON, Thomas W. (*M* 1928) Dist Mgr, American Blower Corp, 1711 Railway Exchange Bldg., St. Louis 1, Mo.
McMICHAEL, Chester L. (*M* 1943) 5519 Aberdeen Rd., Kansas City, Kan.
McMULLEN, Earle W. (*M* 1938) Dir of Research, The Eagle-Picher Lead Co, C and Porter Sts., Joplin, Mo
McMULLEN, Ernest W. (*M* 1942) 240 Winslow Rd., Waban 68, Mass
McNAMARA, William (*A* 1930) Mgr, The Trane Co., 850 Cromwell Ave, St Paul, Minn.
McNAMEE, Earl W. (*M* 1940) Engr., B. & J. Jacobs Co., 1729 John St., Cincinnati 14, Ohio.
McNEVIN, Joseph E. (*M* 1937) Mgr., Colorado Heating Co., 950 Cherokee St, Denver, Colo
McPHERSON, Edwin Robert (*A* 1944) Dir. of Manufacturing Facilities, Crosley Corp., Arlington St., Cincinnati, Ohio.
McPHERSON, William A. (*M* 1929) 86 Dwinnell St., West Roxbury, Mass.
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McQUEEN, Boston (*A* 1945) Engr., Hall-Neal Furnace Co., 1324 N. Capitol Ave., Indianapolis, Ind.
McRAE, Malcolm W. (*M* 1939) Research Engr., Crane Co., 836 S. Michigan Ave., Chicago 5, Ill.
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MEAD, Edward A. (*M* 1926) Sales Mgr., The Nash Engineering Co., South Norwalk, Conn.
MEAD, Harry K. (*A* 1939) Owner, 1100 Guardian Bldg., Portland 4, Ore.
MEAGHER, Arthur T. (*M* 1938) 83 Seymour St., Halifax, Nova Scotia, Canada.
MEANS, James M. (*A* 1944) Owner, Mfrs. Agt., 244 Washington St. Room 433, Boston, Mass.
MEATES, Richard F. (*A* 1944) Sgt., U. S. Army, Det. 482nd Qm. Ref. Mobile, A. P. O 706, c/o Postmaster, San Francisco, Calif.
MEDCALF, Lloyd C. (*M* 1944) Cons. Engr, 100 Stevens Ave., Mt. Vernon, N. Y.
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MEINHOLTZ, Herbert W. (*M* 1936) 1615 N. Springwood Drive, Silver Springs, Md.
MELLON, James T. J. (*Life Member*; *M* 1911), Council, 1915) Mellon Co., 4415-21 Ludlow St., Philadelphia 4, Pa.
MELNICK, Nicholas A. (*M* 1941) 279 Fifth Ave, San Francisco 18, Calif.
MELONEY, Edward J. (*M* 1937) Vice-Pres-Secy, Bowers Bros., Co., 2015 Sansom St., Philadelphia 3, Pa.
MELTON, Rupert D. (*J* 1942) CM 1/C, 117 West 10th St., Charlotte, N. C.
MENDEN, Peter J. (*M* 1935) 706 Fifth Ave. N, Sturgeon Bay, Wis.
MENDITCH, Barney (*J* 1944) Cpl, U S Army, 134th M. P. Co, A. P. O. 782, c/o Postmaster, New York, N Y.
MENSING, Frederick D. (*M* 1920), (Treas, 1931-32; Council, 1931-32) Cons Engr, Mensing & Co., 2845 Frankford Ave., Philadelphia 34, Pa.
MEREDITH, John W. (*M* 1944) 1 Sunnyside Rd., Natick, Mass.
MERENS, Seymour H. (*A* 1939) Partner, Max Miller & Co., 823 N California Ave, Chicago, Ill
MERGARDT, Albert P. (*A* 1940) Prop., American Heating Co., 55 K St S E., Washington, D C.
MERRILL, Carle J. (*M* 1919) Treas., C J. Merrill, Inc., 54 St. John St, Portland, Maine
MERRILL, Frank A. (*M* 1934) Cons. Engr, Office of Hollis French, 210 South St, Boston, Mass.
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MERTZ, Walter A. (*M* 1919) Secy., The Kehm Corp, 51 E. Grand Ave, Chicago 11, Ill.
MERWIN, Gile E. (*M* 1924; *J* 1923) Mgr., The Trane Co., 5012 Parker St, Omaha 4, Nebr.
METCALF, Ralph H. (*M* 1944) Zone Supvsr, Minneapolis-Honeywell Regulator Co., 5005 Euclid Ave., Cleveland 3, Ohio
METZGER, Albert F. (*M* 1940) Supvsr of Steam Utilization Div., Allegheny County Steam Heating Co., 435 Sixth Ave., Pittsburgh, Pa
METZGER, H. J. (*A* 1937) Pres., Wheeler-Blaney Co., 137 E Water St., Kalamazoo, Mich
MEYER, E. Walter (*A* 1944) Partner, Keystone Sales Co., 547 Lincoln Ave., PO. Box 4201, Bellevue Sta., Pittsburgh 2, Pa.
MEYER, Frank L.* (*M* 1932; *J* 1928) Pres., The Meyer Furnace Co., Peoria 2, Ill.
MEYER, Henry C., Jr.* (*Life Member*; *M* 1898), (Council, 1915-16) Pres., Meyer, Strong & Jones, Inc., 101 Park Ave., New York, N. Y.
MEYER, Karl A. (*M* 1938) Prod. Engr., Herman Nelson Corp., Moline, Ill.
MEYERS, Carl F. (*A* 1942) 142 East 35th St, Erie, Pa.
MICHAEL, Leonard A. (*M* 1945) 1346 St. Paul St., Denver 6, Colo.
MICHAEL, Robert K. (*A* 1944) 1316 Steele St, Denver, Colo.
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MILES, James C. (*M* 1943) Miles Heating Enterprises, 1836 Euclid Ave, Cleveland 15, Ohio.
MILLARD, Junius W. (*M* 1929) Lt., U. S. Navy, 1025 Laird St., Key West, Fla.
MILLER, Archibald T. (*M* 1938) 125 Godwin Ave., Ridgewood, N. J.
MILLER, Charles A. (*Life Member*; *A* 1917) Salesman, The H. B. Smith Co., Inc., 331 Madison Ave., New York, N. Y.
MILLER, Charles W. (*M* 1919; *J* 1908) Pres., The Rado Co., Room 204, 759 N. Milwaukee St., Milwaukee 2, Wis.
MILLER, David C. (*M* 1944) 5 Ellis St., Quincy 69, Mass.
MILLER, Edgar R. (*A* 1935) Chief Engr., Winnipeg Cold Storage Co., Ltd, Salter and Jarvis Ave., Winnipeg, Man., Canada.
MILLER, Floyd A. (*Life Member*; *M* 1911) 944 Montrose Ave., Chicago, Ill.
MILLER, Frederick T. J. (*A* 1943) 5713 Herbert St., Norfolk 2, Va.
MILLER, George F. (*M* 1936) Sales Engr, 1614 K St. N W., Washington 6, D. C.
MILLER, H. Deane (*A* 1944) Mgr., Air Cond. and Comml Refrig. Dept., General Electric Supply Corp., P. O. Box 1919, Salt Lake City 12, Utah.
MILLER, Jacob (*M* 1936) Pres., Hy-Grade Construction Co., Inc., 342 West 14th St., New York, N. Y.
MILLER, John W. (*M* 1941) R. 5, Box 498, Lansing, Mich.
MILLER, Leo B. (*M* 1926) Sales Exec., Perfex Corp, 500 W. Oklahoma Ave., Milwaukee 7, Wis.
MILLER, Lorin G.* (*M* 1933), (Council, 1942-44) Prof and Head, Mech. Engrg Dept., Michigan State College, R. E. Olds Hall of Engrg, East Lansing, Mich.
MILLER, Mahlon S. (*A* 1942) Asst. Gas Engr., Iowa Public Service Co., Cherokee, Iowa.
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MILLER, William T. (*M* 1938) Prof., Htg. Vtg, Purdue University, West Lafayette, Ind.
MILLHAM, Franklyn B. (*M* 1938) 4020 Que St. S.E, Washington 20, D. C.
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MILLS, Doremus L. (*A* 1943) Research and Development Engr., Revere Copper & Brass, Inc., Rome, N. Y.
MILLS, Daniel M. (*A* 1940) Mgr, Houston Div, Gray & Crawford, Houston 2, Texas.
MILLS, Hartzell C. (*A* 1935) 4137 Tenth Ave S, Minneapolis 7, Minn
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MINKLER, William A. (*M* 1940) Westinghouse Electric Elevator Co., 150 Pacific Ave., Jersey City 4, N. J.
MINOR, James E. (*S* 1943) Lt. (jg) U.S.N R., 1208 Cherry St., Kalamazoo, Mich.
MINSON, Frederick L. (*J* 1943) Engr, 314-51st St., Newport News, Va.
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MIZENER, Ralph S. (*M* 1944) Sales Engr., Parent & Kirkbride, 4th and Locust Sts., Philadelphia 6, Pa.
MOE, Parker A. (*M* 1944) 3713 W. Branting Lane, Milwaukee, Wis.
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MOHN, H. Leroy (*M* 1937) Exp. Research, Fitzgibbons Boiler Co., Inc., Oswego, N. Y.
MOHRFELD, Herbert H. (*A* 1943; *J* 1935) Vice-Pres. and Treas., C. P. Mohrfeld, Inc., 24 Lees Ave., Collingswood, N. J.
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MOLFINO, Philip (*M* 1938) 125 Clayton St., San Francisco, Calif.
MOLLENBERG, Harold J. (*M* 1936) 172 West Gate Rd., Kenmore, N. Y.
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MONIES, Ellis A. (*M* 1944) Pioneer Ave., Dallas, Pa.
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MOORE, Frank C. (*A* 1938) 44 Lola Rd., Toronto, Ont., Canada.
MOORE, H. Carlton* (*M* 1935) 145 Beaumont Ave., Newtonville 60, Mass.
MOORE, H. Lee (*M* 1919), (Council, 1927-28) Engr., 431 Fulton Bldg., Pittsburgh 22, Pa
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MOORE, Herbert S. (*A* 1923) 107 Clendenan Ave., Toronto, Ont., Canada.
MOORE, MacDonnell (*A* 1940) Pres.-Gen Mgr., The Moore Fuel Corp., 23 Rose St., Danbury, Conn.
MOORE, R. Edwin (*M* 1944; *A* 1928) Vice-Pres., Bell & Gossett Co., 8200 N. Austin Ave., Morton Grove, Ill.
MOORE, Robert E. (*A* 1943) Archt, Moody and Moore, 216 Graham Ave., Winnipeg, Man., Canada.
MOORE, Roscoe S. (*M* 1944) Precipitron Engr., Westinghouse Electric & Manufacturing Co., 20 N. Wacker Drive, Chicago 6, Ill.
MOORE, Wesley R. (*M* 1937) Regional Mgr., Minneapolis-Honeywell Regulator Co., 5005 Euclid Ave., Cleveland, Ohio.
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MORGAN, Robert C. (*Life Member*; *M* 1915) Chairman of the Board, Stewart A. Jellett Co., 1200 Locust St., Philadelphia, Pa.
MORGAN, Robert W. (*M* 1938) R. F. D. No. 1, Norwalk, Conn.
MORIARTY, John M. (*M* 1937) 1616 Baldwin Ave., Arcadia, Calif.
MORIN, A. R. (*A* 1938) 2115 Sherman, Oklahoma City, Okla.
MORIN, Romeo P. (*A* 1944) 3337-89th St., Jackson Heights, N. Y.
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MORRIS, Edward J. (*M* 1942) Mgr., Morris Engineering Co., 813 N. Calvert St., Baltimore 2, Md.
MORRISON, Chester B. (*M* 1931) Managing Director, York Shipley, Ltd., North Circular Rd., Hendon, London N.W. 2, England.
MORRISON, Frank (*M* 1943) Cloverdale, Ind.
MORRISON, W. Bruce (*A* 1942; *J* 1939) 1805 Northeast 27th Ave., Portland 12, Ore.
MORRISON, W. L. (*A* 1943) U. S. Navy, 956-97-15, U. S S. Jubilant A-M 255, F. P. O., New York, N. Y.
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MORROW, J. DeWitt (*A* 1938) Pres. and Mgr., The Warren Co., Inc., 614 Walker Ave., Houston 2, Texas.
MORSE, Clark T. (*Life Member*; *M* 1913) Pres., American Blower Corp., P. O Box 58, Roosevelt Park Annex, Detroit 32, Mich
MORSE, Elwin F. (*J* 1944) Field Engr., B F. Sturtevant Co., 933 Leader Bldg., Cleveland 14, Ohio
MORSE, Louis S., Jr. (*M* 1938; *J* 1936) Lt (jg) U.S.N.R., Naval Torpedo Station, Newport, R I.
MORSE, Robert D. (*M* 1936) 414 Vance Bldg., Seattle 1, Wash
MORTON, Harold S. (*M* 1931) Lt. Col., Ordnance Dept., U S Army, Tech. Div., Office Chief of Ordnance, Pentagon Bldg., Washington, D. C
MORTON, Paul S. (*A* 1943, *J* 1939) 609 Bangor Rd., Lawrence, Mich
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MOSS, Alfred W. (*A* 1945, *J* 1943) Mgr., Walter Woods, Ltd., 179 Bannatyne Ave., Winnipeg, Man., Canada
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MOULSDALE, Thomas DeWitt (*M* 1943) 818 Coleman, Easton, Pa
MUELLER, Ervin J. (*A* 1944) 1928 Lowell Ave., Butte, Mont
MUELLER, Harald C. (*M* 1936; *A* 1930) Gen. Mgr., The Powers Regulator Co., 2720 Greenview Ave., Chicago 14, Ill.
MUELLER, Harold P. (*M* 1936) Pres., L. J Mueller Furnace Co., 2005 W. Oklahoma Ave., Milwaukee, Wis
MUELLER, John E. (*M* 1937) Mgr., Comml Customer Dept., West Penn Power Co., Box 1736, Pittsburgh 30, Pa.
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MULHOLLAND, Daniel J. (*A* 1945) Supt., Consolidated Conditioning Corp., 460 S. Tenth Ave., Mt. Vernon, N. Y.
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MUNHALL, J. Kenneth (*A* 1944) Indus. Htg. Engr., Edwin L. Wiegand Co., 505 Delaware Ave., Buffalo 2, N. Y.
MUNIER, Leon L. (*M* 1919; *J* 1915) Pres., Wolff & Munier, Inc., 222 East 41st St., New York 17, N. Y.
MUNKELT, Frederick H. (*M* 1938) Vice-Pres., W. B. Connor Engineering Corp., 114 East 32nd St., New York 16, N. Y.

MURDOCK, Charles E. (*M* 1944) 1254 E. Third S., Salt Lake City 2, Utah.

MURDOCH, John P. (*M* 1937) Pres., John P. Murdoch Co., 3630 Haverford Ave., Philadelphia 4, Pa.

MURHARD, Erroll A. (*M* 1939) 2805 S.W. Greenview Ct., Portland 1, Ore.

MURPHREE, Robert L. (*A* 1940; *J* 1936) P. O. Box 864, Tuscaloosa, Ala.

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MURPHY, Daniel C. (*A* 1940) Sales Repr., C. A. Dunham Co., 214 Old Colony Bldg., Des Moines 9, Iowa.

MURPHY, Delacour I. (*M* 1941) 3027 Millsbrae Ave., Oakland 5, Calif.

MURPHY, Edward T.* (*Life Member*; *M* 1915) Vice-Pres., Carrier Corp., 300 S. Geddes St., Syracuse 1, N. Y.

MURPHY, Eugene F. (*A* 1944, *J* 1942) Instructor in Mech. Engrg., Dept. of Mech. Engrg., University of California, Berkeley 4, Calif.

MURPHY, Howard C.* (*M* 1923) Vice-Pres., American Air Filter Co., Inc., 215 Central Ave., Louisville, Ky.

MURPHY, Joseph R. (*M* 1934, *A* 1925) Terrace Ave., Riverside, Conn.

MURPHY, William W. (*M* 1930) Treas., W. W. Murphy Co., 424 Worthington St., Springfield, Mass.

MURRAY, G. F. J. (*M* 1944) 72 Holywell Hill, St. Albans, Herts, England.

MURRAY, Hayward G. S. (*M* 1942, *A* 1941, *J* 1936) Sales Engr., Canadian Comstock Co., Ltd., 80 King St. W., Toronto, Ont., Canada.

MURRAY, Thomas F. (*M* 1923) Senior Htg. and Vtg. Engr., New York State Dept. of Public Works, State Office Building, Albany, N. Y.

MURRAY, William A. (*M* 1944) 43 Fairlee Rd., West Hartford 7, Conn.

MURRAY, William W., Jr. (*M* 1944) Vice-Pres., Almirall & Company, Inc., 53 Park Pl., New York 7, N. Y.

MURSINNA, Gilbert P. (*A* 1939) 640 Taft St., Cincinnati 14, Ohio.

MUSE, Mayland H. (*A* 1944) Mgr. and Owner, Southern Welding Co., Box 454, Johnson City, Tenn.

MUSGRAVE, Merrill N. (*M* 1944, *A* 1935) Owner, Merrill N. Musgrave & Co., 2019 Third Ave., Seattle 1, Wash.

MUSSER, John M. (*M* 1942) 1314 Kimball, Richland, Wash.

MUTH, Arnold J. (*M* 1944) 868 W. Bethune Ave., Detroit 2, Mich.

MYER, Haydn (*A* 1940) Pres., Haydn Myer Co., Inc., 2224 Comer Bldg., Birmingham 3, Ala.

MYERS, George W. F. (*M* 1930; *A* 1928, *J* 1923) 476 Pasadena Ave., Webster Groves 19, Mo.

MYLER, William M., Jr.* (*M* 1937) Chief Engr., Janitrol Engrg. Dept., Surface Combustion, 400 Dublin Ave., Columbus 16, Ohio.

MYRICK, James W. H. (*M* 1944) Engr. and Owner, New England Air Conditioning Co., 244 Washington St. Boston 8, Mass.

MYTINGER, Kenneth L. (*M* 1943) Mytinger Air Conditioning Co., Inc., 101 Park Ave., New York 17, N. Y.

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NACHMAN, George P. (*M* 1938) Treas., The Spohn Heating and Ventilating Co., 1775 East 45th St., Cleveland, Ohio.

NAMAN, Israel A. (*J* 1940) P. O. Box 46, Bremerton, Wash.

NAROWETZ, Louis L., Jr. (*M* 1929; *A* 1912) Pres., Narowetz Heating & Ventilating Co., 1722 Washington Blvd., Chicago 12, Ill.

NASS, Arthur F. (*M* 1927) Pres., McGinness, Smith & McGinness Co., 527 First Ave., Pittsburgh 19, Pa.

NATHANSON, Max (*A* 1943) Chief Engr.-Gen. Mgr. and Owner, Canadian Armature Works, 6595 St. Urbain St., Montreal, Que., Canada.

NATION, Oslin (*A* 1944; *J* 1942) 9702 El Patio Drive, Dallas 18, Texas.

NATKIN, Alfred J. (*M* 1944) Partner and Houston Office Mgr., Natkin & Co., 1418 San Jacinto, Houston, Texas.

NAYLOR, James F., Jr. (*M* 1944) Dist. Mgr., The Trane Co., 3930 Lindell Blvd., St. Louis 8, Mo.

NEAL, James P. (*A* 1939) Capt., U. S. Army, 1428 Herschel Ave., Cincinnati 8, Ohio.

NEARINGBURG, Arthur (*A* 1938) Sales Engr., Sheldons, Ltd., 1221 Bay St., Toronto, Ont., Canada.

NEE, Raymond M. (*M* 1936) Principal Engr., Chemical Construction Corp., 30 Rockefeller Plaza, New York 20, N. Y.

NEILANS, John L. (*J* 1943) 330 Woodmount Ave., Toronto, Ont., Canada.

NEILER, Samuel G. (*Life Member*; *M* 1898) Owner, Neiler Rich & Co., 431 S. Dearborn St., Chicago, Ill.

NELSON, Arthur W.* (*M* 1944) Major, C. E., U. S. Army, 12 Sylvan Rd., Sharon, Mass.

NELSON, Axel A. (*A* 1942) Chief Engr., Federal Reserve Bank Bldg., 925-27 Grand Ave., Kansas City, Mo.

NELSON, D. W.* (*M* 1928) Assoc. Prof., Mech. Engrg., College of Engineering, University of Wisconsin, Mech. Engrg. Bldg., Madison, Wis.

NELSON, George O. (*M* 1923) Engr., Carstens Brothers, Ackley, Iowa.

NELSON, Harold M. (*M* 1937) Pres., H. M. Nelson & Co., Inc., 1223 Connecticut Ave., Washington 6, D. C.

NELSON, Herman W. (*Life Member*; *M* 1909) Chairman of the Board, The Herman Nelson Corp., 1824 Third Ave., Moline, Ill.

NELSON, Laurence K. (*M* 1940) Assoc., James M. Todd, Mech. Cons. Engr., 410 Maritime Bldg., New Orleans 12, La.

NELSON, Norris T. (*J* 1944) 951 O'Farrell, San Francisco 9, Calif.

NELSON, Richard H. (*A* 1933; *J* 1928) 1303 30th St., Moline, Ill.

NELSON, Roy O. (*M* 1938) 5636 N. Bernard St., Chicago 45, Ill.

NELSON, Walter L. (*A* 1944) 2225 Quincy St. N. E., Washington, D. C.

NESBITT, Albert J.* (*M* 1921) Pres., John J. Nesbitt, Inc., State Rd. and Rhawn St., Holmesburg, Philadelphia 36, Pa.

NESMITH, Oliver E. (*A* 1936) 107 Warner, Bloomington, Ill.

NESS, William H. C. (*M* 1931) Secy. and Gen. Mgr., Master Fan Corp., 1323 Channing St., Los Angeles 21, Calif.

NESSELL, Clarence W. (*M* 1937) Asst. to Vice-Pres., Minneapolis-Honeywell Regulator Co., 1101 Vermont Ave. N.W., Room 405, Washington, D. C.

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NEUBAUER, Edwin W. (*M* 1939) Engr., Campbell Norquist & Co., 1127 S.W. Morrison St., Portland 5, Ore.

NEVIN, J. F. (*M* 1944) Capt., c/o Chief Engineer, Eastern Command, H. A. India.

NEWBY, Ira P. (*M* 1941) 6325 Bryan Pkwy., Dallas 14, Texas.

NEWMAN, Harold E. (*M* 1938) Owner and Mgr., J. F. Dickinson Co., 716 10th St., Modesto, Calif.

NEWPORT, Charles F. (*Life Member*; *M* 1906) 10001 Longwood Drive, Chicago, Ill.

NEWTON, Albert E. (*A* 1943) Chief Engr., Chrysler Corp., Airtemp Div., Dayton 1, Ohio.

NEWTON, Alwin B. (*M* 1938) Engrg. Dept., Chrysler Corp., Airtemp Div., 1119 Leo St., Dayton 1, Ohio.

NEWTON, David A. (*M* 1944) 55 25th St. N. W., Atlanta, Ga.

NICHOLLS, John M. (*M* 1939) Robbins Gamwell Corp., 68 West St. Pittsfield, Mass.

NICHOLS, Howard R. (*A* 1943) 5416 Park Pl., Minneapolis, Minn.

NICHOLS, Lawrence J. (*A* 1944) 2074 Greenway N., Columbus 15, Ohio.

NICHOLSON, Sterling J. (*A* 1941) Pres., Nicholson, Inc., Box 317, Durhan, N. C.

NICKLE, Arthur J. (*A* 1936) Sales Engr., Darling Brothers, Ltd., 140 Prince St., Montreal, Que., Canada.

NICOLL, Scott F. (*M* 1939) Mech. Engr., York Corp., York, Pa.

NICOLS, John A. (*M* 1941) 440 Kimball Turn, Westfield, N. J.

NIEMOELLER, Arthur R. (*A* 1943) Sales Engr., 5817 Itaska St., St. Louis 9, Mo.
NIESKE, George F. (*M* 1943) 794 Bee St., Meriden, Conn.
NIESSE, Joe H. (*M* 1938) 5837 Winthrop Ave., Indianapolis, Ind.
NIGHTINGALE, George F. (*A* 1931) 2136 Linneman St., Glenview, Ill.
NOBBS, Walter W. (*M* 1919) 50 Fairhazel Gardens, London N.W. 6, England.
NOBIS, Harry M. (*M* 1914) 1827 Stanwood Rd., East Cleveland 12, Ohio.
NOBLE, Milner (*M* 1940) Pres., Aerofin Corp., 410 S. Geddes St., Syracuse, N. Y.
NOLAN, James J., Jr. (*M* 1939) 4024 Calvert St N W., Washington 7, D. C.
NOLAN, John J. (*M* 1943) Prop., J. J. Nolan & Co., 78 Washington Ave., Memphis, Tenn.
NOLL, Michael P. (*M* 1944) Dunlavy Court North, Wilshire Village, Houston 6, Texas.
NOLL, William F. (*M* 1924) Prop, William F. Noll, 629 North 27th St., Milwaukee 8, Wis.
NOORD, Donald F. (*A* 1945; *J* 1941) Browntown Rd., Front Royal, Va.
NORAIR, Henry (*M* 1938) Pres., Norair Engineering Corp., 1114 22nd St., N W., Washington 7, D. C.
NORBY, Karl H. (*M* 1943; *A* 1938) 3237 29th W., Seattle 99, Wash.
NORCROSS, Ivan F. (*A* 1945) Mech Engr, E. L. E. Co, 124 W Fourth St, Los Angeles, Calif.
NORDIN, John G. (*A* 1944) Service Engr, Southern California Gas Co., 725 Channing St, Los Angeles 21, Calif
NORDINE, Louis F. (*M* 1914) 806 Dale Drive, Silver Springs, Md
NORDSTROM, Fielder A. (*S* 1943) 420 Oak St S E., Minneapolis, Minn.
NORFOLK, Leslie W. (*A* 1941, *J* 1939) Lt Col, Royal Engineers, 42 Hampden St, Nottingham, England.
NORMAN, Edward A., Jr. (*M* 1944) 1926 Andover Rd, Columbus 8, Ohio
NORMAN, George C. (*M* 1944) Engr, Kotthan & Johnson, 27 Washington St, Newark 2, N J
NORRINGTON, Walter L. (*A* 1943, *J* 1938) Capt., Ordnance Dept., 305 Barbara Fritchie Beverly Plaza, Alexandria, Va.
NORRIS, William P. (*M* 1944; *J* 1938) Assoc Engr., U. S. District Engineers Office, Room 611 Santa Fe Bldg, Galveston, Texas.
NORTE, Wilbur R. (*J* 1942) 2785 Bailey Rd, Cuyahoga Falls, Ohio.
NORTH, Clarence P. (*M* 1942) Chief Engr Campbell Heating Co., P. O. Box 833, Des Moines 4, Iowa
NORTH, Sam L. (*A* 1942) Partner, North Brothers, P. O Box 252, Atlanta 1, Ga.
NORTH, William R. (*A* 1940) Sales Engr 27 S Gay St., Baltimore 2, Md.
NORTON, John A. (*M* 1940) 136 Hanna Rd, Leaside, Ont., Canada.
NORTON, L. Ivan (*A* 1941) Engr, Eveready Plumbing & Heating Co., 514 W Main St, Washington, Iowa.
NOTKIN, James B. (*M* 1944) 2013 East 63rd St, Seattle 5, Wash.
NOTTAGE, Herbert B.* (*A* 1945; *J* 1943) 118 Main St., Manchester, Conn.
NOTTBERG, Gustav (*A* 1933) Vice-Pres, U S Engineering Co., 914 Campbell St., Kansas City, Mo.
NOTTBERG, Henry J. (*M* 1919) Pres., U S. Engineering Co., 914 Campbell St., Kansas City, Mo.
NOTTBERG, Henry, Jr. (*J* 1937) Secy., U S Engineering Co., 914 Campbell St, Kansas City, Mo
NOVAK, Charles J. (*A* 1944) 5943 Carrollton Ave, Indianapolis, Ind.
NOVOLANSKY, Sydney I. (*J* 1943) 6036 F/S. #8 Repair Depot, R. C. A. F., Winnipeg, Man Canada
NOWITZKY, Herman S. (*A* 1931) 821 Llewellyn Ave., Norfolk, Va.
NOYES, Richard R. (*J* 1938) Field Engr., Canadian Sirocco Co, Ltd., 630 Dorchester St., W., Montreal, Que., Canada
NUGENT, Arthur W. (*M* 1945) Spec. Repr, McQuay, Inc., 14th and G Sts. N W, Washington, D. C
NUNLIST, F. J., Jr. (*M* 1943) 528 North 19th St., Milwaukee 3, Wis.
NUSBAUM, Lee* (*M* 1915) Owner, Pennsylvania Engineering Co., 1119-21 N. Howard St., Philadelphia, Pa.
NUSBAUM, S. Richard (*A* 1944; *J* 1940) Mgr., Pennsylvania Engineering Co., 1119-21 N. Howard St., Philadelphia, Pa.
NUSSBAUM, Otto J. (*A* 1944) Chief Engr., Kramer Trenton Company, 626 Brunswick Ave., Trenton 5, N. J.
NUTTING, Arthur* (*M* 1940) Chief Engr, American Air Filter Co., 215 Central Ave., Louisville, Ky.
NYE, L. Bert, Jr. (*A* 1943, *J* 1936) McLean, Va.
NYLAND, John A. (*A* 1944) Owner, Nyland Sheet Metal Co., 2323 West 10th St, Indianapolis 8, Ind.
NYQUIST, John D. (*J* 1942, *S* 1941) 1535 Bever Ave, Cedar Rapids, Iowa.

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OAKLEY, LeRoy W. (*M* 1937) Owner, L. W. Oakley Sales Co, 408 W. Clinch Ave, Knoxville, Tenn
OAKS, Orion O. (*M* 1917) 119 Oak Ridge Ave, Summit, N J.
O'BANNON, Lester S.* (*M* 1928) Research Engr, Kentucky Agricultural Experiment Station, University of Kentucky, Lexington, Ky.
OBERG, H. C. (*A* 1933) 1362 W. Minnehaha St, St Paul, Minn
OBERSCHULTE, Richard H. (*M* 1944; *J* 1938) Sales and Field Mgr., D. T Randall & Co, 404 Boulevard Bldg, Detroit 2, Mich.
O'BRIEN, Thomas J. (*M* 1943) Partner, Allied Architects and Engrs, 1030 Exchange Bldg, Memphis, Tenn
O'CONNELL, Thomas D'Arcy (*A* 1942) Pres., Thomas O'Connell, Ltd, 1169 Ottawa St., Montreal, Que, Canada
O'DANIEL, James A. (*M* 1942) Owner, Maple City Furnace Co, 603 S Main St, Monmouth, Ill
O'DONNELL, Lawrence D. (*A* 1944) Mgr and Partner, McCarthy's Sheet Metal Works, 113 N Eighth St, Vincennes, Ind
O'DOWER, Hugh J. (*A* 1938) Field Engr, Vilter Manufacturing Co, 114 West 10th St, Kansas City, Mo
OELGOETZ, J. F. (*M* 1938) Sole Owner, J F Oelgoetz Co, 3365 N. High St, Columbus, Ohio
OFFEN, Ben (*M* 1928) Owner, B Offen & Co, 343 S. Dearborn St, Chicago, Ill
OFFNER, Alfred J.* (*M* 1922), (2nd Vice-Pres, 1944, Treas, 1935-38; Council, 1935-44) Cons, Engr, 139 East 53rd St, New York 22, N Y
O'GORMAN, John S., Jr. (*A* 1934) Mgr, Detroit Office, Johnson Service Co, 230 E. Alexandrine Ave, Detroit 1, Mich
OLD, William H. (*M* 1937) Asst Mgr, Glanz & Killian Co., 1761 Forest Ave W, Detroit 8, Mich
OLDS, Dean (*M* 1944) Chief Engr, Gas and Oil Div, The Coleman Lamp & Stove Co, 250 N St. Francis, Wichita, Kan
OLLESHEIMER, Louis T. (*A* 1945) 2539 Woodward Ave, Detroit 1, Mich
OLSEN, Carlton F. (*A* 1925; *J* 1920) 1000 West 100th St, Chicago 43, Ill
OLSEN, Gustav E. (*M* 1930) 68-09 Beach Channel Drive, Arverne, L. I, N. Y
OLSON, Arthur A. (*M* 1944) Vice-Pres and Treas, Lee Engineering Co, 1102 Union National Bank Bldg., Youngstown, Ohio
OLSON, Barney (*A* 1929) Barney Olson, Inc, 122 S Michigan Ave., Chicago, Ill.
OLSON, Erling O. (*A* 1943) 4221 Oakland Ave, Minneapolis 7, Minn.
OLSON, Eugene O. (*M* 1942) c/o Delavan Engineering Co., 414–12th St., Des Moines, Iowa.
OLSON, Milton J. (*A* 1941; *J* 1937) 5830 Hickory St., Omaha, Nebr.
OLSON, Nat I. (*A* 1944) Owner and Gen Mgr, United Blower Co., Inc., 193 Centre St., New York 13, N. Y.
OLVANY, William J. (*Life Member*; *M* 1912) Pres., William J. Olvany, Inc., 100 Charles St., New York 14, N. Y

O'NEILL, James W. (*M* 1929; *A* 1927; *J* 1925) 55 Highview Crescent, Toronto, Ont., Canada.
O'NEILL, Joseph F. (*M* 1944) Mech. Engr., Trane Co. of Canada, Ltd., 4 Mowat Ave., Toronto, Ont., Canada.
OONK, William J. (*M* 1937) 4548 Red Bud Ave., St. Louis 15, Mo.
OOSTEN, Louis S. (*A* 1944; *J* 1938) 725 Case St., Evanston, Ill.
OPPERMAN, Everett F. (*A* 1944) 788 East Main St., Stamford, Conn.
ORABELLA, Michael J. (*A* 1944) 13304 First Ave., East Cleveland, Ohio.
OREAR, Andrew G. (*M* 1942) Pres., Trade-Wind Motorfans, Inc., 5725 S. Main St., Los Angeles 37, Calif.
O'REAR, Lawrence R. (*M* 1934) Pres, Midwest Plumbing & Heating Co., 2450 Blake St., Denver, Colo.
ORGELMAN, George H. (*J* 1942, *S* 1940) U. S. Army, 10 Pearl St, Danbury, Conn
ORMISTON, Jack B. (*A* 1940) Owner, Ormiston Plbg. & Htg. Co., 105 Manning Ave., Yonkers, N. Y.
ORMSBY, H. Kingsley, Jr. (*M* 1944) Owner, Syracuse General Sales Co., 511 E. Raynor Ave., Syracuse 1, N. Y.
ORR, Weldon J. (*A* 1944) 3559 St. Famille St., Montreal, Que., Canada
OSBERGER, T. L. (*A* 1944) Mgr, T L. Osberger Co., 200 N. Division Bldg, Grand Rapids 2, Mich.
OSBORN, Wallace J. (*A* 1927) 1029 Old Post Rd, Fairfield, Conn.
OSBORNE, G. H. (*M* 1922) Managing Dir, The Ventilating & Blow Pipe Co, Ltd, 714 St Maurice St., Montreal 3, Que., Canada
OSBORNE, James M. (*A* 1944) Engr, The Herman Nelson Corp., 1108-16th St. N.W, Washington 6, D. C.
OSBORNE, Stanley R. (*M* 1939) Promotion Engr., Chase Brass & Copper Co, Waterbury 91, Conn.
O'SHEA, John J. (*A* 1941) 714 Greenview Ave. N.E., Atlanta, Ga.
OSTER, William P. (*M* 1940) Vice-Pres, Equitable Equipment Co., Inc., 410 Camp St, New Orleans 12, La.
OSTROM, Eric W. (*M* 1937) c/o John Ericssonsgatan, Stockholm, Sweden. (Present address unknown).
OTIS, Alfred L. (*A* 1944) 43 N Main, West Hartford, Conn.
OTT, M. Earl (*M* 1942) Sole Owner, Sheet Metal Service, 23 East 115th St., Chicago 28, Ill
OTT, Oran W. (*M* 1925), (Council, 1934-36) Cons. Mech. Engr, Washington Bldg, Los Angeles, Calif.
OTTO, Robert W. (*M* 1941) 2147 Carroll Ave, St. Paul, Minn.
OTTS, John G. (*A* 1942) Capt., Ordnance, c/o Dr. W. W. Carson, Ft. Valley, Ga
OTTUM, Alvin L. (*A* 1943) 504 South 50th St, Omaha 6, Nebr.
OURUSOFF, Leon* (*M* 1931) Mgr. of Utilization, Washington Gas Light Co., 1100 H St N.W., Washington, D. C.
OUWENEEL, William A. (*M* 1937) Chief Engr., Belcher Oil Co., 1221 Biscayne Blvd., Miami 30, Fla.
OVERTON, Sidney H. (*M* 1929) Repr., Ideal Boiler & Radiators, Ltd., and American Radiator & Standard Sanitary Corp., P. O. Box 5985, Johannesburg, South Africa.
OWEN, Charles E. (*M* 1941) 1218 S. Thompson St., Carbondale, Ill.
OWEN, Jeff Davis (*M* 1937) 4070 East Blvd, Culver City, Calif.
OWEN, W. Harold (*M* 1943) Engr., Harry Cooper Supply Co., 223 E. Water, Springfield, Mo.

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PAETZ, Herbert E. (*M* 1922) Div. Sales Mgr., American Blower Corp., 632 Fisher Bldg., Detroit 2, Mich.
PAGE, Arvin (*M* 1935) Chief Engr., The Bahnson Co., Salem Station, Winston-Salem, N. C.
PAGE, Vernon C. (*A* 1936) Supt., Navy Fab. Div., Fitzgibbons Boiler Co., Inc., Oswego, N. Y.
PAINE, H. Allan (*J* 1940) Lt. (jg) U.S.N.R., Apt. 101, Newell Apts., Kingwood at Fourth, Brainerd, Minn.
PAINTER, D. Howard (*M* 1943) 8031 Manor Rd., Rt. 7, Kansas City 5, Mo.
PALMER, Elbert M. (*A* 1944) Mgr., Govt. Dept, Kewanee Boiler Corp., Kewanee, Ill.
PALUMBO, Bernard F. (*A* 1943; *J* 1941) 62 Foster Square, Bridgeport 8, Conn.
PANGBORN, Clarence A. (*M* 1944) Mech Engr., 8614 Wallingford Ave., Seattle, Wash.
PAQUET, Jean-Marie (*A* 1940; *J* 1936) 9 Bois Joli, Quebec, Que., Canada.
PARENT, Harold M. (*M* 1938) Partner, Parent & Kirkbride, N.W. Cor. Fourth and Locust Sts, Philadelphia 6, Pa.
PARIZEAULT, Rolland (*A* 1944) Supt., J. & C Brunet, Ltd., 1095 St Lawrence Blvd., Montreal, Que, Canada.
PARK, Harold E. (*A* 1938; *J* 1936) Sales Engr, Shaw-Perkins Manufacturing Co., 1645 Oliver Bldg, Pittsburgh 22, Pa
PARK, J. Frank (*M* 1937; *A* 1936; *J* 1930) Pres, Western Air & Refrigeration, Inc., 1234 S Grand Ave., Los Angeles 15, Calif.
PARK, Nicholas W. (*M* 1936) 509 Jericho Rd, Abington, Pa.
PARKER, Herbert E. (*M* 1943) 74 Sealund Rd., North Quincy, Mass.
PARKER, Richard A. (*A* 1938) Lt., U. S Army, Asst. Vicinity Maintenance Engr., c/o Postmaster, San Francisco, Calif.
PARKES, C. H. (*M* 1944) Owner, Distributor of Heating & Pumping Equipment, 101½ W Chestnut, P. O Box 947, Dodge City, Kans.
PARKINSON, John S.* (*A* 1940) Lt. Comdr, U S N.R, 8907 Oneida Lane, Bethesda, Md
PARKS, Charles E. (*M* 1937) Pacific Coast Mgr, Ilg Electric Ventilating Co, 816 W. Fifth St, Los Angeles 13, Calif.
PARROTT, Lyle G. (*M* 1922) Cons. Engr, Snyder & McLean, 2214 Penobscot Bldg, Detroit 26, Mich.
PARSONS, John H. (*A* 1942) 4119 Fourth Ave, Los Angeles, Calif
PARSONS, Roger A. (*A* 1942, *J* 1933) Htg Engr Board of Water & Electric Light Commissioners, 114-16 W. Ottawa St, Lansing, Mich.
PARTLAN, Robert L. (*J* 1940) Pres, Partlan Sheet Metal Works, 14265 Arlington Ave., Detroit 12, Mich
PARVIS, Ralph S. (*M* 1938) 1208 Delaware Ave, Wilmington 19, Del
PASEK, Leonard E. (*M* 1944) Development Engr, Kimberly-Clark Corp., 1426 G St. N W, Washington 5, D. C
PASTEUR, Hugh W. (*M* 1943) Export Mgr, J E Hall, Ltd., Dartford, England.
PASTOR, John C. (*M* 1938) Design Engr, Room 818, Graham Bldg, Jacksonville 2, Fla
PATORNO, Sullivan A. S. (*M* 1923) Cons. Engr, 101 Park Ave., New York 17, N Y
PATTERSON, Frank H. (*M* 1942) 9201 Boleyn Ave., Detroit 24, Mich
PATTERSON, Granville P. (*M* 1939) Vice-Pres, W. B. Haggerty, Inc., 805 Morgan St., P. O. Box 2971, Tampa 1, Fla.
PATTERSON, James L. (*M* 1943) 1747 Jane St, Weston, Ont., Canada.
PATTERSON, Joseph C., Jr. (*M* 1944) Marine Engr., La-Del Conveyor & Manufacturing Co, 410 Washington Bldg., 15th and G, Washington, D. C.
PATTERSON, Ralph A. (*A* 1943) Sales Mgr., Bell & Gossett Co, Morton Grove, Ill.
PAUL, Dean W. (*A* 1944) 140 Cabrini Blvd., New York 33, N. Y.
PAUL, Donald I. (*M* 1936; *J* 1932) Gurney Foundry Co., Ltd., 4 Junction Rd., Toronto, Ont., Canada.
PAULDING, Lewis G. (*M* 1944) Supvsr., Long Island Lighting Co., 250 Old Country Rd., Mineola, L. I., N. Y.
PAULEY, Robert D. (*A* 1944; *J* 1940) Development Engr., Weyerhaeuser Timber Co., Longview, Wash.

PAVEY, Charles A. (*M* 1937) Dist. Mgr., B. F. Sturtevant Co., 812 Michigan Bldg., Detroit 26, Mich.
PAWKETT, Lawrence S. (*A* 1938) Major, Army Air Corps., 16 E. Bells Mill Rd., Chestnut Hill, Philadelphia 18, Pa.
PAYNE, Raymond L. (*A* 1944) Owner, Air Flow Heating Co., 1409 Platte St., Denver, Colo.
PAYTON, John F. (*M* 1944) 25 West 16th St., Indianapolis, Ind.
PEACOCK, Glenn S. (*M* 1939) 111 Elmont St., Crafton P. O. No. 5, Pittsburgh, Pa.
PEACOCK, Herbert (*M* 1940) Carrier Corp., 405 Lexington Ave., New York 17, N. Y.
PEARCE, Edward A. (*M* 1942) 55 Blenheim Rd, North Harrow, Middlesex, England.
PEARSON, Fred L. (*M* 1942) Mech. Engr., Smith, Hinchman & Grylls, 800 Marquette Bldg., Detroit, Mich.
PEART, Allen M. (*A* 1937) Dist Mgr, Minneapolis-Honeywell Regulator Co., 637 Craig W., Room 812, Montreal, Que., Canada
PECK, Henry E. (*A* 1938) 73 Potter Pl, Fairport, N. Y.
PECK, Robert E. (*M* 1943) Vice-Pres., The Peck Hannaford & Briggs Co., Cor. Sycamore and Court Sts, Cincinnati, Ohio.
PEEBLES, John K., Jr. (*A* 1925, *J* 1924) 1708 Park Ave., Richmond 20, Va.
PEGG, Edward H. R. (*A* 1943) 7 Morningside Pl, Cranford, N. J.
PEISER, Maurice B. (*J* 1937) 1st Lt, A. C., 18th Weather Squad., Det. 102, A. P. O. 639, c/o Postmaster, New York, N. Y
PELLEGRINI, Louis C. (*M* 1939) 6549 Murdoch St., St. Louis, Mo.
PELLER, Leonard (*A* 1942, *J* 1934) Lt. (jg) U.S N R., 1422 Carmen Ave, Chicago, Ill.
PELLMOUNTER, Thomas (*A* 1936) 3308 Euclid Ave, Kansas City 3, Mo
PELLMOUNTER, Thomas V. (*J* 1938) 2nd Lt, 55th Ordnance Co, A. P. O 309, Ft. Lewis, Wash.
PENN, Lester W. (*M* 1942) 3916 Shreve Ave, St Louis 15, Mo
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TAYLOR, R. F. (*M* 1915) Cons Engr, 910 Bankers Mortgage Bldg, Houston, Texas
TAYLOR, Robert B. (*M* 1944, *J* 1938) Repr. Buffalo Forge Co, 1303 Standard Bldg, Albany 7, N Y.
TAYLOR, Thomas E. (*M* 1942, *J* 1937) Mech Engr., 707 Spalding Bldg, Portland 4, Ore
TAZE, D. L. (*M* 1931) Branch Mgr, American Blower Corp, 1302 Swetland Bldg, Cleveland 15, Ohio
TAZE, Edwin H. (*M* 1937) Branch Mgr, American Blower Corp., 620 Court Sq Bldg, Baltimore 2, Md.
TEASDALE, Lawrence A. (*M* 1926) Mgr, Div of Htg. and Lighting, c/o Yale University Service Bureaus, 20 Ashmun St., New Haven, Conn.
TEELING, George A. (*M* 1930) Cons Engr, 1 Columbia Pl, Albany, N Y
TELGEMEIER, A. H. (*A* 1944) 2307 E. Meyer Blvd, Kansas City, Mo.
TEMPLE, W. J. (*M* 1931) 1215 Reed Ave, Kalamazoo 24, Mich.
TEMPLIN, Charles L. (*M* 1921) Pres., Carrier-Atlanta Corp, 348 Peachtree St, Atlanta 3, Ga.
TENKONOHY, R. J. (*M* 1923) 1432 Rockway, Lakewood 7, Ohio
TENNANT, Raymond J. J. (*A* 1929) Chief Engr., Pittsburgh Business Properties, Inc, 2237 Oliver Bldg, Pittsburgh, Pa
TENNEY, Dwight (*M* 1932) 33 Summit Rd., Verona, N. J.
TERHUNE, Ralph D. (*A* 1936) 277 Orchard Pl, Ridgewood, N. J.
TERRANCE, Emmett H. (*M* 1944) Engr -Dist Mgr., Canadian Sirocco Co., Ltd., 630 Dorchester St. W., Montreal, Que, Canada
TERRY, Matson C. (*M* 1936) 508 Old York Rd., Abington, Pa.
TERRY, S. W. (*M* 1941) Pres., Aladdin Heating Corp., 2222 San Pablo Ave., Oakland, Calif.
THACKER, John E. (*M* 1944) Htg.-Vtg. Engr., Vauxhall Motors, Ltd., Luton, Bedfordshire, England.
THAYER, Harding H. (*A* 1943) 920 Maryland Ave., Pittsburgh 6, Pa.
THEISS, Ernest S. (*M* 1944; *A* 1941; *J* 1940) 2645 University Drive, Durham, N. C.
THEOBALD, Art. (*A* 1937) 116½ S. Kings Rd, Los Angeles, Calif
THEORELL, Hugo G. T.* (*Life Member*; *M* 1902) Cons. Engr., Hugo Theorells Ingeniorsbyra, Skoldungagatan 4, Stockholm, Sweden.
THINN, Christian A.* (*M* 1921) Mgr. of Service, C. A. Dunham Co., 450 E. Ohio St., Chicago, Ill.
THOM, George B. (*M* 1937) Chm. Mech Engrg Dept., Swarthmore College, Swarthmore, Pa.
THOM, Herbert C. S. (*M* 1944) 6130 18th Rd. N, Arlington, Va
THOMAN, Estell O. (*A* 1938) Air Cond -Htg. Engr, Higgins Industries, Inc., 1755 St. Charles Ave., New Orleans, La.
THOMAS, Bernard A. (*A* 1938) 405 E Idlewild Ave., Tampa 4, Fla
THOMAS, Ernest R. (*M* 1942) 3830 Marquette, Houston, Texas
THOMAS, Frank M. (*M* 1943) Dist Sales Mgr, Oklahoma Natural Gas Co, 217 W. Randolph, Enid, Okla
THOMAS, Glegge (*M* 1936) Office Mgr, Clarage Fan Co, 723 Albee Bldg, Washington, D C
THOMAS, L. G. Lee (*M* 1934) Vice-Pres, Economy Pumps, Inc, 1000 Weller Ave, Hamilton, Ohio
THOMAS, Melvern F. (*Life Member*, *M* 1909) 74 Rivercrest Rd, Toronto, Ont, Canada.
THOMAS, Ralph C. (*A* 1938) 819 Westover Ave, Norfolk 7, Va
THOMAS, R. H. (*Life Member*, *M* 1920) 765 Ivy Ave., Glendale, Ohio
THOMAS, R. L. (*A* 1943) Field Engr, Fairbanks-Morse & Co, 13th & Liberty, Kansas City, Mo
THOMPSON, Edward B. (*A* 1938) 1198 Coronado Ave, Cincinnati, Ohio
THOMPSON, Frank (*M* 1935) 107 Quebec St, Sherbrooke, Que, Canada
THOMPSON, John (*M* 1942) Administration Building Engr, Hydro-Electric Power Commission of Ontario, 620 University Ave, Toronto, Ont, Canada
THOMPSON, Nelson S.* (*Life Member*, *M* 1917) 3206 Oliver St N W, Washington 15, D C
THOMPSON, Paul J. (*A* 1944) Branch Mgr, National Radiator Co, Room 236, 401 N Broad St, Philadelphia 8, Pa
THOMPSON, William D. (*M* 1944) Mgr, Industrial Div, Laclede Gas Light Co, 1017 Olive St, St Louis 1, Mo
THOMSEN, N. B. (*M* 1938) Vice-Pres, Macdonald Engineering Co, 188 W Randolph St, Chicago 1, Ill
THOMSON, Thomas N.* (*Life Member*, *M* 1927) 37 Irwin Pl, Huntington, L I, N Y.
THORNBURG, Harold A. (*M* 1932; *J* 1929) 1911 Dorchester Rd, Brooklyn, N Y
THORNTON, Thaddeus L. (*M* 1937) 37 Perry St, Belleville, N J.
THORPE, Wano E. (*M* 1943) Estimator-Engr, J. J Nolan & Co, 78 Washington Ave., Memphis, Tenn.
THRUSH, Homer A. (*M* 1918) Pres, H A Thrush & Co, 21 E. Riverside Drive, Peru, Ind
THULMAN, Robert K.* (*M* 1938) 6505 Ridgewood Ave., Chevy Chase, Md
THUNEY, Francis M. (*A* 1939, *J* 1936) Exec Engr., Minneapolis-Honeywell Regulator Co., 2753 Fourth Ave S, Minneapolis 8, Minn.
TICHENOR, Leslie R., Jr. (*A* 1942) Lt., Range Office, Main Post, Ft. Benning, Ga
TIDMARSH, Patrick M. (*M* 1938) Gen. Mgr, Tidmarsh Engineering Co., P. O Box 2425, Tucson, Ariz
TIERNEY, Lawrence J. J. (*A* 1942) Owner, L. J. Tierney Co, 10 High St, Boston, Mass
TILFORD, Leo A. (*M* 1941) Owner-Mgr., Leo A Tilford Co., 1230 Francis St, Jackson, Mich.
TILLER, Louin (*A* 1935; *S* 1933) Engr., Velocity Steam Production Engrg, 38 S. Dearborn St, Chicago 3, Ill
TILLOTSON, John J. (*J* 1943) Engr., Westinghouse Electric Elevator Co., 150 Pacific Ave., Jersey City 4, N. J
TILTZ, Bernard E. (*M* 1930) 22 Villa Rd., Larchmont, N. Y.

TIMMINS, W. W. (*M* 1937) Dist. Mgr., Canadian Powers Regulator Co., Ltd., University Tower Bldg., Montreal, Que., Canada.
TIMMIS, Pierce (*M* 1920) Industrial Engr., United Engineers and Constructors, Inc., 1401 Arch St., Philadelphia, Pa.
TIMMIS, William W. (*M* 1933; *A* 1925) Lt. Comdr, U. S. Navy, 103 Geo. Mason Drive, Arlington, Va.
TODD, Malcolm McM. (*M* 1942) Owner, Malcom Todd, 302 Bay St., Toronto, Ont., Canada.
TODD, Meryl L. (*M* 1940; *J* 1936) Partner, M. L. Todd and Associates, 1111 Independence Ave., Waterloo, Iowa.
TOENSFELDT, Ralf (*M* 1940) 6311 Waterman Ave., St. Louis 5, Mo.
TOLERTON, Hill A. (*A* 1944) Contracting Engr, Lee Engineering Co., 1102 Union National Bank Bldg., Youngstown, Ohio.
TONRY, Robert C. (*M* 1936) Mgr., Wiedebusch Plumbing & Heating Co, 511 First St., Fairmont, W. Va.
TOONDER, Clarence L. (*M* 1933) 13391 Marlowe, Detroit, Mich.
TOROK, Elmer (*M* 1936) Pres., East Tennessee Sheet Metal Works, Inc., P. O. Box 421, Bristol, Tenn.
TORR, T. W. (*M* 1933) 205 Green St., Dowagiac, Mich.
TORRENS, George B. (*M* 1945) 250 West St., Leominster, Mass
TOULOUKIAN, Yeram S. (*J* 1944; *S* 1939) 12 N. Chauncy Ave., West Lafayette, Ind.
TOUPIN, Valerien (*M* 1944) Htg Engr., Municipal Bldgs. Dept., City of Montreal, 720 De Fleurimont, Montreal 10, Que, Canada.
TOUTON, Rush D. (*M* 1933) Tech. Dir., Bayuk Cigars, Inc, Ninth and Columbia Ave., Philadelphia, Pa.
TOWER, Elwood S. (*M* 1930) Engr, 213 Investment Bldg., Pittsburgh 22, Pa
TOWLE, Philip H. (*A* 1942; *J* 1938) 640 Tennyson Ave., Palo Alto, Calif.
TOWNER, Charles E. (*A* 1944) 9141 S Mayfield, Oak Lawn, Ill
TOWNSEND, John M. (*M* 1942) Engr, Minneapolis-Honeywell Regulator Co, 420 N Fifth St., Minneapolis, Minn.
TRACY, John W. (*M* 1945) 431 Marietta Pl, Pittsburgh 16, Pa.
TRACY, William E. (*A* 1943; *J* 1938) Lt. (jg) U.S.N.R., (Address unknown).
TRAMBAUER, Charles W. (*A* 1941) 179-17 134th Ave., Springfield Gardens, N. Y
TRANE, Reuben N.* (*M* 1915) Pres, The Trane Co, La Crosse, Wis.
TRAUGOTT, Mortimer (*A* 1930) Bryant Air Conditioning Corp., 915 N. Front St, Philadelphia 23, Pa.
TRAUTMAN, Fred L. (*M* 1944) Owner, Trautman Engineering Co., 1122 East 17th Ave, Denver, Colo.
TRAYNOR, Harry S. (*A* 1942; *J* 1937) Major, 200 Outer Drive, Oak Ridge, Tenn
TREADWAY, J. Quentin (*A* 1936; *J* 1932) Dist. Sales Mgr., Clarage Fan Co., 210 Reynolds Arcade, Rochester, N Y.
TREFTS, John C., Jr. (*M* 1944) Vice-Pres., Farrar & Trefts, Inc., 20 Milburn St., Buffalo 12, N. Y.
TRICKLER, Earl E. (*M* 1942) 8219 Kenwood Ave., Chicago 19, Ill.
TRIEGEL, Erich V. (*M* 1942) Mech. Engr., Francisco & Jacobus, 511 Fifth Ave., New York, N. Y.
TRIGGS, Fred E. (*M* 1938) 3901 Second St, Des Moines, Iowa.
TRIMBLE, John I. (*M* 1945) 2038 Bedford Rd., Columbus 8, Ohio.
TROLLER, T. H.* (*M* 1943) 1250 Sunsetview Drive, Akron, Ohio.
TROSTEL, Otto A. (*M* 1935) Route 2, Thiensville, Wis.
TROUP, John D. (*M* 1938) Managing Dir., John D. Troup, Ltd., 90 High Holborn, London, W.C. 1, England.
TROWBRIDGE, Carl Y. (*A* 1944) Secy.-Treas - Gen. Mgr., Ray F. Fischer Co., 323 N.W. Tenth St., Oklahoma City, Okla.
TRUMBO, S. M. (*A* 1926) Engr., Buffalo Forge Co., 20 N. Wacker Drive, Chicago 6, Ill.
TUCKER, Frank N. (*M* 1926) 239 Whaley St., Freeport, N. Y.
TUCKER, Leonard A. (*M* 1935) 220 Buttonwood Way, Glenside, Pa.
TUCKER, Thomas T. (*M* 1938; *A* 1936) Chief Engr., Armor Insulating Co., 800 Forrest St. N.W., Atlanta, Ga.
TUCKERMAN, George E. (*M* 1932) 502 Rodman Ave., Jenkintown, Pa.
TUPPER, Edward B. (*J* 1944; *S* 1941) 3248 Park Ave., Minneapolis, Minn.
TURLAND, Charles H. (*M* 1934; *A* 1930) Sales Engr., R. E. Johnston Co., Ltd., 1070 Homer St., Vancouver, B. C., Canada.
TURNER, Edmond S. (*A* 1939) Partner, William S. Turner & Co., 618 Pacific Bldg., Portland 4, Ore.
TURNER, George W. (*M* 1944) Dist. Repr., Minneapolis-Honeywell Regulator Co., 2717 First Ave. S.E., Cedar Rapids, Iowa.
TURNER, John P., Jr. (*M* 1942) Lt. (jg) U.S. N.R., 430 West 116th St, New York, N. Y.
TURNO, Walter G. W. (*M* 1917; *A* 1912) 71 Lafayette Ave., East Orange, N. J.
TUSCH, Walter (*M* 1917) 881 Sterling Pl., Brooklyn 16, N. Y.
TUTHILL, Arthur F.* (*J* 1940; *S* 1938) 2nd Lt., Army Air Corps, Foster Field, Texas
TUTSCH, Rodney J. (*A* 1943; *J* 1939) U. S. Army (Address unknown).
TUTT, Richard D. (*J* 1942) 5845 Haverford Ave., Indianapolis, Ind.
TUTTLE, Albert J. (*A* 1944) Draftsman, 12 Welsford St, Shepparton, Victoria, Australia
TUTTLE, Arthur A. (*A* 1944) Chief of Operations (Post Engineer) U. S. Army, Post Engineers Office, A. P. O. 863, c/o Postmaster, New York, N Y.
TUTTLE, G. H.* (*M* 1937; *A* 1936; *J* 1934) Htg. Engr., The Detroit Edison Co., 2000 Second Ave, Detroit 26, Mich.
TUTTLE, Howard W. (*A* 1944) 18 N. Huron Ave., Columbus, Ohio
TUTTLE, J. Frank (*Life Member*; *M* 1913) Sales Agent, Warren Webster & Co., 127 Federal St., Boston, Mass
TUVE, G. L.* (*M* 1932), (Council, 1939-41; 1944) Prof of Heat-Power Engrg, Case School of Applied Science, 10900 Euclid Ave., Cleveland 6, Ohio.
TUXHORN, D. B. (*M* 1936) 4853 Sedgwick St. N W., Washington, D. C.
TWIST, C. F. (*Life Member*; *M* 1921) Pres., Ashwell-Twist Co., 967 Thomas St., Seattle 9, Wash
TWIZELL, E. W. (*M* 1937) Partner, Connolly & Twizell, Ltd, 1405 Bishop St., Montreal, Que, Canada.
TYDINGS, William F. (*M* 1942) Owner, Tydings Engineering Co, 16895 Lahser Rd., Detroit, Mich
TYKLE, Frederick G. (*M* 1943) Plant Engr, Allison Div, General Motors Corp., P. O. Box 894, Indianapolis, Ind.
TYLER, Roy D. (*M* 1928) 408 Beacon St, Boston 15, Mass.

U

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UHL, W. F. (*M* 1918) Uhl Co., 132 S. Tenth St., Minneapolis, Minn.
UHLHORN, W. J. (*M* 1920) 733 S. Highland Ave, Oak Park, Ill.
UICKER, John J. (*M* 1944) 16261 Littlefield Ave., Detroit 27, Mich.
ULLRICH, A. B., Jr. (*M* 1944) 715 Lipscomb St., Dallas, Texas.
UNDERWOOD, George T. (*A* 1944) 1602 Ripon Pl, Alexandria, Va.
UNDERWOOD, John L. (*A* 1944) Owner, John L. Underwood Co., 139 Edgewood Ave. S.E., Atlanta 3, Ga.
UPDEGRAFF, Lee (*M* 1944) Owner, Franklin Engineering Co., 406 S. Main, Los Angeles 13, Calif.
URBAN, Frank F. (*A* 1939) 6726 S. W. Burlingame Ave., Portland, Ore.
URDAHL, Thomas H.* (*M* 1930), (Council, 1940-44) Capt., U.S.N.R., 2929 Connecticut Ave, N.W., Washington, D. C.

URMSTON, Benjamin S. (*M* 1944) Mgr., Air Cond. Div., Leidy Electric Co., 201 Broad St., Phillipsburg, N. J.
USHER, Wellington J., Jr. (*M* 1942) Sales Engr., Arthur S. Leitch Co., Ltd., 1123 Bay St., Toronto, Ont., Canada.

V

VALE, Henry A. L. (*M* 1929) Managing Dir., Vale Co., Ltd., 141-143 Armagh St., Christchurch, New Zealand.
VALIQUET, H. H. (*A* 1944) 2630 N. Humboldt Ave., Milwaukee, Wis.
VAMOS, George (*M* 1944) 12 Blakey Ave., Wellington W. 3, New Zealand.
VAN ALSBURG, J. H. (*M* 1931) Lt., U.S.N.R., 2222 Eye St. N.W., Washington 7, D. C.
VANCE, Louis G. (*M* 1919) 4402 Maine Ave., Forest Park, Baltimore 7, Md.
VANDERHOOF, Austin L. (*M* 1944; *A* 1933) Dist. Repr., Warren Webster & Co., 233 Hanna Bldg., Cleveland, Ohio.
VAN NOUHUYS, Herbert C. (*A* 1942; *J* 1937) Capt., Signal Corps, 4 Royal Pl., Elberon, N J.
VAN WYNGARDEN, J. E. (*A* 1940) Owner, Day Heating Co., 863 N Liberty St., Salem, Ore.
VAUGHAN, Lillian L. (*M* 1938) Prof. of Mech. Engrg., Acting Dean, School of Engrg., North Carolina State College, Raleigh, N. C.
VERITY, Elbert W. (*M* 1944) Pres, Almirall & Co., Inc., 53 Park Pl, New York 7, N. Y
VERNON, Rex (*M* 1928; *A* 1926) Sales Promotion Mgr., Johnson Service Co, 507 E. Michigan St., Milwaukee 2, Wis
VERVOORT, Edward L. (*J* 1937; *S* 1936) Lt (jg) U. S. Navy, 49 Arizona Ave, Rockville Centre, L I., N Y.
VETLESEN, G. Unger (*M* 1930) Lt Comdr, U. S. Navy, 1 Beekman Pl, New York, N. Y
VIA, Clarence W. (*A* 1943) 18 Aqueduct Rd, Sunflower, Kans.
VINCENT, Harvard B. (*M* 1944) 3840 Sulphur Springs Rd., Ottawa Hills, Ohio
VINCENT, Paul J. (*M* 1931) Chief Engr-Owner, Paul J. Vincent Co., 2208 Maryland Ave, Baltimore 18, Md.
VINSON, Neal L. (*J* 1936, *S* 1932) 1st Lt, Corps of Engineers, Box 3007, Lowell, Ariz
VIRRILL, George A. (*A* 1940) 345 Washington Ave, New Rochelle, N Y.
VISINTINE, Edward C. (*A* 1944) Owner-Mgr, Madison Plumbing & Heating Co, 31 W First St, London, Ohio.
VITALI, Oreste J. (*A* 1944) 4811 Ashoff Pl, North Bergen, N. J.
VIVARTTAS, E. Arnold (*Life Member*; *M* 1910) 26 S. Portland Ave, Brooklyn 17, N. Y.
VOISINET, Walter E. (*M* 1930) Laboratory Mgr., Curtiss-Wright Corp, Airplane Div.-Research Laboratory, Genesee St., Buffalo 5, N Y
VOLK, George H. (*J* 1942, *S* 1940) Cpl., 36806294, 6th Photo Squadron, Jefferson Barracks, Mo.
VOLK, Joseph H. (*M* 1923) Pres-Treas, Thomas E Hoye Heating Co., 1906 W. St Paul Ave., Milwaukee, Wis.
VOLKHARDT, Aquila N. (*M* 1938) Owner, A N Volkhardt, 942 Bay St, Staten Island, N. Y.
VOLLMANN, Carl W. (*M* 1938) Pres., Linde Canadian Refrigeration Co., Ltd, 355 St Peter St., Montreal, Que, Canada.
von OTTO, Robert E. (*J* 1944) 2124 "I" St N.W., Washington 7, D. C.
vonREHBERG, Hugo L. (*M* 1942) 11 Parkton Rd., Jamaica Plain 30, Boston, Mass.
von ROSENBERG, Paul C. (*J* 1939) U. S Army (Present address unknown)
VOORHEES, Guy A. (*M* 1937) Chief Engr., Hall-Neal Furnace Co., 1324 N. Capitol Ave, Indianapolis, Ind.
VROOME, Albert E. (*M* 1932) Air Cond. Engr., Ebasco Services, Inc., 2 Rector St., New York, N. Y.
VYFF, Paul C. (*A* 1944) 3673 Olentangy Blvd., Columbus 2, Ohio.

W

WACHS, Louis J. (*A* 1936; *J* 1930) 1820 Cortelyou Rd., Brooklyn 26, N. Y.
WAECHTER, Herman P. (*A* 1930; *J* 1927) 226 Great Kills Rd., Staten Island 8, N. Y.
WAGENBRENNER, Louis H. (*M* 1944) Chief Mech. Engr., Ralph E. Phillips, 603 Architects Bldg., Los Angeles 13, Calif.
WAGGONER, Jack H. (*M* 1937) 240 Quentin Rd., Newark, Ohio.
WAGNER, Earle K. (*M* 1938) Sales Engr., The Powers Regulator Co., 2240 N. Broad St., Philadelphia 32, Pa.
WAGNER, Wilbur J. O. (*A* 1943) 731 W. Main St., Jefferson City, Mo.
WAHLIN, Bernard J. (*A* 1941) 18 Stoneleigh Rd., West Newton, Mass.
WAID, Glen H. (*A* 1937) Engrg and Field Service, Scott Valve Manufacturing Co., 15 S. Clinton St., Chicago 6, Ill.
WALDEN, Harold K. (*A* 1942; *J* 1939) CSF, U.S.N.R., 1409 Thirty-Fifth St., Belview Heights, Birmingham 8, Ala
WALDO, Harry J. (*A* 1944) Sales Repr., 332 Camp St., New Orleans 12, La.
WALDON, Charles D. (*A* 1932) 32 Ferndale Ave., Toronto, Ont., Canada.
WALDON, O. D. (*A* 1944) 5915 Norwaldo Ave., Indianapolis, Ind.
WALDREP, James E. (*A* 1943; *J* 1939) 1209 Magnolia Ave., Rome, Ga.
WALDSMITH, Earl C. (*A* 1944) 4917 Wornall Rd., Kansas City, Mo.
WALFORD, L. C. A. (*M* 1938) 515 Churchill St, Westboro, Ont., Canada
WALKER, Edmund R. (*M* 1934) 365 McKinley Ave., Kenmore, Buffalo 17, N. Y.
WALKER, J. Herbert* (*M* 1916), (2nd Vice-Pres, 1941; Council, 1925, 1938-41) Asst. to Gen Mgr, The Detroit Edison Co, 2000 Second Ave., Detroit 26, Mich.
WALKER, Kirby (*M* 1944) Consultant, 53 Broadway, Hagerstown, Md
WALKER, Theodore R. (*M* 1944) Vice-Pres, Columbus Heating and Ventilating Co, 182 N Yale Ave, Columbus, Ohio.
WALKER, Wythe F. (*A* 1941) 3637 Sixth Ave., Los Angeles 16, Calif.
WALLACE, George J. (*Life Member*, *M* 1923) 27-36 Ericsson St., East Elmhurst, N. Y.
WALLACE, James B. (*A* 1944) 9 Byfield Lane, Dearborn, Mich.
WALLACE, William M., II (*M* 1929) 1505 Duke University Rd., Durham, N. C.
WALLIS, Walter M. (*A* 1940) 11225 Tenth Ave. S W, Seattle, Wash.
WALSH, Edward R., Jr. (*M* 1936; *A* 1935) Wyndham Hills, York, Pa.
WALSH, James A. (*M* 1943) Pres., J. A. Walsh & Co, Inc, P. O Box 1773, Houston, Texas
WALSH, John W. (*M* 1944) Pres-Treas, John W. Walsh, Inc., 2 Foster St, Worcester 8, Mass.
WALTER, Burwell J. (*M* 1945) Sales Engr., R L. Deppmann Co, 5853 Hamilton Ave, Detroit 2, Mich.
WALTERS, Arthur L. (*M* 1926; *A* 1925; *J* 1924) 704½ Hinman Ave., Evanston, Ill.
WALTERS, Joseph K. (*M* 1943) Greenville, Del
WALTERS, Thomas A. (*M* 1944) 19359 San Juan Drive, Detroit 21, Mich.
WALTERS, William T. (*M* 1917) 12747 Wallace St, Chicago, Ill.
WALTERTHUM, John J. (*A* 1922) 42-A Van Reipen Ave., Jersey City, N. J
WALTON, Charles W., Jr. (*M* 1934) 422 Coleman St, Chippewa Falls, Wis.
WALZ, Chester D. (*A* 1939) 6621 W. Sixth St, Los Angeles 36, Calif.
WALZ, George R. (*A* 1944) Application Engr., Minneapolis-Honeywell Regulator Co, 1101 Vermont Ave N.W., Washington, D. C.
WANGSGAARD, Dee (*A* 1942) 1st Lt., N.M.B, A.U S., 167 S. Main St, Logan, Utah.
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WARD, Cyril B. (*M* 1944) 626 Race, Denver 6, Colo.
WARD, Frank J. (*M* 1935) Owner, The Frank J. Ward Co., 237 W. Court St., Cincinnati, Ohio.
WARD, Joseph A. (*M* 1944) Sunset Rd, Pompton Plains, N. J.

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WARREN, Clarence N. (*M* 1943) Vice-Pres.-Asst. Mgr., Hayes Brothers, Inc, 236 W. Vermont St, Indianapolis, Ind

WARREN, Francis C. (*M* 1934) 329 Gladstone Ave, S E, Grand Rapids 6, Mich.

WARREN, Hugh P., Jr. (*J* 1943; *S* 1940) 2309 Aster St, Fort Worth 3, Texas.

WARREN, Robert M., Jr. (*A* 1943; *J* 1938) Lt, Army Air Force, 313 N. Main St, Sumter, S. C.

WASHINGTON, L. W. (*M* 1929) Creekdale Farm, R. R. 2, Pecatonica, Ill.

WASSON, Robert A. (*M* 1938) Asst Gen. Mgr., Clarage Fan Co., Kalamazoo 16, Mich.

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WATERMAN, John H. (*M* 1931) Engr, Charles T Main, Inc., 201 Devonshire St, Boston 10, Mass

WATERS, G. G. (*M* 1931; *A* 1926) Dist. Mgr., American Blower Corp, 1841 Oliver Bldg., Pittsburgh 16, Pa.

WATKINS, George B. (*A* 1936) Dir of Research, Libby-Owens-Ford Glass Co, 1701 E Broadway, Toledo, Ohio.

WATKINS, J. Z. (*A* 1944) Merchandising Mgr, Rulan Gas Co., P. O. Box 29, Charlotte, N C

WATKINSON, Gerald H. (*A* 1944) Defense Industries, Ltd, 1155 Phillips Pl, Room 123, Montreal, Que., Canada.

WATSON, Charles E. W. (*M* 1944) Pres, John Watson & Co, Ltd., 1359 Greene Ave, Westmount, Que, Canada.

WATSON, Gerald M. (*S* 1941) Lt, H.D.C, Ft Stevens, Ore.

WATSON, William W. (*A* 1942) Box 91, Clarkston, Mich

WATT, Robert D. (*A* 1943; *J* 1937) 3617 47th N.E., Seattle, Wash.

WATTERSON, William B. (*A* 1944) 1183 Brentwood Rd, Cleveland Heights, Ohio.

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WATTS, L. Copeland (*M* 1943) Cons. Htg-Vtg Engr., J. Roger Preston and Partners, Dilke House, Malet St., London, W.C.1, England

WAY, James B. (*A* 1945) Mgr, Equipment Dept., Vipond-Tolhurst, Ltd, 845 Querbes Ave., Outremont, Montreal 8, Canada

WAY, Norman C. (*M* 1945) Engr., Way Engineering Co., P. O. Box 8066, Houston, Texas.

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WEATHERBY, Edward P., Jr. (*A* 1944; *J* 1936; *S* 1935) Major, Army Air Force, 518 Martinique St., Dallas 10, Texas.

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WEAVER, J. v. O. (*M* 1940) Lt. Col, Air Corps, 1017 Laurel St., Elkhart, Ind.

WEBB, Ernest C.* (*M* 1935) Engrg. Service Mgr., Iron Fireman Manufacturing Co., 3170 West 106th St., Cleveland 11, Ohio.

WEBB, John S. (*M* 1945) W. D. Cashin Co., 69 A St., South Boston 27, Mass.

WEBB, John W. (*M* 1926) Managing Dir., Webb Dust Removing & Drying Co., Vinery Works, Town Lane, Denton, N. Manchester, England.

WEBBER, Charles H. (*A* 1940) Sales Engr., Pacific Scientific Co., 1430 Grande Vista Ave., Los Angeles, Calif.

WEBER, Erwin L. (*M* 1936) Cons. Engr., 632 Medical Arts Bldg., Seattle, Wash.

WEBER, Eugene F. (*A* 1940; *J* 1937) 607 Forest Court, Clayton 5, Mo.

WEBER, Frank J. (*M* 1943) Owner, Frank J. Weber & Associates, 443 Delaware Ave., Buffalo, 2, N. Y.

WEBSTER, Chester C. (*A* 1940) Pres., John Hankin & Brother, 120 Greenwich St., New York 6, N. Y.

WEBSTER, E. Kessler (*M* 1915) Pres., Warren Webster & Co, 1625 Federal St., Camden, N. J.

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WOODBRIDGE, William L. (*A* 1944) Owner, Woodbridge Plumbing & Heating, R F. D 3, 610 Farr St., Milford, Mich
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Samuel Burns
New York, N. Y.
B. Harold Carpenter
Wilkes Barre, Pa.
R. C. Carpenter
Ithaca, N. Y.
Albert A. Cary
New York, N. Y.
Robert C. Clarkson
Philadelphia, Pa.
Geo. B. Cobb
New York, N. Y.
H. D. Crane
Cincinnati, Ohio
Albert A. Cryer
New York, N. Y.
T. B. Cryer
Newark, N. J.
Mark Dean
Boston, Mass.
John Demarest
Boston, Mass.
Thos. J. Douglass
Norwich, Conn.
A. C. Edgar
Philadelphia, Pa.
Hermann Eisert
Baltimore, Md.
John A. Fish
Boston, Mass.
Frank W. Foster
Boston, Mass.
C. F. Gessert
New York, N. Y.
Judson A. Goodrich
New York, N. Y.
John Gormly
Philadelphia, Pa.
James A. Harding
New York, N. Y.
L H. Hart
New York, N. Y.
Charles F. Hauss
New York, N. Y.
Jno. D Hibbard
Chicago, Ill
William H. Hill
New York, N Y.
Geo. D. Hoffman
Chicago, Ill.
Charles S. Hopkins
Rochester, N. Y.
Alfred A. Hunting
Boston, Mass
Stewart A. Jellett
Philadelphia, Pa.
J. H. Kinealy
St. Louis, Mo.
Jos. A. Langdon
Pittsburgh, Pa.
Chas. W. Light
Saginaw, Mich.
H. E. Light
Saginaw, Mich.
Chas. C. Lincoln
New York, N. Y.
C. K. Longenecker
New York, N. Y.
James Mackay
Chicago, Ill.
Wm. M. Mackay
New York, N. Y.
A. S. Mappett
New York, N. Y.
Wm. McMannis
New York, N. Y.
George L. Mehring
Chicago, Ill.
Edward A. Munro
Brooklyn, N. Y.
Robert Munro
Pittsburgh, Pa.
Charles W. Newton
Baltimore, Md
Theodore C. Northcott
Elmira, N. Y.
Charles S. Onderdonk
Philadelphia, Pa.
John A Payne
Providence, R. I.
John H. Petherick
Chattanooga, Tenn
Geo. W. Plastow
Jersey City, N J
Henry B. Prather
Buffalo, N. Y
Leon H. Prentice
Chicago, Ill
D. M. Quay
Chicago, Ill
Wm. A. Russell
New York, N. Y
U. G. Scollay
Brooklyn, N Y
Percival H. Seward
New York, N. Y
Le Roy B. Sherman
New York, N. Y
Fred P. Smith
New York, N. Y
B. F. Stangland
New York, N Y.
Geo. P. Steel
Philadelphia, Pa
Joseph M. Stoughton
Yonkers, N. Y
H. M. Swetland
New York, N. Y.
Geo H. Underhill
Boston, Mass.
T. J. Waters
Chicago, Ill
J. R. Wendover
New York, N. Y
W. B. Wilkinson
New York, N. Y.
James R. Willett
Chicago, Ill
J. J. Wilson
Troy, N Y
Wiltsie F. Wolfe
Boston, Mass

A.S.H.V.E. MILESTONES

1894 Organization meeting September 10, at Broadway Central Hotel, New York, with 75 Charter Members.

1895 First Annual Meeting, January 22-24 at 12 West 31st Street, New York. Incorporation under laws of New York State, TRANSACTIONS established.

1896 Society Emblem adopted at 2nd Annual Meeting—Dr. J. S. Billings, First Honorary Member.

1897 First Semi-Annual Meeting held at the Windsor Hotel, New York, June 18.

1906 First Chapter organized in Chicago.

1907 Membership reached 301.

1911 Membership totaled 405. New York Chapter formed. Society established Headquarters in Engineering Societies Building, 29 West 39th Street, New York.

1915 Journal first published in April as a quarterly; later issued monthly.

1916 Full time Secretary employed.

1918 Members in Armed Services—64.

1919 After several years of consideration and planning the Society established its Research Laboratory, in the United States Bureau of Mines, at Pittsburgh, Pa., having determined a preliminary *modus operandi* and provided an operating fund. The Society is justly proud that it is the only professional engineering organization which maintains and operates its own Research Laboratory.

1922 The A.S.H.V.E. GUIDE published in September.

1923 Comfort Zone Established by A.S.H.V.E. Research.

1925 Dues raised to $25.00 and 40 per cent of dues of members and associates allocated to Research Fund.

1927 Membership of Council increased from 12 to 17.

1928 Appointment of Technical Secretary and completion of Code for Minimum Requirements for Heating and Ventilation of Buildings. Benjamin Franklin honored as Patron Saint of Society.

1929 Journal incorporated as a Special Section of *Heating, Piping and Air Conditioning.* Society Headquarters moved to 51 Madison Ave.

1930 Endowment created by Thornton Lewis for the F. Paul Anderson Medal. First International Heating and Ventilating Exposition held in Philadelphia.

1932 First Award of F. Paul Anderson Medal to W. H. Carrier.

1933 New Constitution and By-Laws adopted—Membership dues reduced.

1938 Membership passed 3000.

1941 Society requested to do Research for U. S. Navy.

1943 Membership at all-time high, 3300—Chapter with largest Charter Membership organized in Indianapolis.

1944 50th Anniversary.

1945 Members serving in Armed Forces—416.

Meetings of
AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

Year	Annual Meeting	Date	Place	Semi-Annual Meeting	Place
1895	1st	Jan. 21-23	New York, N. Y.	None	
1896	2nd	Jan. 21-23	New York, N. Y.	None	
1897	3rd	Jan. 26-28	New York, N. Y.	June 18	New York, N. Y.
1898	4th	Jan. 25-27	New York, N. Y.	July 15	Atlantic City, N J.
1899	5th	Jan. 24-26	New York, N. Y.	None	
1900	6th	Jan. 23-25	New York, N. Y.	None	
1901	7th	Jan. 22-24	New York, N. Y.	July 12-13	Chicago, Ill.
1902	8th	Jan. 21-23	New York, N. Y	June 16	Atlantic City, N. J.
1903	9th	Jan. 20-22	New York, N. Y.	July 17-18	Niagara Falls, N Y.
1904	10th	Jan. 19-21	New York, N. Y.	July 15-16	Detroit, Mich.
1905	11th	Jan. 17-19	New York, N. Y.	July 7-8	Chicago, Ill.
1906	12th	Jan. 16-18	New York, N. Y.	July 19-20	Chicago, Ill.
1907	13th	Jan. 22-24	New York, N. Y.	July 18-19	Milwaukee, Wis.
1908	14th	Jan. 21-23	New York, N. Y	July 24-25	Niagara Falls, N. Y
1909	15th	Jan. 19-21	New York, N. Y.	July 15-16	Indianapolis, Ind
1910	16th	Jan. 18-20	New York, N Y	June 30-July 1	St. Louis, Mo.
1911	17th	Jan. 24-26	New York, N. Y.	July 6-8	Chicago, Ill.
1912	18th	Jan. 23-25	New York, N. Y.	July 11-12	Detroit, Mich.
1913	19th	Jan 21-23	New York, N. Y.	July 17-19	Buffalo, N Y.
1914	20th	Jan. 20-23	New York, N. Y	July 9-11	Cleveland, Ohio
1915	21st	Jan. 20-22	New York, N. Y	Sept. 16-17	Atlantic City, N. J.
1916	22nd	Jan. 18-20	New York, N. Y.	July 19-21	Detroit, Mich
1917	23rd	Jan 16-18	New York, N Y	July 18-20	Chicago, Ill.
1918	24th	Jan 22-24	New York, N. Y	June 26-28	Buffalo, N. Y
1919	25th	Jan. 28-30	New York, N. Y.	June 10-12	Pittsburgh, Pa
1920	26th	Jan. 27-29	New York, N. Y	May 26-28	St Louis, Mo
1921	27th	Jan. 26-28	Philadelphia, Pa.	June 14-17	Cleveland, Ohio
1922	28th	Jan. 24-26	New York, N Y.	June 6-7 June 8-10	Buffalo, N Y Detroit, Mich
1923	29th	Jan 23-26	New York, N Y and Washington, D. C.	May 21-23	Chicago, Ill
1924	30th	Jan 22-25	New York, N. Y.	June 19-22	Kansas City, Mo
1925	31st	Jan. 27-30	New York, N. Y. and Boston	June 15-17	Atlantic City, N J.
1926	32nd	Jan. 28-30	Buffalo, N Y	May 26-28	Lexington, Ky
1927	33rd	Jan. 26-28	St. Louis, Mo.	June 28-30	White Sulphur Springs
1928	34th	Jan. 23-25	New York, N. Y.	June 26-29	West Baden, Ind
1929	35th	Jan. 28-31	Chicago, Ill	June 26-28	Bigwin Inn, Ont.
1930	36th	Jan. 27-31	Philadelphia, Pa	June 24-27	Minneapolis, Minn
1931	37th	Jan. 26-29	Pittsburgh, Pa	June 22-25	Swampscott, Mass
1932	38th	Jan. 25-29	Cleveland, Ohio	June 27-29	Milwaukee, Wis.
1933	39th	Jan. 23-25	Cincinnati, Ohio	June 22-24	Detroit, Mich
1934	40th	Feb 5-9	New York, N. Y	June 20-22	Buck Hill Falls, Pa
1935	41st	Jan. 28-30	Buffalo, N. Y.	June 16-19	Toronto, Ont
1936	42nd	Jan. 27-30	Chicago, Ill	June 22-24	Buck Hill Falls, Pa.
1937	43rd	Jan. 25-27	St. Louis, Mo	June 24-26	Swampscott, Mass
1938	44th	Jan. 24-28	New York, N Y	June 20-23	Hot Springs, Va
1939	45th	Jan 23-26	Pittsburgh, Pa.	July 4-6 *Oct. 30-31	Mackinac Isl., Mich. Atlanta, Ga
1940	46th	Jan. 23-26	Cleveland, Ohio	June 17-19 *Oct. 14-15	Washington, D C. Houston, Tex.
1941	47th	Jan. 27-29	Kansas City, Mo.	June 17-19	San Francisco, Calif.
1942	48th	Jan. 26-28	Philadelphia, Pa	June 15-17	St. Paul, Minn.
1943	49th	Jan 25-27	Cincinnati, Ohio	June 7-8	Pittsburgh, Pa
1944	50th	Jan. 31-Feb. 1-2	New York, N. Y.	June 19-20	Grand Rapids, Mich
1945	51st	Jan. 22-24	Boston, Mass.		

*Fall Meeting